

Mass Optimization Method for Automotive Sheet Metal Components based on Load Path Analysis using Bonded Steel Patches

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Abstract

For mass optimization of sheet metal components in automotive applications, reducing the base sheet thickness and locally reinforcing with a second sheet metal layer offers significant potential for more efficient material use. This thesis presents a novel finite-element-based method for designing such patched components. The method considers sheet metal manufacturability constraints and combines load-path analysis using the *U* Inspection Load Method* with elements of graph theory (*Dijkstra algorithm*). As a result, geometrical constraints can be explicitly incorporated by enabling the re-routing of load paths from load introduction to support locations. The formulation is extended by a discretization model for patch geometry, creating uniform-thickness patch geometries and calculating mass reduction potential. The manufacturing of the resulting patched components by adhesive bonding is evaluated with respect to fatigue strength. The method is validated through the optimization of a serial automotive wishbone link from a Toyota Camry front suspension. For this component, lightweight versions achieving 15 % (410 g) and 21 % (560 g) mass reduction were built and tested against the serial part.

Für die Massenoptimierung von Blechbauteilen im Automobilbereich bieten das Reduzieren der Basismaterialstärke und die lokale Verstärkung durch eine zweite Blechlage erhebliches Potenzial für eine effizientere Materialnutzung. In dieser Arbeit wird eine neuartige finite-elemente-basierte Methode zur Auslegung solcher gepatchter Bauteile vorgestellt. Die Methode berücksichtigt Fertigungsbeschränkungen von Blechen und kombiniert die Lastpfadanalyse mittels der *U* Inspection Load Method* mit Elementen der Graphentheorie (*Dijkstra-Algorithmus*). Dadurch können geometrische Einschränkungen explizit berücksichtigt werden, indem alternative Lastpfade von den Lastangriffspunkten zu den Auflagestellen gebildet werden. Die Vorgehensweise wird durch ein Diskretisierungsmodell für die Patch-Geometrie erweitert, wodurch einheitliche Patchdicken erzeugt und das Potenzial zur Massenreduktion berechnet werden können. Die Fertigung dieser Blechverstärkungen durch klebtechnische Verfahren für schwingungsbelastete Fahrwerksteile wird untersucht und an einem Serienbauteil aus dem Automobilchassis, einem Vorderachsquerlenker eines Toyota Camry, validiert. Für dieses Bauteil wurden Leichtbauvarianten mit 15 % (410 g) bzw. 21 % (560 g) Gewichtseinsparung gefertigt und vergleichend zum Serienbauteil getestet.

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Previous Publications

Resulting components based on this project's method are patented in German patent DE 20 2022 104 853.3 [NKP22].

Contents

1	Introduction	1
1.1	Structure of this Thesis	2
2	State of the Art	5
2.1	Sheet Metal Application and Manufacturing	5
2.1.1	Sheet Metal in Automotive	5
2.1.2	Sheet Metal Forming	6
2.1.3	Joining of Sheet Metal	8
2.2	Lightweight Design for Sheet Metal	13
2.2.1	Material Lightweighting	13
2.2.2	Regional Material and Shape-Wise Lightweighting	14
2.3	Sheet Thickness Optimization	16
2.3.1	Geometry Optimization with Local Concept	16
2.3.2	Geometry Optimization with Global Concept	19
2.4	Path Optimization (Graph Theory)	22
3	Discussion and Requirements	27
3.1	Comment on the Current State of the Art	27
3.2	Requirements Derivation for a Patch Optimization Method	28
4	Design Method	33
4.1	Core Concept	33
4.1.1	Basic Principle	33
4.1.2	Incorporating the Requirements	36
4.2	Function Structure	38
4.2.1	Prepare Data	39
4.2.2	Find Paths	41
4.2.3	Create Patches from Paths	45
4.2.4	Tune Thickness	48

5	Software Implementation	51
5.1	Software Architecture	52
5.2	Function Structure	53
5.2.1	Geometry and Constraint Definition	54
5.2.2	Preprocessor	55
5.2.3	U^* FEA	56
5.2.4	Path Processor	58
5.2.5	Patch Processor	60
5.2.6	Thickness Processor	61
5.2.7	Data Export	64
5.3	Interfaces	64
5.3.1	Input Interface	64
5.3.2	Graphical Interface	65
5.3.3	FEA Interface	66
5.3.4	Output Interface	67
6	Methods for Patch Bonding	69
6.1	Adhesive Selection	69
6.2	Adhesive Testing	70
6.2.1	Application	70
6.2.2	Static Lap Shear Testing	71
6.2.3	Fatigue Lap Shear Testing	73
6.2.4	Conclusions for Adhesive Selection	77
7	Validation	79
7.1	Component Selection	79
7.2	Reverse Engineering	81
7.3	Loading	82
7.4	Mass Optimization by Patch Optimizer	86
7.4.1	Geometry Definition	86
7.4.2	Loads and Constraints	86
7.4.3	Restricted Areas	88
7.4.4	U^* results	89
7.4.5	Path Results	89
7.4.6	Patch Results	91
7.4.7	Thickness Tuning Results and Discussion	92
7.5	Derivation of Prototypes from Optimizer Results	96
7.6	Virtual Validation	97
7.6.1	Simulation Setup	98
7.6.2	Simulation Results	98

7.6.3	Discussion on Simulation Results	100
7.7	Prototype Manufacturing	102
7.7.1	Forming of the Main Parts	102
7.7.2	Joining of the Main Parts and Bushings	104
7.7.3	Patch Manufacturing	105
7.7.4	Patch Joining	106
7.7.5	Prototype Build Results	108
7.8	Experimental Validation	108
7.8.1	Test Rig Design	109
7.8.2	Static Test Setup	110
7.8.3	Static Test Results	111
7.8.4	Static Test Discussion	115
7.8.5	Fatigue Test Setup	117
7.8.6	Fatigue Test Results	118
7.8.7	Fatigue Test Discussion	120
8	Summary	121
8.1	Outlook	122
A	Virtual Validation Figures	125
B	Prototype Figures	129
C	Experimental Validation Figures	133
	Methodological Remarks	147

CONTENTS

List of Figures

2.1	Schematic of a forming tool [Dix20]. A workpiece is held between a blank holder and a die, and then plastically deformed by a punch.	6
2.2	Forming limit curve for plane stress state. Own illustration based on [Cha15]. The plane strain deformation of a sheet ought to stay within the grey area during forming. Overly high compression may lead to wrinkling, while excessive stretching leads to tearing.	7
2.3	Inversely proportional relationship of ultimate tensile strength and elongation at break for first generation advanced high strength steels [ST15]. . . .	8
2.4	Typical joint shapes of selected permanent joining methods used for sheet metal components.	9
2.5	Shear stress in the adhesive on lap shear test [Hab09]. The stress reaches a maximum on the bond edge in load direction.	11
2.6	Tensile stress in the sheets on lap shear test [Hab09]. Stress decreases gradually in the bonded area without any peaks.	11
2.7	Schematic of a base sheet with glued patch under tensile load with tensile stresses in the sheets and shear stress in the adhesive shown. Load gets transferred to the patch via the adhesive, until equally shared between base and patch based on their stiffness ratio. After a transition zone on the edges, where load is not distributed yet, this package then behaves as monolithic.	12
2.8	Three methods of structural optimization, from top to bottom: Sizing, Shape, Topology. left image shows initial, right the optimized state [BS04].	17
2.9	Topology optimization performed for a simple structure with one load and two supports. Penalization shows a convergence to a black-white result while the unpenalized case allows intermediate (grey) densities. Computed with DTU's TopOpt software [ANJAS12].	18
2.10	General description of a structure with load paths P_l visualized. The paths connect areas of load application l to supports s over domain Ω	20
2.11	Basic concept of U^* index.	21
2.12	Concept of how to calculate U^* for point x	21

2.13	Alternative calculation using the <i>inspection load method</i>	21
2.14	Dijkstra's algorithm on a simple network searching the shortest path from A to F. Each node stores a tentative distance and a predecessor pointer. In each iteration, the open node with the smallest tentative distance is selected and its neighbors are evaluated. Open nodes are white, the currently processed node is gray, and closed nodes are black.	23
2.15	Schematic of region searched before finding shortest path from l to target s on a uniform rectangular network for path finding algorithms. While Dijkstra's Algorithm searches in all directions until finding a target, A^* Algorithm searches in target direction first, thus being potentially far more efficient. However, if several targets exist, it cannot be applied.	25
4.1	Arrangement of $m = 2$ parallel "legs", consisting of $n = 2$ in-line truss-elements each that transfer a load to the support. Altering individual stiffnesses changes the load share of the legs.	34
4.2	General function structure of the patch optimizer, consisting of 4 modules. From an initial geometry, an optimized geometry with reduced overall mass is computed.	38
4.3	Function structure for patch optimizer module 1, Prepare Data. Here, the model is prepared and necessary information given to run the optimization.	39
4.4	Grid over a mesh with closest nodes to the grid's vertices selected. This schematic explains the selection process for achieving a spatially uniform selection of nodes, even if the nodes themselves are not spread uniformly.	41
4.5	Function structure for patch optimizer module 2, Find Paths. Here, the desired main load paths are computed from U^* -analysis, restricted areas and path-finding algorithms.	41
4.6	"Velocity" ν over gradient ∇ , affecting the expense for traveling a path. A high negative gradient causes high velocity. Positive gradients cause low velocities.	44
4.7	Function structure for patch optimizer module 3, Create Patches from Paths. The paths determined before get discretized into patched regions based upon path properties width, density and a correction factor.	46
4.8	Width $w(x(p_l))$ being a function of location x on a path p_l . This allows the translation of a path into a patch, with width perpendicular to path direction.	46
4.9	Qualitative description of influence factors on patch width: Gradient and path density. Together, these factors control patch width along paths.	47

4.10	Orientation of grow direction for path width. Grow is perpendicular to the path direction. Curvature of the surface is accounted for by growing in a intersecting plane perpendicular to path direction and having the patch width be the arc length.	48
4.11	Function structure for patch optimizer module 4, Tune Thickness. From the patches made in module 3, thicknesses get processed to achieve minimal overall mass but maintaining component performance, i.e. maximum stress, to the initial design for the given load case, the baseline performance. Thickness is iteratively decreased and evaluated until performance matches the baseline performance.	49
4.12	Plot of resulting optimal volume ratio $Q_V = V_1/V_0$ over base thickness t_b and width factor k_w . the optimal combination of parameters can be observed by interpolation of the response surface and determining the minimum.	50
5.1	Patch optimizer software architecture. Highlighted in grey is developed within this project. Left: Function group. Middle: Functional blocks. Right: Environment used.	52
5.2	Function structure of the patch optimizer software. It follows the logic from the general method function structure, see figure 4.2.	53
5.3	Function structure patch optimizer software module 1, Geometry and Data Preparation. Here, the geometry is meshed and all user input data is transferred into the system.	54
5.4	Function structure patch optimizer software module 2, Preprocessor. Here, from the input geometry, the FEA-Model for U^* -computation is developed. This includes mapping of neighboring elements, grid-based computation node selection and constraint creation.	56
5.5	Function structure patch optimizer software module 3, U^* FEA. This part of the process runs mainly in MAPDL, until the point where results for the computation nodes are finished. They get re-imported into MATLAB and there interpolated to all elements.	57
5.6	198 U-Star-Nodes selected on an exemplary geometry. Calculated nodes are highlighted in red, while the subsequent interpolation of their results to all elements is mapped by the element color.	58
5.7	Function structure patch optimizer software module 4, Path Processor. Here, from U^* -data, restricted elements and boundary conditions, efficient main load paths can be determined.	58
5.8	Function structure patch optimizer software submodule 4.2, Dijkstra's Algorithm. Beginning from start elements, expense to reach neighboring elements is evaluated, until the least expensive path to a target is found.	59

5.9	Function structure patch optimizer software module 5, Patch Processor. Paths get analyzed w.r.t density and gradient to determine width. With that, paths are grown to become patches, and merged.	60
5.10	Smoothing of a path leading across discrete elements centers. The smoothing leads to less jitter in the path direction and decreases mesh dependency.	61
5.11	Function structure patch optimizer software module 6, Thickness Processor. Performance of patched and unpatched state is compared and minimal required patch thicknesses evaluated.	62
5.12	Function structure patch optimizer software submodule 6.1, Get Initial Performance. Performance for the initial, unpatched state, is evaluated by finite element analysis.	62
5.13	Function structure patch optimizer software submodule 6.2, Get Patched Performance. Performance for the patched state is evaluated by finite element analysis.	63
5.14	Function structure patch optimizer software submodules 6.1.2 / 6.2.3, Performance FEA. This process is the same for the unpatched (6.1) and patched state (6.2). Finite element analysis is performed in MAPDL and the results reimported.	63
5.15	Function structure patch optimizer software module 7, Data Export. The optimized geometry is exported as stl model and achieved volume decrease is given.	64
6.1	Curing fixture for adhesive tape. Specimens (1) and tape (2) are aligned in the fixture (3) with parallelism maintained by spacers (4). Clamping force is applied by weights (5).	71
6.2	Achieved shear strength τ for adhesives (B) Dupont Betamate 1630, (H) Henkel Loctite EA 9514 for bond thicknesses 0.1, 0.2 and 0.3 <i>mm</i> and 3M SAT 1100S tape with 0.4 <i>mm</i> tape thickness. Mean μ and standard deviation σ for 5 samples each.	72
6.3	Specimen faces after lap shear test. Betamate 1630 (a) shows little adhesive failure indicated by the metal surface visible in the joint area. Loctite EA 9514 (b) shows adhesive failure of up to one third of the bonded area. 3M SAT 1100S tape shows no adhesive failure.	72
6.4	Setup for fatigue testing 3M SAT 1100S tape. The specimens are held in a way to not induce any undesired moments and achieve pure shear in the adhesive. This is achieved by universal joints above and below the specimens, and support plates on the specimen package to counteract peeling stress from specimen bending.	73

6.5	Hydraulic universal test bench for performing the pulsating fatigue testing of adhesive specimens with the specimen clamping rig mounted to it. Force is exerted by the vertical hydraulic piston above the rig and measured by a pressure sensor below it.	74
6.6	SN-Curve for 3M SAT 1100S 0.4 <i>mm</i> tape from fatigue testing. Shear stress amplitude at $R_\tau = 0.1$. Moderate spread in the results and a good predictability are observed.	75
6.7	SN-Curve for Henkel Loctite EA 9514 0.1 <i>mm</i> from fatigue testing. Shear stress amplitude at $R_\tau = 0.1$. High spread in the results and a slightly worse predictability compared to the tape are observed.	76
7.1	Toyota Camry 2012 model and its front suspension, based on [MBP ⁺ 14]. .	80
7.2	Toyota Camry wishbone link consisting of two main shells (ET1 and ET2), cylindrical bushing (ET3) and spherical bushing (ET4).	81
7.3	CT scan of Toyota Camry wishbone link [BKPN21]. The clearance between ET1 and ET2, as well as the weld geometry, can be observed.	82
7.4	CARLOS-Multi level crossing frequencies for lateral and longitudinal forces, [HKN24] based on [SKH94].	83
7.5	Wheel forces directions [HKN24].	83
7.6	Toyota Camry McPherson suspension detail, [HKN24] based on [MBP ⁺ 14].	83
7.7	Principal stress vectors for load cases as in table 7.3 from left to right [HKN24]. While some load cases show compressive stress and others tensile stress, the direction shows only minor deviations.	85
7.8	Maximum equivalent stress locations for all load cases as listed in table 7.3 [HKN24]. Depending on the load case, the maximum stress occurs either at the right or left shoulder of B. In this figure, mounting region B is not shown. In both cases, the stress concentration is located at the weld ends.	85
7.9	Wishbone link section with shells and weld indicated. The real geometry is abstracted to a singular shell for the optimization software.	86
7.10	Wishbone link mesh and constraints. At A, the spherical bushing is represented by fixing the indicated circle's center. At C, the cylindrical bushing is represented by allowing movement along the axis of C-A. At B, the load is applied.	87
7.11	Wishbone link with ball joint attachment at B [HKN24].	88
7.12	Wishbone link weld seam elements for restricted element set.	88

7.13	U^* results on wishbone link with 504 computed nodes. Red dots show the computed nodes and the coloring shows their interpolation to all elements. As would be expected, elements closer to B are more connected to it, and thus, show a high value close to 1. Elements further away decrease in U^* -index with respect to B.	89
7.14	Wishbone link with paths highlighted for the restricted case. Paths do not cross the weld seam, as that region is restricted, but go alongside it.	90
7.15	Wishbone link paths highlighted for unrestricted case. Some paths cross the weld seam, as it is not restricted here. Still, most paths go alongside it.	90
7.16	Wishbone link patches restricted case at width $k_w = 20$. The patch does not cross the restricted elements. Instead, patches grow away from it. Two separate patches form, as they cannot connect to each other due to the restricted weld seam. Both patches exhibit a rather simple and straight shape.	91
7.17	Wishbone link patches unrestricted case at width $k_w = 20$. One singular patch forms, that connects over the weld seam halfway from B to C. Apart from issues with bonding this patch above the irregular surface of the weld seam, this shape is also more complex to form as it grabs around the geometry.	92
7.18	Wishbone link equivalent stress for initial, unpatched state. The maximum stress hotspot can be seen on the left shoulder at B (red circle) and has a magnitude of $g_0 = 324 \text{ MPa}$. Another hotspot lies on the right shoulder of B (orange circle) with a magnitude of approximately 275 MPa	93
7.19	Response Surface for the wishbone link volume reduction Q_V over base thickness t_b and width factor k_w . Both variants behave very similar. Although heavily constrained, the restricted case shows a comparable maximum mass reduction potential, highlighting the method's ability to handle restrictions.	94
7.20	Wishbone link equivalent stress for restricted state. The stress hotspot did not change from the initial, unpatched state, but also did not increase in magnitude even though the base thickness at that spot decreased and no patching on that very spot was performed.	95
7.21	Wishbone link equivalent stress for unrestricted state. Likewise, the stress hotspot did not change from the initial, unpatched state, but also did not increase in magnitude even though the base thickness at that spot decreased and no patching on that very spot was performed.	95
7.22	Wishbone link 2.2 mm variant patched. The patch follows the computed design with some smoothing on edges and fill of the contact area to the later added ball joint at B. The adhesive is modeled as a separate solid with 0.2 mm thickness.	97

7.23	Probe on the solved model for type B design. The probe is located on the right shoulder at B, close to the weld end. Equivalent stress depicted. The probe is selected as the local maximum in that region.	99
7.24	Load-deflection behavior of types A, B and C under same load. Specimens type B and C show considerably lower stiffness.	100
7.25	Wishbone link prototype manufacturing process. First, the main parts ET1 and ET2 are formed. They are joined with the bushings ET3 and ET4. Then, after forming it, the patch is attached to the base part.	102
7.26	Wishbone link manufacturing operation OP1, forming base ET1 and ET2. The sub-steps are the same for both shells, consisting of forming and trimming operations.	103
7.27	Wishbone link preforming step OP1.1 lower die with blank outline. Here, the overall curvature, contour at region B, and the shoulder between A and C are pre-formed on a large blank to achieve uniform thinning in those regions and counteract wrinkling.	104
7.28	Wishbone link forming step OP1.3 lower die with blank outline. After trimming the parts, here they get formed to their final shape. The preformed contours from OP1.1 help align the sheets. The whole edge is formed down.	104
7.29	OP2.1 Wishbone link welding fixture. Again, the two halves are forced to target position, counteracting any shape deviations. That way, the wishbone link may spring back slightly after welding, but most deviations are canceled out from permanently joining the two parts and the cross sectional area of all prototypes is consistent.	105
7.30	Wishbone link after OP2.2 welding, 2.2 <i>mm</i> variant.	105
7.31	Wishbone link 2.2 <i>mm</i> variant patch forming tool. Only a single-step forming is required.	106
7.32	Adhesive applied to the lower 1.2 <i>mm</i> patch for specimen B1. 0.2 <i>mm</i> steel spacers can be seen embedded within. The amount of glue is 4 <i>ml</i> , corresponding to an overall bonding thickness of 0.2 <i>mm</i> with a 100 % surplus to fill any possible gaps larger than 0.2 <i>mm</i> , e.g. in tight radii.	107
7.33	Wishbone link adhesive curing set-up.	107
7.34	Wishbone link prototype after curing.	108
7.35	Wishbone link test rig. The link can be mounted on the bushings at A and C. Load is applied on B. The constraints are congruent with the finite element simulation performed. To be able to change direction of load application, the whole link can be panned around Z at load application point B.	109

7.36	Test rig 0° , 45° and 90° testing orientations [HKN24]. 0° refers to a load case acting against driving direction of the car (in negative X direction). 90° refers to a lateral load case. Intermediate values can be used for combined load cases.	110
7.37	Force-displacement curves and respective gradients for three tested specimens - unpatched $3mm$ version A7 and patched versions B1 and C1 in 2.2 mm and 1.8 mm , respectively. An onset of local yielding can be observed at $F_{y1,A7} = 7\text{ kN}$, $F_{y1,B1} = 4.8\text{ kN}$ and $F_{y1,C1} = 3.3\text{ kN}$, beginning of major plastic deformation at $F_{y2,A7} = 16.5\text{ kN}$, $F_{y2,B1} = 15\text{ kN}$ and $F_{y2,C1} = 17\text{ kN}$ and maximum force at $F_{max,A7} = 30.4\text{ kN}$, $F_{max,B1} = 29.2\text{ kN}$ and $F_{max,C1} = 23.5\text{ kN}$. Variants A7 and B1 behave comparably, while C1 shows a lower stiffness, initially high resistance to major plastic deformation but ultimately fails at much lower force than the others.	112
7.38	Deformation/ failure of the specimens after static test in the region of ball joint attachment. From left to right: Overview, frontal view on the attachment region, view from the right side to the attachment region. Specimen A7 shows mayor plastic deformation and localized thinning. Specimen B1 additionally shows a failure of the material around one mounting hole, a crack in the highly-stressed right-side shoulder of the base material, and slight patch delamination in the vicinity of high base layer deformation. Specimen C1 shows similar failures as B1, with a larger area of delamination.	113
7.39	Wishbone link fatigue fractures after cyclic testing and subsequent overload. Both specimens show a similar crack origin and fatigue fracture length.	118
7.40	Force-displacement curves comparing an undamaged component with one featuring a fatigue crack of approximately 5 mm length present before static overload test.	119
A.1	Maximum Von Mises equivalent stress distribution for the three design types. Maximum location is flagged.	126
A.2	Shear stress distribution for the two patched design types. Maximum location is flagged.	127
A.3	Total deflection of the variants under design load. Linear-elastic analysis performed in ANSYS. Deformation scaled by a factor of 100 for visualization.	128
B.1	Wishbone link triaxial failure diagrams (TFD).The tight bending radii along the edges induce thickening of the regions. This leads to visible wrinkles. Some regions are stretched, but still in a safe manner due to the two-step forming process.	129
B.2	Wishbone link springback after forming OP1.3. Considerable springback can be observed in region C.	130

B.3	Wishbone link trimming step OP1.4 milling fixture for both ET1 and ET2. The parts get clamped down to their target shape, so any minor deviations are compensated. Then, the whole edge is milled down to final dimensions.	131
C.1	Final test rig design with a 3 <i>mm</i> specimen mounted. In the front-right, the attachment of the load piston to point B can be seen, while in the rear-left, the mountings to the fixture's frame at A and C are visible.	133
C.2	3 <i>mm</i> specimen with bearings at A and C and ball joint at B attached. Bearings at A and C substitute rubber bushings from the serial parts. The ball joint at B is an aftermarket component, geometrically equivalent to the respective OEM part.	134
C.3	Force-displacement curve of unpatched specimen C7 with 1.8 <i>mm</i> base thickness compared to the patched probe C1. It can be seen that initial stiffness is comparable, while C1 shows some irregularities early on. Later after onset plastic deformation, C7 stiffness decreases rapidly and it reaches a much lower maximum load before failure.	134

List of Tables

2.1	Comparison of sheet metals used in automotive structural components, based on [Mal12] and [OYK15]. While specific tensile strength differs highly, specific young's modulus is very consistent for all metals.	14
5.1	Patch optimizer input data formats. When following the suggested process of using ANSYS Mechanical for data preparation, these formats are used automatically. If other environments for data preparation may be used, the formats must match accordingly.	65
5.2	Patch optimizer node data format 1i9,3e20. As stated before, this comes standard in ANSYS Mechanical but must be maintained when using other software for manual preparation.	65
5.3	Patch optimizer element data format 19i9. As stated before, this comes standard in ANSYS Mechanical but must be maintained when using other software for manual preparation.	65
5.4	Structure of the ds.dat ANSYS APDL batch file and respective MATLAB functions	66
5.5	Structure of a single STL element block, format 1a20,3e20. This format is readable by most CAD software. If using some software that cannot process STL, the data may be manually processed based on this structure.	67
6.1	Supplier data for selected structural adhesives [DuP19] [Hen14] [3M20]. ^a derived from tensile modulus at poisson ratio $\nu = 0.3$	70
6.2	Fatigue results for 3M SAT 1100S 0.4 mm tape.	75
6.3	Fatigue results for Henkel Loctite EA 9514 liquid adhesive.	76
7.1	S500MC $t = 3$ mm mechanical properties [Fla14]	81
7.2	CARLOS Multi recorded distance, extrapolation factors and total distance [HKN24]	83
7.3	Degree of utilization for wishbone link at CARLOS-Multi load cases, according to FKM-Guideline, for 200 000 km [HKN24].	84
7.4	Wishbone link optimized patch thickness t_p results for base thickness t_b and width factor k_w combinations.	93

7.5	Wishbone link performance and relative volume results for optimal base thickness and patch thickness combinations. The restricted case achieves almost the same volume reduction Q_V as the unrestricted case. A combination of low base thickness t_b and high patch thickness t_p yield the lowest volumes. The results from which were selected for prototype building are highlighted in bold in table (a).	94
7.6	Summary of finite element simulation results for the three specimen types A (3 mm base without patch), B (2.2 mm base + 1.2 mm patch) and C (1.8 mm base + 2.2 mm patch). Von Mises equivalent stress for selected location, adhesive shear stress and total load application point deflection. .	99

Nomenclature

x	Placeholder variable, arbitrary point, element or set
A_x	Area of set/region x
b	Doubler joint width
B_K	Element's energy deviation from mutual energy density
β	Half-doubler joint stiffness ratio
C	Closed list for path algorithm
d	Displacement
D	Displacement vector
Δ	Lap joint stiffness ratio
δ	Expense for path algorithm
e	Single element (general)
e_x	Single element in set x
E	Set of Elements (general)
E_x	Set of elements of set x
E_Ω	Element representation of the domain
ε	Strain
F	Force
FLD_0	FLD knee-point
F	Load vector
F_x	Set of loads on element / set x
g	Performance
g_0	Baseline performance
G_a	Adhesive shear modulus
h	height of component
K	Stiffness matrix / stiffness tensor
k_w	General width factor
K_x	Stiffness of element / set x
l	Load location, start point
L	Set of load locations
l_o	Lap joint overlap

LS	List of load steps
Λ	Mutual energy density
μ	Arithmetic mean
n_x	Single node in set x
N	Set of nodes (general)
N_x	Set of nodes of element / set x
∇	Gradient
∇_w	Gradient width modifier
N_Ω	Nodal representation of the domain
ν	Velocity modifier for path algorithm
O	Open list for path algorithm
Ω	Design space / domain
p	Penalty factor
P	Remote point
πO	Pointer for path algorithm
p_l	Load path
P_l	Set of load paths
Q_g	performance ratio
Q_V	Overall volume ratio from reduction
$Q_{V,min}$	Lowest volume ratio for highest reduction
q_x	Quantity of x
R	Set of restricted regions
$\mathcal{R}^2$	Coefficient of determination
$\mathbb{R}^2$	2D-Space
$\mathbb{R}^3$	3D-Space
R_e	Elastic limit
R_m	Tensile strength
R_{spec}	Density-specific tensile strength
R_τ	Shear stress ratio
ρ	Density
ρ_{min}	Minimum density
ϱ_w	Path density modifier
s	Support location, target point
S_e	Endurance limit
σ	Standard deviation
σ_∞	Doubler joint stress in the base plate outside the joint region
t_0	Initial base thickness
t_a	Adhesive thickness
t_b	Base thickness

t_{bp}	Patched region total thickness
t_p	Lap joint patch sheet thickness
t_{p1}	Minimal patch thickness
t_s	Lap joint sheet thickness
$\tau(x)$	Location dependent shear stress in the adhesive
τ_{amp}	Shear stress amplitude
τ_m	Average shear stress in the adhesive
$\tau_{max,50}$	Maximum shear stress amplitude for 50% survival probability
τ^{max}	Maximum shear stress in the adhesive
$\tau_{l \rightarrow \infty}^{max}$	Maximum shear stress in the adhesive for infinite length
U^*	U^* -Index
w	Width of component
x	Displacement
Y	Elastic modulus
Y_e	Element stiffness
Y_e^0	Initial element stiffness
Y_p	Lap joint / doubler joint patch elastic modulus
Y_s	Lap joint / doubler joint sheet elastic modulus
Y_{spec}	Density-specific elastic modulus
ζ	Maximum step size

Chapter 1

Introduction

One of the main interests in automotive development is reducing vehicle weight as lighter vehicles tend to require less energy for driving. This results in lower carbon emissions, either directly for internal combustion engine cars or from the production of energy required to charge battery electric vehicles - if the energy is sourced from non-renewable, carbon dioxide emitting methods. To highlight the necessity to decrease passenger car's energy consumption: The transport sector accounts for approximately 23 % of worldwide carbon emissions in 2022 with 48 % of it being passenger vehicles. [JCM⁺24],[IEA23a],[IEA23b]. Energy consumption is based on the required pulling force for moving the car. This force needs to overcome driving resistance, of which rolling, acceleration and gradient resistance are all mass-dependent - approximately linearly proportional to vehicle gross weight [Bra19]. Up to velocities of around 60 km/h, depending on the vehicle specifications, these make up the majority of driving resistance until the wind resistance, quadratically proportional to velocity, exceeds them [Lin18]. Thus, any weight reduction directly leads to less pulling force required and less energy consumed. Additionally, a lighter car requires only a smaller engine, fuel storage, or other aggregates which leverages any individual components weight reduction, creating a "weight spiral" that describes the amplifying effect [GGS17].

The principle of mass reduction in automotive, as well as in other fields, is called *lightweight design* or *lightweighting*. The potential for mass reduction of some function, assembly or component can be called *lightweight potential*. Different materials, manufacturing methods or design concepts exhibit different development and manufacturing cost at different lightweight potential. They must be evaluated with regard to their cost-effectiveness, especially for high-volume production [Kur11]. For automotive body and chassis, which of the car's total mass make up around 40 % and 25 %, respectively [VDA11], one concept very suitable for mass-production is the use of sheet metal. Sheet metal manufacturing is cheap for high production volumes due to efficient production processes such as cold forming and welding. Sheet metal components exhibit favorable mechanical properties at low mass: High-strength and high elastic modulus materials can

be used for creating light but stiff *shell-structures* such as unibodies. Most structural parts in today’s cars are made from sheet metal [MFS⁺24].

Shell structures are thin-walled, curved, hollow geometries which have material mainly on the outermost boundary of their space. They can achieve high strength and stiffness at relatively low mass due to high section modulus for overall low cross sectional area [BS12]. Sheet metal shell parts generally exhibit uniform material thickness. Minimal thickness is determined by local stiffness or strength requirements. In practice, stress distribution is non-uniform and regions contribute differently to the overall stiffness. Consequently, the region with the highest minimum thickness requirement determines the material thickness for the entire component. In theory, some regions could have reduced thickness which would lead to mass reduction without compromising overall stiffness or strength. With conventional sheet metal manufacturing processes for automotive components however, planned non-uniform thickness cannot be achieved.

To overcome this issue while keeping sheet metal’s advantages, some advanced manufacturing methods have been developed. Tailor welded blanks, tailor rolled blanks and patchwork blanks aim to create a sheet metal part with non-uniform thickness or non-uniform material properties to react locally to local demands [MJLK14]. Designing for such manufacturing methods requires additional effort as the optimal distribution of thickness throughout a structure is not obvious. Manufacturing constraints need to be incorporated within the design process. Complexity considerations such as count of regions with differing thickness may influence design.

This thesis presents a new method to design and implement the concept of varying material thickness for mass reduction of sheet metal parts in an intuitive way while maintaining the ability to restrict design with regard to manufacturability and complexity. While not restricted to it, this is thought of as strengthening a base layer with a local patch. The method is implemented as a software and tested on a real-world automotive component to evaluate its usability and effectiveness. Manufacturing methods for patched components designed with this method are developed and prototypes built. Finally, those lightweight-prototypes are tested under realistic load cases to give an indication of mass-optimization success.

1.1 Structure of this Thesis

At first, in chapter 2, sheet metal design and manufacturing methods are introduced as they are highly used for lightweight mass-produced structural automotive components. To better understand why this is the case and how manufacturing and design possibilities are intertwined, an overview on relevant manufacturing technologies involved, as well as design constraints, is given. The current state of research and industrial application of new lightweight materials, designs and respective manufacturing methods is shown. As

this thesis focus lies on lightweight design methodology, a more broad view on other lightweight design and optimization methods is included, as well.

State-of-the-art's limitations and the proposed method's requirements are discussed in chapter 3. Challenges and shortcomings of current approaches are examined and evaluated. Additionally, the requirements for the proposed method to address these shortcomings are outlined.

Chapter 4 gives a detailed introduction to the proposed method. Its core concept is thoroughly described, along with an explanation of how and why it differs from current approaches. The chapter then shows the function structure and the principles of each individual functional element. Their underlying mathematical models are explained in detail and relevant parameters deduced.

The method is being implemented in a stand-alone and fully functioning optimization software, which is described in the following chapter 5. Starting with the overall environments used, subsequently, the software architecture, function structure and relevant interfaces are shown, being in- and output as well as internal interfaces between different environments.

As this thesis should not only give the theoretical method but also an idea of implementation in industrial application, chapter 6 investigates upon manufacturing methods to achieve non-uniform thickness sheet metal components by adhesive patch bonding. This includes adhesive selection and testing its applicability for structural components, including fatigue strength.

Finally, in chapter 7, the method, its software and the manufacturing process are validated on a real-world component with demanding geometry and use case. The optimization results, prototype built process, test setup and testing are presented.

The thesis concludes with summary and outlook in chapter 8.

Chapter 2

State of the Art

2.1 Sheet Metal Application and Manufacturing

This thesis focuses on lightweight design for sheet metal. On average, more than half of a car's mass is made from steel [wor16] and more than 75 % of structural parts in cars are made from sheet metal [HMLZ13]. Why is this manufacturing concept so dominant? To give a better understanding, reasons for use of sheet metal in automotive and the main manufacturing methods including their advantages and limitations are explained in the following.

2.1.1 Sheet Metal in Automotive

When cars were first introduced, their load carrying structures were built similar to horse carriages, mainly out of wood, later then from steel tubing. The chassis and body were split, with the body often manufactured individually and put onto a rolling chassis [LR18]. This concept, called body-on-frame, still exists, albeit mainly for niche vehicles, off-road cars or trucks where there is a need to put different bodies onto the same frame. These designs are generally disadvantageous with respect to lightweight design, as the frame must be built very rigid to carry all load, be crash resistant and stiff on a limited cross-section [Fan23]. With more modern manufacturing possibilities and higher production volumes, from the 1940's, the new concept of unibodies was deployed. In unibodies, body and frame are integrated into one. This way, much larger cross sections carry torsional and bending loads, highly increasing the car's stiffness and yielding more efficient material usage. Most modern cars are built this way [Fan23]. The car's outermost shell carries most load. For bending or torsional loads, this is the most efficient use of material [Kle09]. Thin and stiffly joined shells are favorable - properties, which can be achieved well by sheet metal [LK05]. It can be manufactured efficiently into complex parts with spatial shapes.

2.1.2 Sheet Metal Forming

As in DIN 8580, manufacturing methods can be grouped into *casting*, *forming*, *separating*, *joining*, *coating* and *change of material properties* [FF23]. Within group 2, forming, there are the sub-groups of *pressure forming*, *tension-compression forming*, *tension forming*, *bend forming* and *shear forming* which are classified by the state of stress within the formed material. To create sheet metal, a cast slab is pressure formed (specifically, *rolled*) to a sheet with desired thickness. The sheet can then be further formed - in automotive, typically by tension-compression forming process *deep drawing*. Deep drawing is suitable for mass production due to fast production, leading to low individual part cost, good manufacturing quality and high possible material strengths [FS10].

In deep drawing, a plane metal sheet is plastically deformed to a desired shape without any desired change in sheet thickness. This is due to the aforementioned tension-compression stress state. Deformation is achieved by pressing the sheet into a die with a punch while clamping it on the edges by a blank holder. A schematic of this can be seen in figure 2.1.

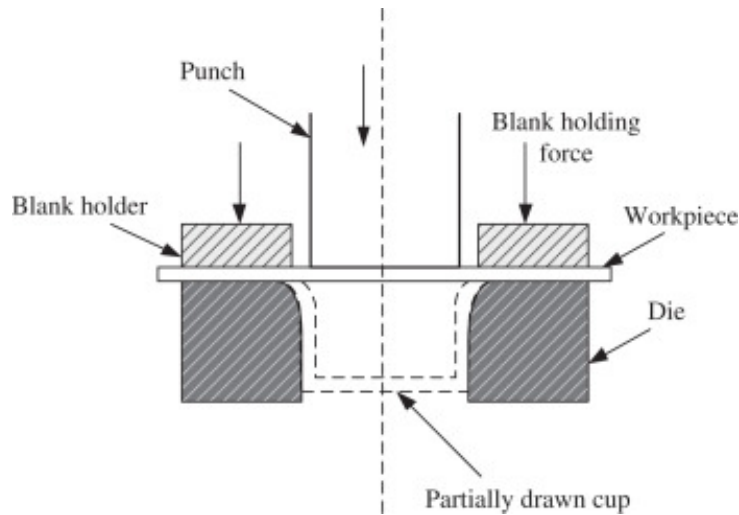


Figure 2.1: Schematic of a forming tool [Dix20]. A workpiece is held between a blank holder and a die, and then plastically deformed by a punch.

Deep drawing gives some geometric restrictions to a component design. There are limits to the maximum deformation a sheet can undergo before failure. This property, specific to sheet material and thickness, can be displayed as the *forming limit curve (FLC)*, see figure 2.2, valid for plane stress state. As the FLC shows, a negative minor strain ε_2 enables a high mayor strain ε_1 - correlating to a plane tension-compression stress state. The sheet deformation ought to stay within the limits of deformation throughout the whole forming process for each location, otherwise material failure such as cracking or wrinkling is to be assumed. This can lead to complex multi-step forming processes required to get to a highly deformed geometry which would otherwise be impossible to achieve in one direct forming operation.

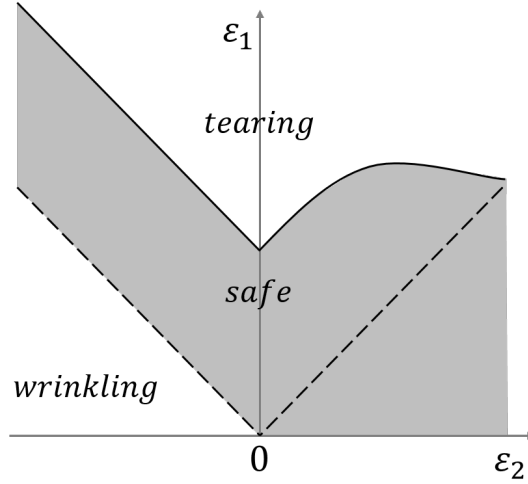


Figure 2.2: Forming limit curve for plane stress state. Own illustration based on [Cha15]. The plane strain deformation of a sheet ought to stay within the grey area during forming. Overly high compression may lead to wrinkling, while excessive stretching leads to tearing.

For deep drawn structural components in automotive, mainly metals and sometimes plastics or reinforced plastics find application. Materials for deep drawing must be highly plastically deformable. An indication to this is the materials elongation at break as this gives the amount of plastic deformation a material can sustain before breaking [SSP⁺17]. For structural application, materials should have high tensile strength and elastic modulus as these influence the final components strength and rigidity. All this is well-accomplished by steels which additionally show the effect of work hardening, which is an increase in strength due to forming [VSM17]. Most car body components are made from steel [HBP⁺17]. Generally, higher strength steel have a tendency to show lower elongation at break and require higher forming force. Some material developments are aimed to increase those limits: New steel formulations lead to increased formability of high strength steels [ST15], press hardening increases material strength after forming at high temperature and subsequent quenching [NAG⁺06], and bake hardening leads to an increase in material strength by aging at moderate temperatures [YYZQ23]. Still, the correlation between high material strength and low formability remains, see figure 2.3.

For a given geometry there are some limitations to which material can be used and what part strength will be achieved accordingly. As formability increases with increasing thickness, as can be seen from Keeler’s approximation of the FLC’s knee-point FLD_0 [Kee65] being proportional to sheet thickness, one could argue that to increase formability, thickness can be increased. This, however, leads to high forming forces and part weight, after all, so is not reasonable. More common method to increase formability is to form above recrystallization temperature, see hot working [Ils73] which can be done even only locally on highly deformed regions [JSS22].

To summarize, components made for deep drawing must be designed with the (multi-step) forming process in mind. They generally exhibit uniform thickness throughout.

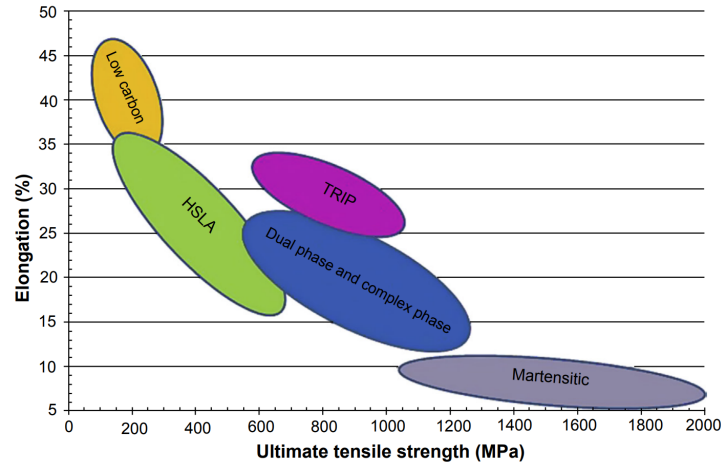


Figure 2.3: Inversely proportional relationship of ultimate tensile strength and elongation at break for first generation advanced high strength steels [ST15].

For a defined shape, component strength can be achieved by sheet thickness, which increases part weight, or higher strength materials, which however increase forming process complexity.

2.1.3 Joining of Sheet Metal

Chapter 2.1.2 explained the sheet metal forming process and its limitations. Unibodies, as well as other separate sub-components such as closures or chassis parts, are normally made from several parts which are permanently joined to create complex and/or closed-shell geometries. In the following, permanent joining methods which can be applied to deep drawn sheet metal components are described. From DIN 8580, relevant joining methods are *joining by forming*, *welding*, *soldering* or *adhesive bonding* [FF23]. Within joining by forming, there is beading, clinching or riveting applicable. Within welding, there is pressure- and fusion welding among which spot welding, arc welding and laser welding find application. Within soldering, usually hard soldering is used for structural sheet metal parts. More recently, adhesive bonding finds increasing application for sheet metal components, even in structural applications [Jü18].

These joining methods can also be classified by joint shape. Spot welding, clinching and riveting create point-shaped joints. Arc welding, laser welding and beading create linear joints. Soldering and gluing create areal joints, where two sheets are lying flat on each other. This is summarized in figure 2.4.

For joining deep drawn parts all of these shapes can be useful. However, for stiffening deep drawn parts by a second layer while maintaining geometry – which is the focus of this thesis – areal shapes appear to be more suitable as they can use the whole contact surface. Those methods, being soldering and adhesive bonding, will thus be described in greater detail in the following with the others only touched briefly.

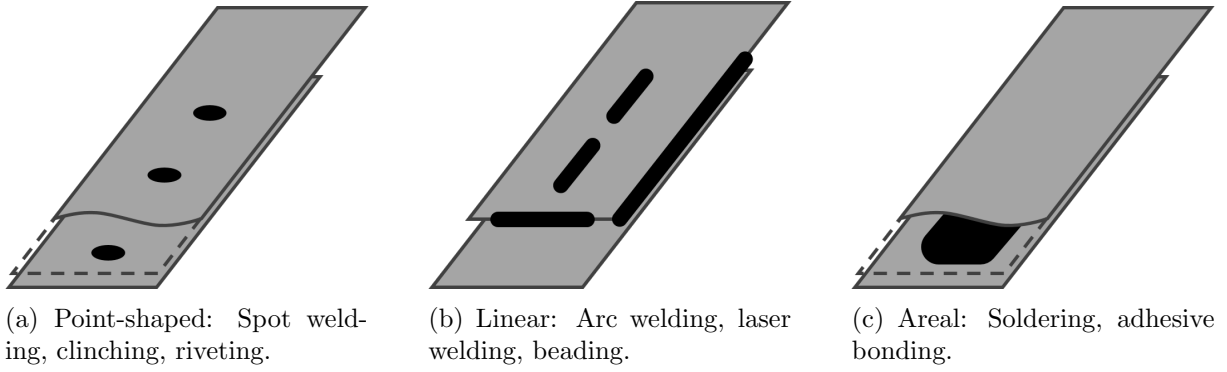


Figure 2.4: Typical joint shapes of selected permanent joining methods used for sheet metal components.

Point-shaped Joints by Spot Welding, Clinching or Riveting

Today, the most commonly used welding technique is spot welding, making up a share of upwards of 80 % among all joints in high-volume cars [RK18]. In mass production of automotive body-in-white, most parts are spot welded fully automated at the assembly plant and subsequently painted as one unit. Two layers of sheet metals are clamped and high electric current going through the clamp melts them together in the clamped spot. Main advantage of spot welding is its fast and efficient use while showing good strength and ductility [SPA21]. Disadvantages are the required accessibility of the joint, generally from both sides, and material compatibility - mostly, only similar materials with respect to electric conductivity and melting temperature can be spot welded [ibid.]. Although, some advancements in joining dissimilar materials have been made such as ultrasonic [ADH⁺23] or laser assisted spot welding [LTZ⁺22]. Also, in components exposed to moisture, crevice corrosion can be of concern and localized stresses may occur at the joints.

In riveting, a third component, the rivet, is pressed into the overlapping sheets and clamps them. Clinching is similar, but no third component is required. Advantages are low cost and ability to join dissimilar materials [WKTW21].

Linear Joints by Arc Welding, Laser Welding or Beading.

In arc welding, the sheets are locally molten by an electric arc and a filler material is fed to the melt, bonding the sheets. Unlike spot welding, arc welding creates linear joints and can thus provide a more spread out connection and sealing functionality. It finds use in highly stressed structural components such as seat frames requiring high strength [NMM12] and chassis components exposed to the elements [PJKM22]. *Metal active gas welding* (MAG welding) induces high temperature into the workpieces, potentially leading to heat distortion and change in material properties. Developments to reduce this are processes aimed at reducing the heat input such as cold metal transfer welding where the arc is interrupted by an oscillating electrode [TVLS12]. However, here, sensibility to

cold-cracks may be higher as the workpiece remains too cold, cooling the weld down too quickly, requiring additional preheating [RHN23].

More recently, laser welding replaces arc welding due to its decreased heat induction, high process speeds and weight reduction potential as often no filler material is required. It finds application in car body, chassis and exhaust. Disadvantages are gap-bridging ability, requiring tight mating component tolerances, see [LMD⁺15], [PJKM22] or [Rob16].

Beading is the process of rolling up the edges of sheets to create a tight bond and rounded edge. It is mainly used in body panels and not for high load-bearing applications [MMA⁺96].

Areal Joints by Soldering or Adhesive Bonding.

Soldering is categorized into *soft soldering*, *hard soldering* (also known as *brazing*) and *high temperature brazing*, based on the process temperature. Brazing ranges above 450 °C and high-temperature brazing above 900 °C. Soldering is mainly used for metals as defined in DIN 857-2 [Gmb19] as a permanent metallurgical bond between two components, typically metals, by a filler material that is distinct to the mating materials. The filler material has a lower melting point than the base materials which remain solid during the process. The liquid filler coats the mating surfaces, diffuses into them and solidifies after cooling [FF23]. Compared to welding, the lower process temperature creates less thermal distortion or micro-structural changes within the bonded components [DBR24]. Dissimilar materials can be soldered, as long both their melting points are above the process temperature. For example, it is possible to braze aluminum to steel [Nn14]. Areal joints can be created either by capillary force pulling the filler into a tight gap between the mating surfaces during the process or by placing the filler between the surfaces in advance. Soldering is sensible to surface contamination and mating surfaces must be aligned well to achieve a uniform bond. Brazed lap joints can achieve high strength, with high temperature brazing coming close to the base material's shear strength [GW12], [ZYH⁺14], [ZWX⁺24].

Finally, there is **adhesive bonding**, in the following also called *gluing*, which finds huge application in today's automotive. Very frequently used in trim and window framing, also structural and crash-relevant components from sheet metal are glued. For those, polyol-based [BRCdS18] or epoxy-based [KYH19] adhesives are usually used. For metal bonding, structural adhesives, which are classified as having shear strengths of $> 6.9 \text{ MPa}$, are mainly thermosetting with strong covalent bonding for cohesion within and Van-der-Waals bonding for adhesion to the metal partners [Pet21] [HKN21]. Most of these use a hardener, usually amines, to kick off the crosslinking reaction. It is either a second component added to the resin during the application process (*2K-Adhesives*) or microencapsulated within (*1K-Adhesives*) [ZWL24]. The encapsulation is forced to break most commonly using temperature, starting the curing process. The adhesive typically cures

at temperatures of 180 °C for around 30 minutes. There are 1K-Adhesives where the microcapsules breaks from pressure instead, used as threadlocker, see [Ind24].

Sheets should be glued over big area and loaded in shear [Jü18]. That way, they can transfer force well as it does not induce stress concentration in the joined sheets and only moderately in the adhesive. Figures 2.5 and 2.6 show the stresses in adhesive and sheets under lap shear configuration. Peeling stresses due to overlap eccentricity is disregarded. As seen, the tensile stress in the sheets increases uniformly, while there is moderate shear stress concentration on the adhesive on the edges which needs consideration in design. Average shear stress in the adhesive τ_m can be calculated based on load F and lap joint area $A_{overlap}$ by

$$\tau_m = F/A_{overlap}. \quad (2.1)$$

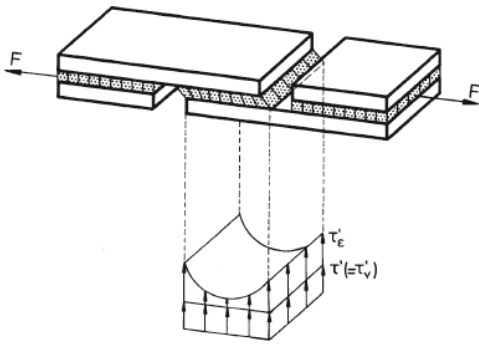


Figure 2.5: Shear stress in the adhesive on lap shear test [Hab09]. The stress reaches a maximum on the bond edge in load direction.

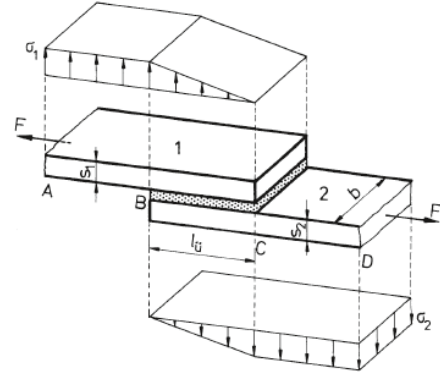


Figure 2.6: Tensile stress in the sheets on lap shear test [Hab09]. Stress decreases gradually in the bonded area without any peaks.

Stress concentration occurs on the edges due to elastic deformation of the joined mates. This concentration is dependent on adhesive properties thickness t_a , shear modulus G_a , overlap l_o as well as the mating sheet's elastic modulus Y_s ¹ and thickness t_s . It can be calculated according to [Vol38] assuming isotropic homogeneous adhesive, linear elasticity and no bending due to eccentric loading (inducing peel stress) by

$$\tau(x) = \tau_m * \sqrt{\frac{\Delta}{2}} * \frac{\cosh \sqrt{2\Delta}x/l_o + \cosh \sqrt{2\Delta}(1 - x/l_o)}{\sinh \sqrt{2\Delta}} \quad (2.2)$$

with stiffness ratio

$$\Delta = \frac{G_a l_o}{t_a} * \frac{l_o}{Y_s t_s}. \quad (2.3)$$

If the setup is not a lap joint but a patch as in figure 2.7, the calculation slightly changes. With the same assumptions as before it can be calculated similar to bonded

¹Y is used instead of the common notation E for elastic modulus to avoid confusion with the notation for *elements* later on.

doubler joints, which are two-sided reinforcement plates in aerospace, as suggested in [WD16] by assuming force split in terms of joint geometry.

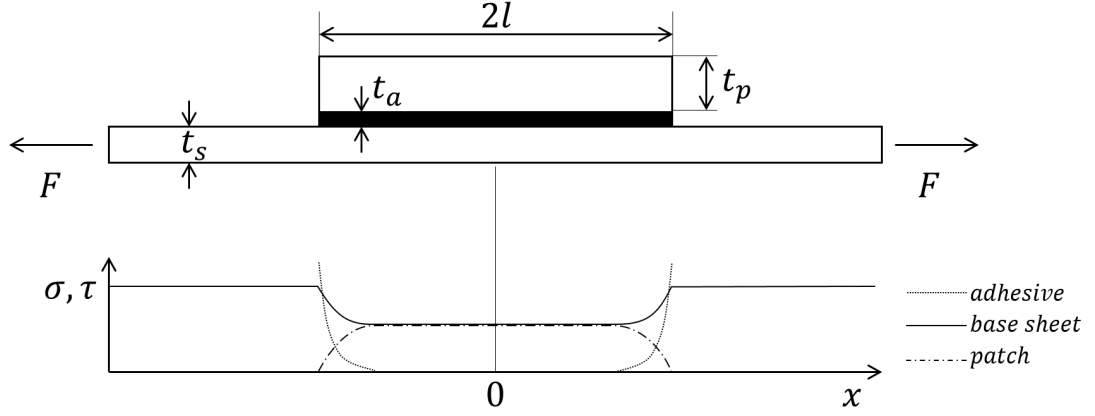


Figure 2.7: Schematic of a base sheet with glued patch under tensile load with tensile stresses in the sheets and shear stress in the adhesive shown. Load gets transferred to the patch via the adhesive, until equally shared between base and patch based on their stiffness ratio. After a transition zone on the edges, where load is not distributed yet, this package then behaves as monolithic.

The calculation uses sheet and patch thicknesses t_s and t_p , their respective elastic moduli Y_s and Y_p , sheet width b as well as adhesive thickness t_a and shear modulus G_a :

$$\beta = \sqrt{\frac{G_a}{t_a} \left(\frac{1}{Y_s t_s} + \frac{1}{Y_p t_p} \right)} [mm^{-1}] \quad (2.4)$$

describes the geometry, leading to local stress in distance x from the patch center being

$$\tau(x) = -\frac{G_a \sinh(\beta x)}{\beta t_a \cosh(\beta l)} \left(\frac{\sigma_\infty}{Y_s} \right) \quad (2.5)$$

with σ_∞ being the cross sectional tensile stress in the base plate outside the joint region, which on constant width strips under uniform tension would be $F/t_s b$. Maximum shear stress occurs at the edges $x = \pm l$. Plugging this into equation (2.5) leads to

$$\tau^{max} = \frac{G_a}{\beta t_a} \tanh \beta l \left(\frac{\sigma_\infty}{Y_s} \right) \quad (2.6)$$

which now depends on the joint geometry as well as patch length. With l being strictly positive, τ increases with growing l up to

$$\tau_{l \rightarrow \infty}^{max} = \frac{G_a}{\beta t_a} \left(\frac{\sigma_\infty}{Y_s} \right). \quad (2.7)$$

This holds true also in the case of known local stresses without patch if the stress gradient is low, thus can be used for approximate patch design even for non-uniform

stress within the base layer. After a short transition zone of high shear with length on the order of $1/\beta l$ in the adhesive the sheet-patch package behaves as monolithic [WD16].

2.2 Lightweight Design for Sheet Metal

Lightweight design aims at achieving requirements to a part or assembly with minimal weight. Lightweighting strategies are often differentiated as in [EGGH13]: shape-wise, material-wise and conditional lightweighting. Conditional lightweighting aims more on decreasing a components requirements, for example by reducing safety factors required, and not on component design itself and is thus beyond the scope of this thesis. For more information, see for example [WK88]. For the following, material and shape lightweighting for sheet metal is further classified in the following way:

- material lightweighting in the commonly known fashion of picking a material with good density-specific mechanical properties,
- regional material-wise lightweighting aiming at changing sheet properties or thickness in some areas before forming,
- regional shape-wise lightweighting aiming at locally altering properties for already formed sheet metal components.

2.2.1 Material Lightweighting

Material lightweighting is the use of materials which exhibit good performance at low specific mass. For sheet metal parts at a given design, the strength and stiffness can be estimated to be linearly proportional to the parts sheet thickness. This is valid if assuming no considerable effect of sheet bending stiffness and structures with cross sectional dimensions $d \gg t$, where the effect of thickness on area and area moment of inertia is proportional as well. Then, a materials lightweighting performance can be written as the specific stiffness or specific strength, since mass is proportional to thickness and density, while stresses and deformation are inversely proportional to thickness, canceling out. $Y_{spec} = Y/\rho$ and $R_{spec} = R_m/\rho$ with Y as young's modulus, R_m as tensile strength and ρ as material density describe the material-specific performance. The tensile strength can be interchanged with elastic limit R_e or endurance limit S_e given the use case. Big efforts have been made to increase these performances for sheet metal. In automotive body trends to aluminum [Hir11] and high strength steels [KKW08] are visible. Table 2.1 shows exemplary performances for typically used materials in automotive. Specific stiffness is comparable for metals due to the elastic modulus being strongly related to interatomic bonding which is similar for metals [Ash10].

Specific strength differs highly even throughout same material families such as steels. Material strength is closely related to micro-structure, defects and alloy elements. Press hardened steels (PHS) and titanium alloys show greatest performance among metals. Carbon fiber reinforced plastics (CFRP), which are not regarded here in detail as their manufacturing processes differ highly from sheet metal, show even better results, even in quasi-isotropic state [Mal12]. This explains CFRP’s extensive use in high end lightweight applications such as luxury sports cars [AMPA20]. In higher volume high end sheet metal applications, *press-hardening steels* (PHS) are seen most commonly, for example on important crash components such as B-pillars [KT10].

Table 2.1: Comparison of sheet metals used in automotive structural components, based on [Mal12] and [OYK15]. While specific tensile strength differs highly, specific young’s modulus is very consistent for all metals.

Material	Density kg/m^3	Young’s Modulus GPa	Tensile Strength MPa	Spec. Young’s Modulus $10^6 m^2 s^{-2}$	Spec. Tensile Strength $10^3 m^2 s^{-2}$
DQ low carbon steel	7870	207	317	26	40
DP700 high strength steel	7870	207	700	26	89
TRIP450/800 steel	7870	207	800	26	102
22MnB5 PHS steel (hardened)	7870	207	1500	26	191
304 stainless steel	7860	200	862	25	110
5182-H24 aluminum	2700	70	310	26	115
6111-T62 aluminum	2700	70	360	26	133
AZ91 magnesium	1800	45	240	25	133
Ti-6Al-4V Titanium	4430	114	896	26	202

Manufacturability and manufacturing cost need to be kept in mind, which limits the possible selection. As seen before, formability decreases for higher strength among steels requiring costly forming processes. Joining and forming of light metals is complicated and those materials generally come with higher cost. Finally, stiffness can hardly be maintained while reducing mass without part design changes except for the use of unidirectional CFRP. Thus, for sheet metal other methods are to be considered as well.

2.2.2 Regional Material and Shape-Wise Lightweighting

Looking at a component and its stress or strain energy distribution under use, it is rarely uniform throughout the geometry. As sheet thickness and material are driven by local requirements some other areas may be loaded far below the materials capacity. Some

methods for sheet metal aim at reducing this imbalance by introducing non-uniform material properties. These properties are sheet thickness, strength or stiffness - while thickness itself is not a material property per se but more a sheet property, as seen before in geometric conditions commonly seen in sheet metal components it can have a predictable effect on stresses or strains and thus be handled as such.

Tailor welded blanks (TWB) are two different sheets in terms of thickness or material either continuously welded side-by-side from coil or welded after blanking but still in flat state. Blanks from this material can then be processed just as normal but show different regional characteristics. The advantages of this process over separate component manufacturing and subsequent welding are weld quality, geometric accuracy and manufacturing efficiency. [Kin11].

Tailor rolled blanks (TRB) follow a similar concept but only varying in sheet thickness. This variation is achieved by a change in gap when rolling a sheet. While continuously welded TWB show the difference lateral to rolling direction, TRB differ longitudinal. Also, the change in thickness is not abrupt but continuous with slope up to 0.01 mm/mm [KS16] but can be below, giving some more degree of freedom in design and enabling a smoother stress gradient between sections. Downsides are hard to define forming behavior in the sloped region and the one-dimensionality of thickness-change making this just like TWB more suitable for large lengthy components, although two-dimensional methods are being developed [HHO⁺18].

Patchwork blanks (PB) give another degree of freedom by adding a local second layer onto the blank by either spot welding, laser welding or gluing. In contrast to adding a second component later on, both sheets can be formed in one tool. The joint needs to withstand the forming process which limits deformation. Patchwork blanks are suitable for similar components as the other tailored blank methods but give more lightweighting potential. Use cases can be found in automotive B-pillars [SWN⁺22] or shock towers [LMG05].

Inherently, tailored blanks can only give an approximation to optimal material distribution among a part surface due to their manufacturing restrictions. Methods that alter a sheet metal part after forming can come closer to optimality. One possibility is changing the sheet itself in the region, for example by shot peening [YRJ22] or coating [GHR⁺22] for fatigue strength increase or localized hardening for breaking strength increase [ITTE12]. Alternatively, material may be added to some regions assisting the base sheet metal part. This material may be different from the sheet metal offering unique hybrid properties. Recent developments show additive laser metal deposition [UHS⁺19], adhesively bonded CFRP patches in areas of high stress uniaxiality [Tri23], riveted CFRP inlays for crashworthiness [sLySgK17] or thermoplastic inlays for fatigue loaded components [Klo20]. Generally, these methods exhibit high mass reduction such as 57 % in [Tri23], 44 % in [sLySgK17] or 20 % in [Klo20] but come with high manufacturing cost.

2.3 Sheet Thickness Optimization

The previous chapter showed a variety of manufacturing methods to achieve lightweight design while keeping sheet metal manufacturing as core process. High potential can be seen on methods where sheet metal thickness is non-uniform, either by modifying blanks or later patching. For both these methods, the question arises how components can be designed to this and how for a given method optimality can be reached - as is, where the ideal material distribution lays.

Now, one could consider two vastly different approaches to this:

- Firstly, by analyzing a given structure or a preliminary design concept in terms of local performance, e.g. local stresses or deformations and reacting to this by changing properties just there, or
- secondly, by analyzing a geometry as a whole and altering thicknesses to make the geometry follow design principles such as higher area moment of inertia or uniform flow of forces.

These two concepts will be referred in the following as *local* and *global* concept, which should not imply any approaches ability to find global optima or risk to get stuck in local ones.

2.3.1 Geometry Optimization with Local Concept

Within the general concept of structural optimization, three approaches to geometry optimization can be distinguished (see figure 2.8), following the classification of [BS04]:

- topology optimization where an optimal geometry is created within a general design space without considering manufacturing process. Subsequent discretization with the lightweighting method of choice is required
- shape optimization where a given geometry is refined around its border while general shape will be kept
- sizing optimization where a manufacturable geometry is optimized within the bounds of the given lightweighting method by strengthening or weakening some discrete areas or elements.

One should not handle these methods completely separate, since, for example, some topology optimization methods use the same concept as sizing optimization. Also, they may be intertwined, with shape optimization being a suitable subsequent process after initial topology optimization. They all share the inherently local aspect though, meaning that in the discrete case in computation, each element reacts to its own state. There is no

further information about the behavior of the structure as a whole required, making these methods very computationally efficient and easy to use. To showcase this, the most widely used method of topology optimization is described in the following to elaborate more on the core mechanic. The following explanations all base upon the ground-laying work of Bendsøe and Sigmund as summarized in [BS04] where also all following formulas are cited of. Here, only the basic idea is described, for more computational details, please see the reference.

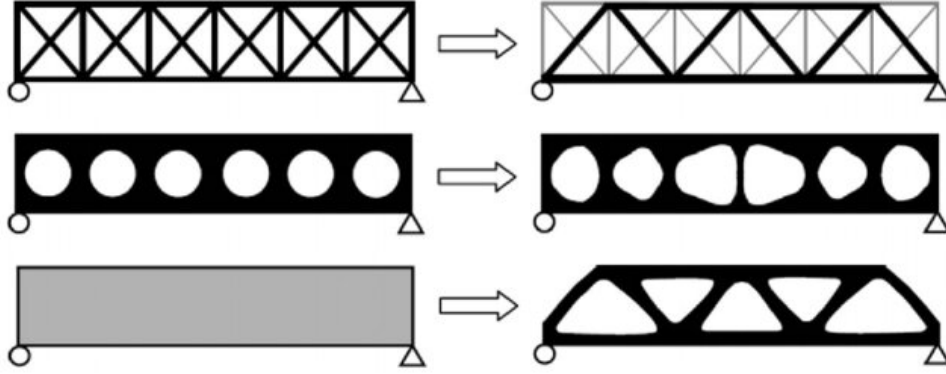


Figure 2.8: Three methods of structural optimization, from top to bottom: Sizing, Shape, Topology. left image shows initial, right the optimized state [BS04].

Starting point is a design space Ω which is exhibited to loads and constraints on their respective locations. The design space is then partially filled with material so that a desired overall material volume is reached, for example filling up 50 % of the design space. This partial filling can be thought of as relative density of the material within Ω in $\mathbb{R}^3$ or $\mathbb{R}^2$ while it can be thought of as shell thickness for shell topology optimization as explained later in this section.

For minimum compliance, the problem then becomes that of minimal internal energy, written in discrete form:

$$\begin{aligned} & \min_{\mathbf{D}, Y_e} \mathbf{F}^T \mathbf{D} \\ \text{s.t. : } & \mathbf{K}(Y_e) \mathbf{D} = \mathbf{F} \\ & Y_e \in Y_{\text{ad}} \end{aligned} \tag{2.8}$$

with D and F as displacement and load vectors. Stiffness matrix K is based upon element stiffnesses Y_e . From an initial stiffness Y_e^0 specific to each element, this stiffness is then adjusted to incorporate the element's density. This could be done by a threshold, setting above-average density element's stiffness to $Y_e(x) = 1 * Y_e^0$ and $Y_e(x) = 0$, otherwise. However, it is required to allow intermediate densities and having them converge to 1 or 0 by penalizing intermediate densities instead of the hard threshold, as otherwise, elements cannot "switch" their state easy enough. This can be achieved using the

SIMP-Model (solid isotropic material with penalization), so element stiffness becomes

$$Y_e(x) = \rho(x)^p Y_e^0, \quad 0 \leq \rho(x) \leq 1, \quad p > 1 \quad (2.9)$$

introducing penalty factor p to be chosen according to desired strength of forcing convergence, typically set to $p \geq 3$.

The result is iteratively determined by increasing density of elements having above-average strain energy and decreasing otherwise using optimality criteria method:

$$\rho_{K+1} = \begin{cases} \max \{(1 - \zeta)\rho_K, \rho_{\min}\} & \text{if } \rho_K B_K^\eta \leq \max \{(1 - \zeta)\rho_K, \rho_{\min}\} \\ \min \{(1 + \zeta)\rho_K, 1\} & \text{if } \min \{(1 + \zeta)\rho_K, 1\} \leq \rho_K B_K^\eta \\ \rho_K B_K^\eta & \text{otherwise} \end{cases} \quad (2.10)$$

with ζ being a maximum step size, $\rho_{\min}$ a mathematically necessary lower limit greater than zero, typically chosen to be $\rho_{\min} = 10^{-3}$, and B_K describing the element's energy deviation from mutual energy density Λ :

$$B_K = \Lambda_K^{-1} p \rho(x)^{p-1} Y_e^0 \varepsilon_{ij}(D_K) \varepsilon_{kl}(D_K), \quad (2.11)$$

$$\Lambda = p \rho(x)^{p-1} Y_e^0 \varepsilon_{ij}(D) \varepsilon_{kl}(D). \quad (2.12)$$

Figures 2.9a and 2.9b show exemplary topology optimization results with different penalty factors.



Figure 2.9: Topology optimization performed for a simple structure with one load and two supports. Penalization shows a convergence to a black-white result while the unpenalized case allows intermediate (grey) densities. Computed with DTU's TopOpt software [ANJAS12].

Some further efforts need to be considered to increase quality of the output such as reducing mesh dependency and so-called checkerboarding meaning undesired creation of a microstructure. Also it is possible to restrict areas to being either full material or void

if required design-wise.

If the problem is not that of stiffness but minimal internal stresses the formalization needs modification since unlike element strain energy, stress does not converge to zero with element stiffness going to zero. There are numerous ways proposed to solve this by, for example, constraint relaxation [CG97], damage minimization [VLvK15] or subsequent shape optimization [LCT⁺17], but no universally agreed upon method known to the author as is for minimum compliance.

In the context of sheet thickness optimization one could use topology optimization of a 2D shell in $\mathbb{R}^3$ by interpreting it as a problem in $\mathbb{R}^2$ mapped onto the shell and interpreting the density parameter ρ as shell thickness. Here, intermediate densities may be allowed in the final design - this can be achieved by setting p in equation (2.9) to 1 and $\rho_{\min}$ in equation (2.10) according to desired minimal sheet thickness.

As equation (2.10) shows, an element's behavior at each iteration is determined only by its own deviation B_K from an overall state and not on neighboring elements, its general location or history. This freedom leads to fast computation and the possibility of finding counter-intuitive results as little user-predefinition is required. Also, it has been shown that for the minimum compliance problem in the discrete case with sufficiently large exponent p , topology optimization finds the global optimum [Rie01].

2.3.2 Geometry Optimization with Global Concept

Evolutionary Algorithms

As introduced before, in contrast to the local approach, global geometry optimization methods change a structure as a whole (within given design constraints) and evaluate its performance iteratively. Evolutionary algorithms, which work for lots of optimization problems, can also be used for mass optimization of sheet metal components as in [BAOD16]. Evolutionary algorithm's main concept is that of random alteration of a design, so-called mutation, and evaluation of this in comparison to previous generations. The randomness of mutation requires lots of computational resources, especially for complex problems with lots of degrees of freedom [MS20]. Also, there is a chance of getting stuck in a local optimum [TWP⁺18]. Nonetheless, their ability to handle complex problems and lots of constraints without needing in-depth knowledge on the underlying mechanisms [Fog97] make them highly feasible also for geometry optimization. Challenges then lie within constraint definition which may get rather complicated, requiring advanced techniques [LPB02]. If, for instance, thinking about the enormous variety of possibilities in sheet thickness optimization with respect to location, shape and number and thickness regions, it becomes visible that unrestricted constraints may lead to high complexity.

Sensitivity Analysis

If, on contrast, complexity and thus optimization degrees of freedom are low enough, evolutionary computation may not be necessary. As in [KS16] for sheet metal this may be done by predefining regions and only alternating their thickness, thus needing only one degree of freedom (DoF) for each region. By doing so, one may apply much easier and faster method of sensitivity analysis. Sensitivity analysis checks the effect of input variation to model output, which in the example of regional thickness optimization may be total deformation. What results is a response function for which an optimum within the domain may be determined subsequently, e.g. the variant exhibiting least deformation.

Load Path Based Optimization

A different and more novel approach is that of load path based optimization. The basic idea behind is the concept of load paths, which will be described more in-depth in the following.

Generally, as figure 2.10 shows, a structure that omits a domain Ω may be subjected to external constraints such as load application l and supports s . Note that while shown on the border of Ω , l and s may also be localized anywhere within the structure. The load is then transferred from l to s through Ω , inducing local deformation (strain). Following the principle of minimal potential energy [Vin93], stiffest regions take on the most load. The stiffest path through Ω can thus be referred to as *main load path* denoted P_l .

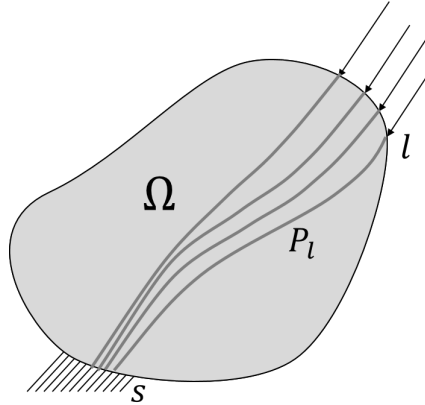
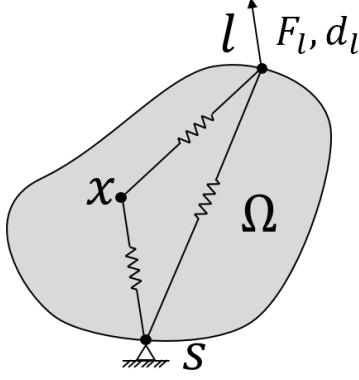
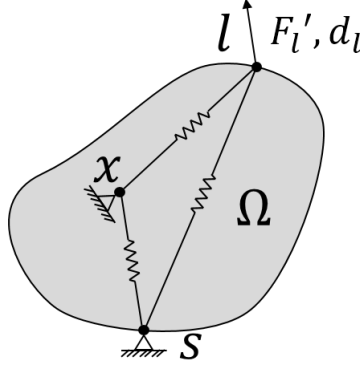
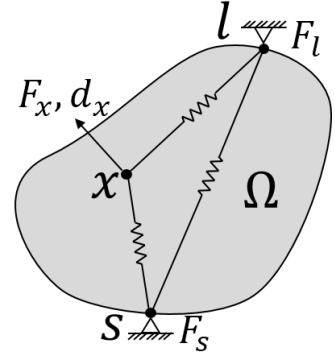


Figure 2.10: General description of a structure with load paths P_l visualized. The paths connect areas of load application l to supports s over domain Ω .

To calculate load paths, some methods exist such as analyzing principal stress trajectory as suggested in [KE95]. Another is the use of a metric for any points stiffness of connection to load application and support areas - the so-called U^* index, first introduced by [SOI⁺95]. Figure 2.11 shows the basic concept: any point x within Ω can be thought of as connected by springs with some stiffness to the supports and load application points. The stiffer this connection is, the more load will be transferred through x .


 Figure 2.11: Basic concept of U^* index.

 Figure 2.12: Concept of how to calculate U^* for point x .

 Figure 2.13: Alternative calculation using the *inspection load method*.

From [WWW⁺21], constraining the point as shown in figure 2.12 and comparing this to the unrestricted case in terms of internal energy gives a metric for this before-mentioned stiffness as

$$U^* = \frac{U' - U}{U'} \quad (2.13)$$

with U' and U being deformation energy in the constrained and unconstrained case being

$$U = \frac{1}{2} F_l d_l = \frac{1}{2} (K_{ll} d_l + K_{lx} d_x) d_l \quad (2.14)$$

and, as in the constrained case $d_x = 0$,

$$U' = \frac{1}{2} F'_l d'_l = \frac{1}{2} (K_{ll} d'_l) d'_l \quad (2.15)$$

so that

$$U^* = \left(1 - \frac{2U}{(K_{lx} d_x) d_l} \right)^{-1}. \quad (2.16)$$

Now, to get the denominator in equation (2.16) for each x , a new stiffness tensor K_{lx} would need to be obtained. For software implementation, this is computationally extensive.

A workaround is the *inspection load method* as presented in [STKA07]. Instead of constraining x for each $U^*(x)$ of interest, l is constrained for all instead, see figure 2.13. Then, inspection loads in coordinate directions are applied to x and reaction at l is recorded. From

$$F_l = K_{lx} D_x \quad (2.17)$$

with D_x being the set of deflections under all inspection loads, one can rewrite

$$K_{lx} = F_l D_x^{-1} \quad (2.18)$$

which can then be plugged into equation (2.16). That way in the discrete case, one stiffness

matrix with several load steps is applicable, instead of requiring the determination of a new stiffness matrix for each point. This makes the computation much more manageable for commonly used finite element solvers, as efficient methods to compute several load steps are available.

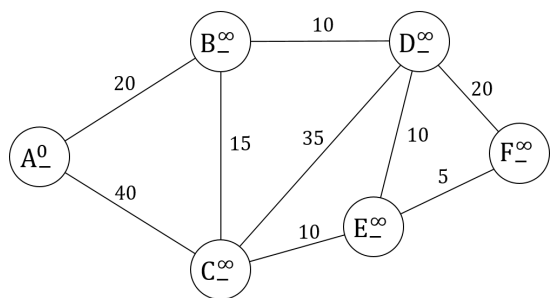
To then obtain the load path from U^* index, [SOI⁺95] and [STKA07] define it as the ridgeline of U^* - that is, the path of smallest possible gradient starting from l . However, this definition lacks generality as when switching load application and supports, which is possible by the principle of superposition, a different path would result. This dependence of load paths on the choice of load and support is acknowledged in the literature, and achieving consistency between the two paths is proposed as a potential structural optimization goal, e.g. [WPTW17] and [WWW⁺21]. Further suggested optimization goals include a uniform decrease of U^* along the path, smooth curvature and homogeneity of the so-called U_{sum}^* index, defined as the summation of U^* indices with respect to either the support or load application point (using superposition in the U^* computation). Finally, it is suggested that iterative design improvements can be performed using evolutionary algorithms or parameter studies based on these design goals as performance metrics or, in a more direct sense, by strengthening the main load path while removing material elsewhere.

2.4 Path Optimization (Graph Theory)

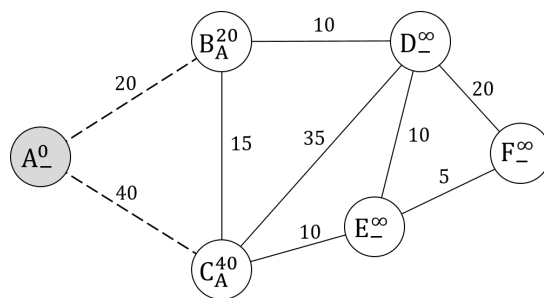
In the previous chapter a load path was defined as the ridgeline of U^* defined by the gradient from load application to support. In general, this should be applicable for all structures. However, if one may wish to find not the single main load path but one alternative path that holds true to some constraints, the introduced form may not be suitable. In graph theory there are numerous methods for finding a shortest path through some domain, some of which will be introduced here and connected with load path optimization later in chapter 4.

The core question to be answered here is that of the shortest path from one location to another. This question has a wide variety of application, e.g. in routing or network analysis. The most prominent method is *Dijkstra's algorithm* [Dij59]. This algorithm or its quicker version for singular targets, the A^* algorithm, find application in most GPS map systems such as Google Maps [McC24].

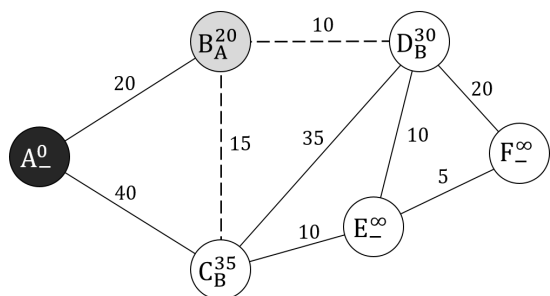
To explain Dijkstra's main functionality, consider a network of nodes connected by edges. The set of figures 2.14 gives an exemplary execution of the algorithm with detailed step-by-step visualization. Each edge has an associated *weight*, which may, for example, represent the distance between the two nodes it connects. A start node and one or more target nodes are defined.



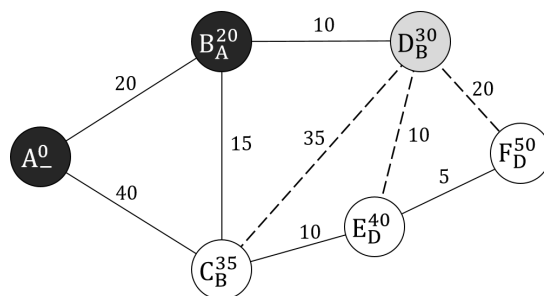
(a) Initial state. Starting at A. All edge weights known; all nodes have infinite tentative distance and empty pointers. All nodes are open.



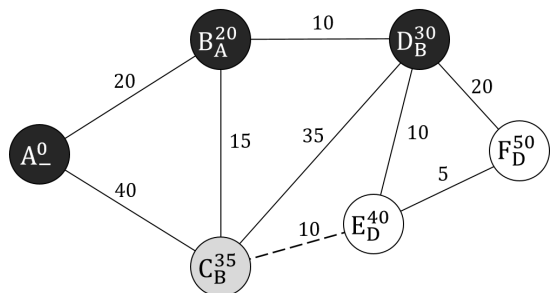
(b) Iteration 1. Neighbors of A are evaluated, tentative distances and pointers updated. A is closed.



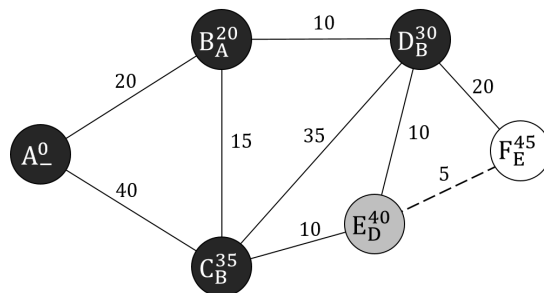
(c) Iteration 2. B has the smallest tentative distance and is visited. Neighboring open nodes are evaluated; C is updated with a shorter path. B is closed.



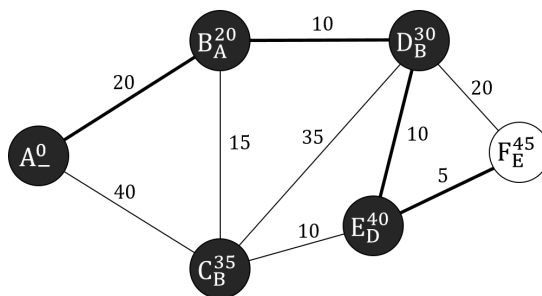
(d) D is visited as the node with the smallest tentative distance. Neighbors are evaluated; a tentative path to target F is found but not yet finalized. D is closed.



(e) Iteration 4. C is visited. Its remaining open neighbor is evaluated, then C is closed.



(f) Iteration 5. E is visited. A shorter path to F is found and F is updated. E is closed. The algorithm terminates as F is reached.



(g) Backtracking. The shortest path is reconstructed by following predecessor pointers from F back to A.

Figure 2.14: Dijkstra's algorithm on a simple network searching the shortest path from A to F. Each node stores a tentative distance and a predecessor pointer. In each iteration, the open node with the smallest tentative distance is selected and its neighbors are evaluated. Open nodes are white, the currently processed node is gray, and closed nodes are black.

Each node stores the total path length from the start node, determined by the sum of the edge weights along the path. For all nodes except the start node, this value is initially unknown and therefore set to infinity. In addition, each node contains a so-called *pointer* that can store the label of another node; this pointer is initially empty for all nodes.

Visited nodes are stored in a *closed* list, while nodes that have not yet been visited are kept in an *open* list. Initially, all nodes are in the open list.

Starting from the start node, the edge weights to all neighboring open nodes are evaluated. These weights are assigned as tentative path lengths, and the pointers of the neighboring nodes are set to the start node. The algorithm then moves to the nearest neighboring node and closes the start node. At this point, the shortest path to the newly visited node is guaranteed to be the direct path from the start, as no shorter alternative can exist.

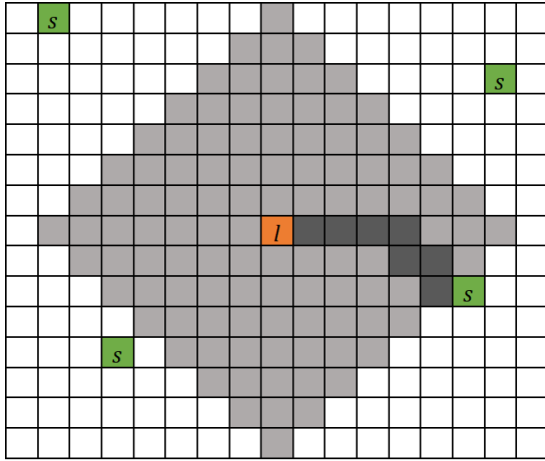
Now, again, the edge weights to all neighboring open nodes are evaluated. If the sum of the current node's path length and the connecting edge weight is smaller than a previously assigned value, the corresponding node's path length is overwritten and its pointer is updated to the current node. After this evaluation, the current node is closed, and the algorithm proceeds to the open node with the globally smallest tentative path length. This node either points directly to the start node or to the previously visited node - in both cases, its stored path is guaranteed to be the shortest possible.

Repeating the process of evaluating and, if a shorter path to neighboring nodes is found, overwriting their distance and pointer, until the node visited is the target node, successfully gives the shortest path length. Subsequent backtracking using the predecessor pointers stored in the nodes gives the shortest path. This is always true, regardless of number of nodes or their layout, as long as all distances are strictly positive.

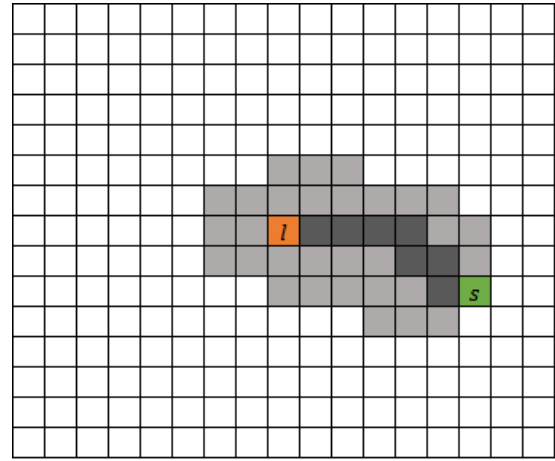
Dijkstra's algorithm is thus able to find a shortest path with very little input and moderate programming effort required. It can be modified to find paths with other properties as well, e.g. the quickest path where instead of using distance one may use traveling time from node to node, or some other edge weight function.

However, Dijkstra's algorithm does not actively search for a path leading to the target, it is more by pure chance that it eventually finds it after evaluating all paths to nodes closer to the start. On big networks with relatively uniformly spread nodes one can think of the search region expanding spherically around the start, up until a radius close to the straight-line distance between start and target, as figure 2.15a shows with the start labeled l and the target labeled s . A lot of nodes in a region going from l to the opposite direction of s will be analyzed before the end is reached. The likelihood of the shortest path going through that area is - while certainly not nonexistent - small. While not omitting nodes in that area, it would make sense to search the path first where it's expected, which is the general direction of s . This is incorporated in the A^* Algorithm [HNR68] which adds the minimum expedited distance from any node to s , which is their straight-line distance,

to the nodes path length data. That way, the next visited node is one which is close to the start as well as prospectively close to the end, while not violating the core mechanism of Dijkstra’s algorithm and thus finally finding the shortest path overall. This method is generally much quicker on huge networks as far less nodes need to be visited (except for the rare case of shortest path being counter-intuitively far around), see figure 2.15b, however it works only for a single target node while Dijkstra can have an arbitrary amount. As many problems such as routing in navigation software only require one target location though, A^* finds huge application in software.



(a) Dijkstra Algorithm. Start l with several targets s .



(b) A^* Algorithm. Start l with singular target s .

Figure 2.15: Schematic of region searched before finding shortest path from l to target s on a uniform rectangular network for path finding algorithms. While Dijkstra’s Algorithm searches in all directions until finding a target, A^* Algorithm searches in target direction first, thus being potentially far more efficient. However, if several targets exist, it cannot be applied.

Chapter 3

Discussion and Requirements

3.1 Comment on the Current State of the Art

In the previous chapter, it has been shown that sheet metal manufacturing is suitable for lightweight design. This is due to its ability to create thin-walled structures with high area moment of inertia at low mass. Modern materials with high mass specific properties like specific strength drastically increase lightweighting potential for strength optimization. However, stiffness optimization with sheet metal is barely possible like that due to comparable specific stiffness for all metals. Also, higher performance materials usually come with a trade-off in manufacturability. An alternative approach to reduce mass is leaving sheet metal structure's main property of uniform thickness and getting a variation in sheet thickness instead. Lots of methods are applied for this, trying to compromise for cost effectiveness and ease of use with high lightweighting opportunity. It is concluded that the field of non-uniform thickness sheet metal components for automotive application has potential for further research.

A lot of potential in terms of mass reduction within this field lies in patching of a component after forming the main shell, as this allows for high geometry and material degrees of freedom. This way, the main shell can be made of thinner material than conventionally, benefiting manufacturing while significantly reducing mass. It has further been shown that adhesive bonding of these patches is a feasible joining method: It shows theoretically monolithic behavior of the patched regions, except for boundary zones, and no stress concentration in the sheets within those, showing feasibility to strengthen stressed regions. Also, it already finds widespread application in automotive with gluing and curing being easily integrated in automotive manufacturing processes.

In terms of designing patched sheet metal components, there is a variety of methods assisting in the search for optimal geometry. Topology optimization finds widespread application as an indicator for then manual design optimization, however it is not feasible for immediately producing close to manufacturable designs as they miss an evaluation

of the latter and do not give discrete results in terms of thickness and patch boundary. Still, their ability to find global optima and ease of use makes them a very valuable tool also for sheet metal lightweighting. Other methods exist to more directly define a patch, which are evolutionary algorithms and sensitivity analyses (for predefined patch regions). The first being very computationally extensive, the latter highly dependent on predefinition, being less-able to produce counter-intuitive results. It can be deduced that methods for sheet metal lightweighting must compromise between the ability to develop unrestricted designs on the one hand, and the being able to account for manufacturing constraints on the other. While a purely unrestricted method may produce the lightest and most counter-intuitive concepts, by then translating them manually into a working design, a lot of lightweight potential will be lost and the optimality of the result may not be given any more. On the other hand, a method with high pre-restriction may produce immediately manufacturable results, requiring little post-processing, but is highly biased towards the user's own idea of a lightweight design and can thus be used best only for refinement of an initial idea. There seems to be a gap between these two approaches: an optimization method specific to sheet metal that requires little restrictions on the design space but is able to account for manufacturability.

From these observations it is concluded that there is potential for a structural optimization method specifically for sheet metal components that uses bonded patches to achieve non-uniform thickness and incorporates manufacturing restrictions while still being able to produce counter-intuitive results without too much restriction on an initial design. Such a method's requirements will be defined further in the following section before presenting this thesis' approach to it.

3.2 Requirements Derivation for a Patch Optimization Method

The hypothetical method of determining the optimal location, boundary and thickness combinations for reinforcing sheet metal components is referred to as *patch optimization method* in the following. The method should address the gap between practical manufacturability and universal applicability while maintaining the discrete nature of sheet metal thickness. Some key requirements can be derived to adequately fill this gap. They are listed here by descending priority and further explained subsequently, as follows:

1. Performance Evaluation: Optimize a sheet metal geometry so that it sustains performance with respect to strength and stiffness at lower overall mass than initially.
2. Manufacturability: Give results which can be manufactured from sheet metal.

3. User Independency: Be able to produce counter-intuitive results by requiring no user-predefinition of patches.
4. Restrictability: Be able to handle user-defined geometrical restrictions such as non-reinforcable regions while still considering them for the overall performance assessment.
5. Efficiency: Be computationally efficient.

Requirement 1: Performance Evaluation

The first requirement means that the method should improve an existing initial design with respect to its overall mass without decreasing the part's strength or stiffness. A structural component (in automotive) within the scope of this method is designed to carry load defined by one or several load cases. It is designed to be stiff, so to deform little under load, and be strong, so be able to handle the load without failure, which would be breaking or deforming plastically, depending on the use case. Based on the load case and the component design, either stiffness or strength may be the deciding factor for the component's rigidity (and thus, sheet material and thickness). It can be assumed for lightweight optimization, that the component must stay as stiff and as strong as the initial existing design is, which is the initial designs *structural performance*. So, whenever mass is reduced, it must be checked that the structural performance is still met. The method must thus be able to check for this and in the final result give an indication how much mass is reduced at consistent structural performance. This requirement is common to most other mass optimization methods based upon initial designs, such as the aforementioned sensitivity analyses.

Requirement 2: Manufacturability

The second requirement is to ensure that the resulting geometry can be manufactured with sheet metal manufacturing processes. As sheet metal is of discrete thickness, the resulting geometry must give uniform thickness base geometry and also uniform thickness patches. Furthermore, there needs to be a finite amount of patches in the design. For achieving the latter, the method must incorporate a concept of location and connection to be able to design singular connected patches. It must analyze the given structure as a whole and evaluate, which areas are suitable to patch, not for each singular point individually but interlinked with its surroundings. For example, in topology optimization, this is generally not the case as each element is analyzed individually, see equation (2.10) which describes element density growth. Only a concept of where any points location and connectivity to other points are part of the evaluation can achieve this, which are global concepts as differentiated previously in section 2.3. For achieving discrete thicknesses, the method

must be able to transfer its considerations of where to place a patch into uniform thickness patches. This thickness, as well as the base part’s thickness, must be tuned to the optimal ratio. This requirement is crucial to a method’s novelty and usability in this niche. The abilities to *find regions for patching*, *to discretize the regions to uniform-thickness patches* and *to tune base geometry and patch thickness* are fundamental for addressing it.

Requirement 3: User Independency

While the second requirement differentiates from general methods such as topology optimization, the third requirement describes the differentiation from state-of-the-art sheet metal optimization methods that start from a given patch design and only alter thickness or move the patch boundary. The user should only give overall restrictions based on manufacturing possibilities (such as a thickness range or material) and the overall design space and have no influence on the patch solution in itself. Otherwise, the chance for counter-intuitive patch designs decreases. For the method, this means that no predefinition of the patch should be made.

Requirement 4: Restrictability

Contrary to requirement 3, the fourth requirement states, that some user input should still be possible to account for manufacturing or design space restrictions. This benefits overall manufacturability as it can be assumed that not every location on a complex shaped geometry can be accessed with a patch, e.g. undercuts, inner radii, highly distorted surfaces from forming or uneven weld seams. On a first glance, it seems straight-forward to be able to exclude some regions from patching. However, it must be taken into account that restricted regions still add to the overall structure in terms of strength or stiffness, since the base part’s thickness will generally be nonzero for a patching method as discussed here. So, restricted regions can also see stress and cannot be excluded from the performance evaluation. This is especially important in the case of stress concentration: If some stress hotspot lies within a restricted region, a decrease in sheet thickness there (which is required to give room for patching elsewhere) would, in the first place, increase stress as stress does not converge to 0 with decreasing thickness due to nonlinearity (e.g. buckling). One needs to be able to take this into account and, figuratively, shift stress away from it, further emphasizing the need for a global concept.

Requirement 5: Efficiency

Finally, any method aimed at practical use is required to be computationally efficient. Optimizations for components in the size range of structural automotive sheet metal parts should run in a reasonable time-frame at sufficient resolution. While this requirement is

not further defined here, it stays as a reminder to consider efficiency when developing the method for the other main requirements.

Chapter 4

Design Method

4.1 Core Concept

In the following, a method to find regions to reinforce and subsequently design fully bonded reinforcement layers for shell components is presented. It addresses the requirements stated in section 3.2. For clarification, the main component is called *base* while the added layer is called *patch*. A *region* describes some connected and defined area on the base. A base part's region labeled as *unpatched* means that from an initial design, thickness is reduced. A reinforced region labeled *patched* has, while obtaining the same decrease in base's thickness, an increase in thickness due to the added patch layer. *Performance* describes the components fulfilment of a requirement such as maximum internal stress or stiffness with respect to a selected load case. The method is also referred to as the (*patch*) *optimizer*.

4.1.1 Basic Principle

Underlying idea of this method is, that by removing thickness from the base and using a part of that then available mass to put patches in the right regions, an overall net reduction of mass can be achieved while maintaining performance. This is due to imperfect use of material for constant thickness components which in reality will never be uniformly stressed throughout or taking the same strain energy everywhere, respectively.

Generally, the question arises: What is the most useful region on a component to add a patch to? As this can only be done by removing material elsewhere, i.e. reducing base thickness (since otherwise, the overall component gets heavier than initially), it follows: Where can material be removed without issues? This shift of material from one region to another is then finally a question of: *Where would material be used most efficiently?* The way this thesis tries to answer this is that the most efficient use of material is where it can fulfill a component's function, which is carrying load from one place to another (or from several places to several others) as directly as possible. The most direct way to

carry load is by creating stiff and short load paths. This makes an overall stiff component, so less internal energy through deformation and also possibly less internal stress in the component as there are no high local deformations.

The formation and strengthening of main load paths is inherent to all structural optimization, independent of the method. For example, it can be seen in topology optimization, even in the unpenalized case that does not force convergence, see figure 2.9a earlier. The reason is that strengthening of some location will increase its stiffness, but also increase its share of the total load which stresses everything that lies in the load path leading through it, requiring stiffening of the whole path.

This can be visualized with a simple example as follows: Imagine a structure made of truss elements E_{ij} forming m parallel load paths, in the following called "legs", as figure 4.1 suggests. The elements act as linear springs. Each leg consists of n elements in line. Only a single degree of freedom in direction of the springs is considered, so buckling or bending is disregarded.

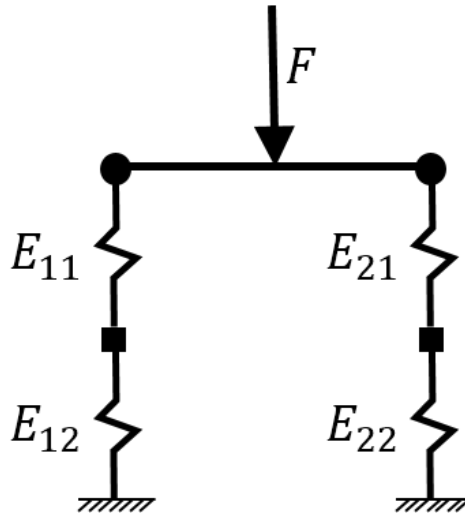


Figure 4.1: Arrangement of $m = 2$ parallel "legs", consisting of $n = 2$ in-line truss-elements each that transfer a load to the support. Altering individual stiffnesses changes the load share of the legs.

Each truss has stiffness K_{ij} which is proportional to cross sectional area and thus, its mass with $i = 1 \dots m$ referencing the leg and $j = 1 \dots n$ the position within the leg. Using the standard formulations for parallel and series spring arrangements [WMJV11], in-line arrangements within legs add up to a stiffness of

$$K_i = \left(\sum_j^n \frac{1}{K_{ij}} \right)^{-1} \quad (4.1)$$

and the parallel leg arrangement leads to overall stiffness

$$K = \sum_i^m K_i. \quad (4.2)$$

The load F is transferred through the legs to the ground, split between each leg depending on their stiffness ratio, each seeing the load share:

$$F_i = F * \frac{K_i}{K}. \quad (4.3)$$

Now, consider transferring a fraction of the mass of all other elements to a single element. The stiffness of that element increases proportionally, while the stiffness of the remaining elements decreases accordingly. If all initial stiffness $K_{ij}(0)$ are the same, an increase by factor x for one element leads to a stiffness increase for the element $K_{ij}(x) = xK_{ij}(0)$ but, as the others decrease, overall stiffness also decreases, as can be seen from equations (4.1) and (4.2), leading to

$$K(x) = \frac{m(mn - x)(mnx - mx + m - 1)}{n(mn - 1)(mnx - mx + m - x)} * K_{ij}(0). \quad (4.4)$$

Which, in the case from figure 4.1 is

$$K(x) = -\frac{(x - 4)(2x + 1)}{3x + 6} * K_{ij}(0) \quad \text{for } n = m = 2. \quad (4.5)$$

This function decreases for $x > 1$ if $n, m > 1$. Stiffening a single element does not increase overall stiffness. Also, by doing so, load share in its leg increases with

$$F_i(x) = \frac{(nm - 1)x}{nm^2x - m^2x + m^2 - m} * F \quad (4.6)$$

as load share is proportional to leg stiffness K_i , see equation (4.3). In the example this would be

$$F_i(x) = \frac{3x}{4x + 2} * F \quad \text{for } n = m = 2. \quad (4.7)$$

Thus, all elements in-line with the reinforced one see increasing load. For non-reinforced elements this occurs at decreasing stiffness (and corss sectional area), leading to higher internal stress in those elements. It can be concluded, that stiffening only a single element is not beneficial in this arrangement. If one, however, shifts mass from one leg to another leg as a whole, things change: Now, overall stiffness stays the same, i.e.

$$K(x) = \frac{x}{m}K_{ij}(0) + (1 - \frac{x}{m})K_{ij}(0) = K_{ij}(0) = K(0). \quad (4.8)$$

Also, load shift is proportional to stiffness, so each element will be stressed equally.

What can be observed from this example? Whenever any elements get reinforced, the whole branch, so all elements in-line to it must be reinforced as well to keep stress and stiffness consistent. If in-line elements do not get stiffened they see an increase in stress since load gets "shifted" towards them without compensating for it. A parallel shift of mass (from one leg to another) does not change stress, nor does it alter overall stiffness, if all parallel legs exhibit the same initial stiffness. Transferring this back to structural optimization in general: Any reinforcement must reach all the way from load application point to support point, i.e. along the load path, to be effective. Load can be shifted from one parallel load path to another without compromising overall stiffness.

In an idealized one-dimensional case as in the example above, there are no benefits to reinforcing one load path over the other. All load paths (or there, legs), had the same initial stiffness. Taking a fraction of mass from one path to the other had a proportional increase in stiffness on the latter just as it had a decrease on the former. However, in a real-world geometry, two load paths may not be of the same "efficiency", in this case meaning how much stiffness they can add to the structure for a given amount of mass. If shifting mass from an inefficient load path to a more sophisticated one, a net increase in overall stiffness could be achieved, or likewise, consistent overall stiffness could be achieved with less overall mass. The same applies to strength, meaning the induction of local stresses in the elements, as an efficient and uniform load path deforms less and thus induces less stress in its individual elements.

This, as introduced earlier, is the fundamental assumption of this thesis' method: The selection of efficient load paths achieves the highest stiffness. With a global view onto the component, strengthening it along the most suitable paths as a whole, a lightweight design can be created.

4.1.2 Incorporating the Requirements

So far, only the basic principle of reinforcing along load paths was described. How can this be extended to a functional optimization method that meets the requirements from section 3.2?

Firstly, the optimizer, after defining patches, will search for the ideal combination of base to patch thickness that shows minimal mass while still fulfilling performance. Thus, requirement 1, *performance evaluation*, is met, by definition, since any patched design that may unintentionally lead to less performance than the initial design cannot be suggested by the optimizer.

A more crucial step is the incorporation of requirement 2, *manufacturability*. The concept of load paths must be translated into a manufacturable design consisting of a base with discrete thickness and patches with discrete boundaries and thickness. A path, which by itself is only a spline in 3-D space, must be translated into a volumetric patch.

For doing so, the patch width, i.e. its extension perpendicular to the path direction and the patch thickness are used as parameters along the path, describing the cross-sectional area. While the patch thickness must be uniform from the sheet metal manufacturing constraints, its width can alter more or less freely, depending on the existing space on the base, meaning if it can extend far enough without reaching the base's edge or a restricted area. Given that, the continuous parametrization of the patch width - instead of its thickness, which may be the intuitive approach when disregarding manufacturability - thus, allows a representation of non-uniform cross-sectional area along a path. As the path may go through areas of less stiffness (by design), the parametrization of width can lead to an overall uniform patch with continuous stiffness.

Requirement 3, *user independency*, states that the user should not influence the optimization in a way that can falsify the result. As the user only gives boundary conditions, such as constraints, which are not subjective, independence of the optimizer is maintained. The optimizer then finds main load paths independently. To account for requirement 4, *restrictability*, however, some user restrictions can be given. Some areas may be unfavorable for patching as described earlier in the requirement definition. This restrictability heavily influences the path finding. State-of-the art approaches for load path finding such as in U^* , which uses the ridgeline of the U^* -gradient as described in section 2.3.2, do not allow for restrictions in the pathway definition. Disregarding some areas in the U^* computation would falsify the results, also for unrestricted areas. Instead, general route-finding algorithms commonly used in navigation as introduced in section 2.4 are applied to this: By separating the location-based connectivity indication of U^* and the pathfinding, the path algorithm is able to find new load paths that are stiff but also efficiently incorporating restrictions - the algorithm searches not the *stiffest* route, but the *stiffest unrestricted* route.

To account for requirement 5, *efficiency*, some functions are incorporated into the method. A quick variant of U^* -computation, the inspection load method proposed by [STKA07] as introduced in equation (2.18), is used. Interpolation of U^* -values from a selection of calculated nodes to the whole structure is implemented. The Dijkstra-Algorithm, an efficient route-finding algorithm is selected. Base and patch thickness is iteratively determined and interpolated.

In summary, the key novelties introduced with this method to address the optimization of sheet metal components by patching are the following:

- Optimizing a geometry not by local analysis of internal stress or strain energy but by globally adapting a design principle of efficient load paths.
- Calculating load paths based on route-finding algorithms to be able to not only find existing main load paths but create new main load paths incorporating manufacturability restrictions,

- Discretizing main load paths to defined patches, keeping the need for manual post-processing low

In the following, the method is described more in detail. The structured process and mathematical models for the individual functions are explained.

4.2 Function Structure

The patch optimizer can be subdivided into functional elements, see functional block diagram figure 4.2. Starting and end point are geometries, in practice this being 3D-CAD models. Some geometry needs to be existent beforehand or be created initially; in contrast to some other methods such as topology optimization, a design space is not sufficient. However, one may use the outermost boundaries of the design space as a starting point, representing maximum cross section when designing for closed hollow structures. When aiming for an open single-shell design, one could use some subsection of the design space boundary accordingly.

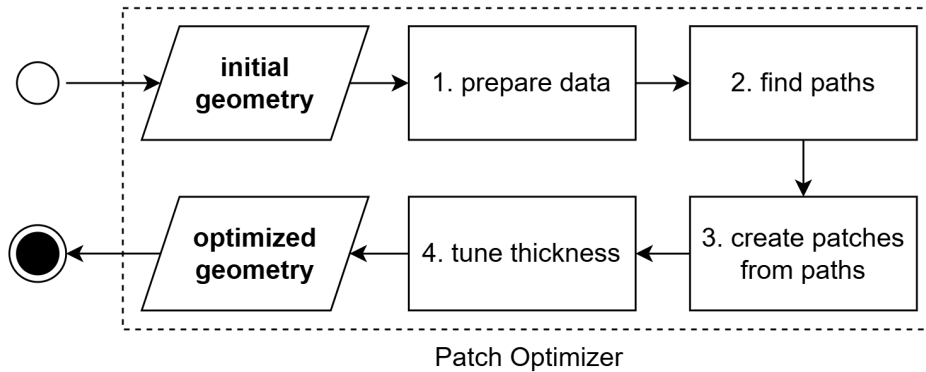


Figure 4.2: General function structure of the patch optimizer, consisting of 4 modules. From an initial geometry, an optimized geometry with reduced overall mass is computed.

From the initial geometry, analysis is being prepared by importing the geometry and adding relevant constraints and properties. After that, paths are searched and patches based on those created. In the end, thicknesses of base material and patches are adjusted to meet optimization targets, finally resulting in a new geometry which will still have the same overall shape but a change in thicknesses throughout.

The function details will be presented in the following. For clarification, elements will be called e , element sets E , nodes n and their sets N , while, as each element contains some nodes, any e can also be interpreted as a set of nodes. In iterative process description, a variable $x \rightarrow x'$ describes an overwrite of x at end of iteration (with x being a placeholder for any variable).

4.2.1 Prepare Data

In this subsection, all is prepared for the following calculations and relevant input given. Figure 4.3 shows the process.

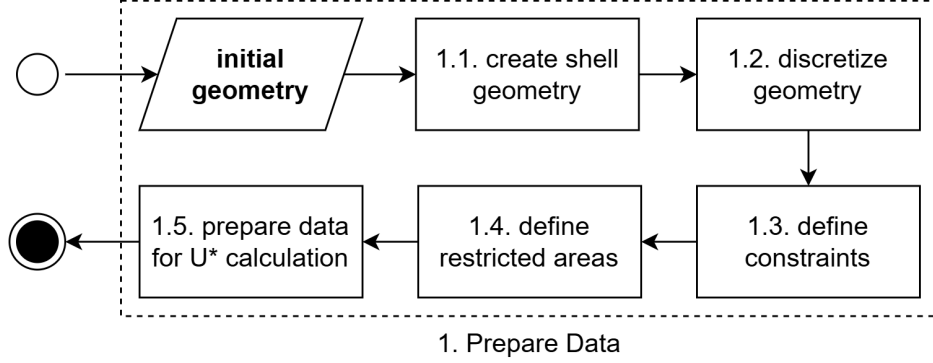


Figure 4.3: Function structure for patch optimizer module 1, Prepare Data. Here, the model is prepared and necessary information given to run the optimization.

From the initial geometry, first, a surface body is to be derived. Typically shell geometries are designed as surface models with thickness added as a parameter only. If not, one typically tangential surface of the body is to be extracted. For a closed geometry, the inner surface should be chosen as the reference surface: With material growth direction pointing outwards, the patch thickness will be added on top, placing the patch on the outside of the structure. This way, manufacturing with the patch added onto a closed shell is accounted for. For open geometries this selection is up to choice, depending on the desired side to patch. The base geometry thickness is to be kept as t_0 . If the geometry is already on the very edge of its design space, one must take notice that the patched geometry later may slightly exceed the design space due to the higher thickness of base+patch even with the inner surface as reference. In that case, one would have to move the reference surface inwards by some safety margin.

The shell geometry is then to be discretized, meaning approximating it by a finite amount of simple geometry elements such as triangles or quadrilaterals. The number of discrete elements should be fine enough to create patch geometry in the desired resolution with respect to edges and surface curvature. A good approximation is a general element size of two times the initial base thickness and refinement in curved regions and on edges to around half the radius of curvature. Each element must consist of nodes and information on element connectivity by shared nodes must be given, as well as their coordinates. The shell defines domain Ω with E_Ω being its element and N_Ω its nodal representation.

Then, the load case is to be described. As of now, the optimizer can only account for one singular load case, so several load cases must be combined beforehand. For the load case, some nodes can be constrained in directional or rotational movement, representing the component supports. Also, some nodes can be loaded according to the load case.

There is no restriction to the amount of them, however the optimizer cannot account for different loads on individual nodes or node groups. Load is F_L . The support nodes define the set N_S while the load application nodes make up N_L .

Some regions R and their respective elements E_R might not be suitable for adding a patch. This could be due to physical accessibility when patching, or functions those areas need to maintain, such as attachment points for other non-structural components. Also, some areas might be possible to patch, but unfavorable. They can be penalized for use in path finding by penalty factor p , which is infinite in the case of hard restriction and between 0 and ∞ otherwise.

Some other preparation for the following calculations need to be done. As U^* calculation is computationally extensive, but element and node numbers are high to have a good geometric representation, evaluating U^* for each node would use a lot of resources. Since, however, U^* is a quantity not expected to have a high local variation, a calculation on a subset of nodes and subsequent interpolation can be used. As a preparatory step, those calculation nodes are chosen so that they spread uniformly over the domain at random. This ensures sufficient representation while decreasing computation time significantly.

From an initial user-specified target number of to be selected nodes $q_{X,target}$, first, a rectangular bounding box encompassing nodes $N_\Omega \notin \{N_S, N_L\}$ with according outer dimensions is created. The box is then subdivided into a uniform cubic grid with approximately $q_{X,target}$ vertices (being strictly higher than $q_{X,target}$, i.e. the next highest resolution for the given bounding box dimensions). The vertices are slightly jittered to counteract grid-dependency. Closest nodes to each vertex within a fixed range smaller than the grid resolution are chosen, resulting in $q_X(1)$ selected nodes. See figure 4.4 for clarification. Since the geometry does not necessarily occupy its bounding box completely in terms of nodes existing within each grid quadrant, there will not be a node found for all vertices. As a result, the number of selected nodes will be significantly smaller than desired, i.e. $q_X(1) \ll q_{X,target}$. Thus, iterative grid refinement until coming close to the target amount, i.e. $q_X(\infty) \approx q_{X,target}$, is required. What results is a geometrically uniform node selection N_X independent on node location, grid and importantly, independent on local mesh size – pure random selection by node index would favor regions which are finely meshed, which is undesired. The geometry, constraints and nodes for U^* calculation are then given to the next function block *find paths* for calculation.

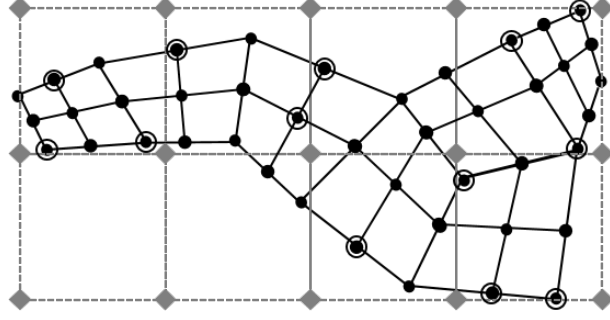


Figure 4.4: Grid over a mesh with closest nodes to the grid's vertices selected. This schematic explains the selection process for achieving a spatially uniform selection of nodes, even if the nodes themselves are not spread uniformly.

4.2.2 Find Paths

For the pathfinding function structure, see figure 4.5. In the patch optimizer's first solver function element, U^* is calculated. After interpolation, restricted areas are modified and subsequently paths searched. What results is a list of all desired main load paths.

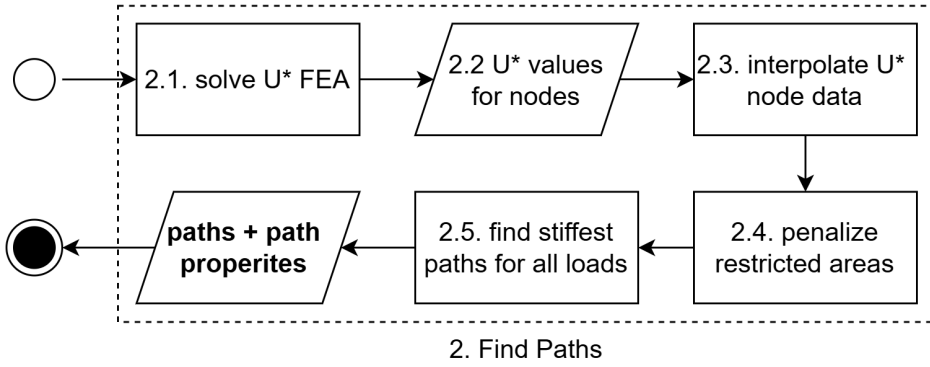


Figure 4.5: Function structure for patch optimizer module 2, Find Paths. Here, the desired main load paths are computed from U^* -analysis, restricted areas and path-finding algorithms.

U^* calculation follows the Inspection Load Method introduced in chapter 2.3.2, originally proposed by [STKA07] and equation (2.18). Initially, nodes N_L are loaded with design load F_L to get deflections D_L for U in equation (2.14) which is done by solving the global stiffness equation

$$F = KD \quad (4.9)$$

for all nodal displacements D . Then, N_L are fixed and instead, inspection loads are step-wise applied to all N_X . As there are three independent inspection loads for each, a total of $3q_X$ load steps are performed.

$$F_{X1\dots q} = \begin{bmatrix} F_u \\ F_v \\ F_w \end{bmatrix} \quad (4.10)$$

describes the three axis-aligned inspection loads in general while LS is the list of load steps, which is

$$LS = \begin{bmatrix} F_{X1} \\ F_{X2} \\ \vdots \\ F_{Xq} \end{bmatrix}. \quad (4.11)$$

While not necessarily required, inspection loads for all nodes are set the same. Any cartesian coordinate system may be chosen. Now, from solving equation (4.10) for all load steps, the reaction forces F_L at N_L as well as displacements D_X at N_X can be obtained to solve for K_{LX} analogous to equation (2.18) by

$$K_{LX} = F_L D_X^{-1}. \quad (4.12)$$

Plugging into equation (2.16) finally yields U^* , here being

$$U^*(X) = \left(1 - \frac{2U}{(K_{LX}d_X)d_L} \right)^{-1}. \quad (4.13)$$

To get element U^* values for all E_Ω , interpolation is to be performed. This is done by searching three nearest nodes $N_{i,j,k}$ containing information, i.e. $N_{i,j,k} \subset N_X$. Then, a weighted average of these is taken by

$$\hat{\delta}x_{i,j,k} = \left(\frac{\delta x_{i,j,k}^{-1}}{\sum \delta x_{i,j,k}^{-1}} \right) \quad (4.14)$$

$$U^*(E) = \sum_{i,j,k} \left(U_{i,j,k} * \hat{\delta}_{i,j,k} \right) \quad (4.15)$$

with $\delta x_{i,j,k}$ being distances to the nodes and $\hat{\delta}x_{i,j,k}$ being their relative proximity, weighting near nodes higher. Note that neither node selection nor their distances can be evaluated by vector distance alone since nodes can be close in space but not closely connected. Instead, distance over Ω is to be evaluated. An algorithm searching nearest nodes is to be implemented by iteratively checking neighboring element's nodes and tracking path lengths. Exemplary for one element e_i – which, as stated earlier, can be seen as a set of nodes – this is done by finding the set of elements $E_{neb} \subset E_\Omega$ that are directly connected

to E_i . These are all elements which share at least one node, so

$$E_{neb} = \{e \subset E_\Omega \mid e \cap e_i \neq \emptyset\}. \quad (4.16)$$

If, now,

$$|E_{neb} \cap N_X| \geq 3 \quad (4.17)$$

one has found a sufficient amount of nodes. Otherwise, the process is repeated by growing E_{neb} to

$$E'_{neb} = \{e \subset E_\Omega \mid e \cap E_{neb} \neq \emptyset\} \quad (4.18)$$

until condition (4.17) is true. Element distance can be determined by element center distance to E_i noted $\delta x_{e_i, e_{neb}}$ for each $e_{neb} \in E_{neb}$ over the corresponding rings of neighboring elements. For performance, already evaluated element's nodes can be added to N_X after each interpolation step to potentially decrease search time for the next element. The set of all element's U^* values is denoted as U_Ω^* .

To penalize areas as discussed earlier, their U^* value is artificially increased by penalty factor

$$U_R^* = p U_\Omega^*. \quad (4.19)$$

This may seem counter-intuitive as it appears that increasing their stiffness makes them more favorable for patching. However, as will be presented in the following, the path finding algorithm favors steep negative gradients to find an overall steep path.

For path finding, first, starts and targets need to be defined. As path finding is element-wise (due to R being a set of elements), start elements are defined as being connected to load nodes N_L :

$$E_L = \{e \subset E_\Omega \mid e \cap N_L \neq \emptyset\} \quad (4.20)$$

and accordingly targets,

$$E_S = \{e \subset E_\Omega \mid e \cap N_S \neq \emptyset\}. \quad (4.21)$$

Path finding is carried out for each start element to account for spread out load application. Basis is the aforementioned Dijkstra Algorithm. Initially, all elements are in open list O while closed list C is empty. For each element e_O in O there is an initially infinite path length δ_O and a so far empty pointer πO that can point to other elements.

From start element $e_i = e_s$, expense $\delta z_{i,j}$ to each neighboring element e_j is to be evaluated. Dijkstra Algorithm requires this to be strictly positive, which is why evaluating based upon gradient ∇ alone from

$$\delta U_{i,j}^* = U_j^* - U_i^*, \quad (4.22)$$

$$\delta x_{i,j} = x_j - x_i, \quad (4.23)$$

$$\nabla = \frac{\delta U_{i,j}^*}{\delta x_{i,j}} \quad (4.24)$$

would not be sufficient - both negative and positive gradients exist. Furthermore, using the gradient directly in the cost function would cause the cost of entering and again leaving a penalized region to cancel out, so that traversal of such regions would not be penalized. Instead, the variable ν is introduced to transform gradient into a "velocity":

$$\nu = \sqrt{1 + \frac{-\nabla}{\sqrt{(-\nabla)^2 + 1}}}, \quad (4.25)$$

which is used in the cost function

$$\delta z_{i,j} = \frac{\delta x_{i,j}}{\nu}. \quad (4.26)$$

That way, $\delta z_{i,j}$ is kept positive while favoring directions of steep negative gradient. Figure 4.6 shows the graph of ν over ∇ . The equation (4.25), albeit designed by the author to fit the specific task, is inspired by physical velocity over gradient of an object with constant propulsion, adding to the velocity analogy. With this, the expense δz can be thought of as travel time expense, when relating this to the navigation analogy. Like this, not the geometrically shortest path as conventionally, but the least expensive one is found. As can be seen from the graph, ν goes to zero and thus expense δz increases strongly for high positive gradients. That way, entering a penalized region with artificially high values of U^* is impeded while leaving it again does not fully compensate for it, making crossing penalized regions inefficient.

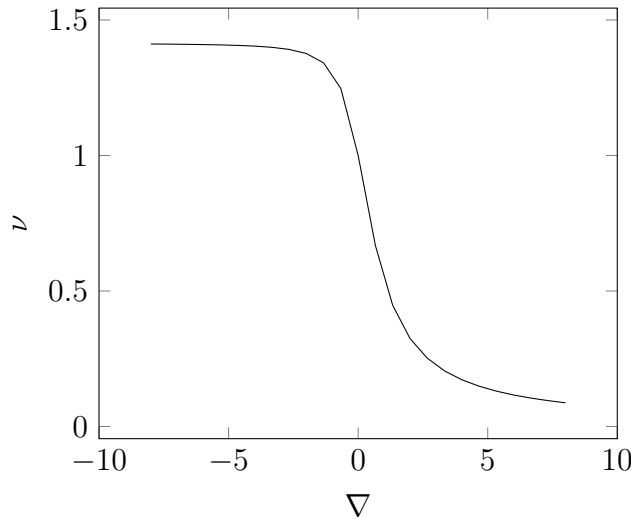


Figure 4.6: "Velocity" ν over gradient ∇ , affecting the expense for traveling a path. A high negative gradient causes high velocity. Positive gradients cause low velocities.

After this evaluation, the entries for all evaluated neighbors in O are updated so that

$$\delta O'_j = \min \{ \delta O_i + \delta z_{i,j} , \delta O_j \} \quad (4.27)$$

$$\delta O'_j < \delta O_j \implies \pi O'_j = e_i \quad (4.28)$$

The element with lowest current cost δO_j is chosen for the next step, i.e. $e'_i = e_O(\min)$, while e_i is moved to the closed list C . The process is then repeated until $e_i \in E_S$ so a target e_s is found. Back-tracing through pointers finally gives the path

$$p_l = \{ \pi O_s, \pi O_{\pi O_s}, \dots, e_l \}, \quad (4.29)$$

$$p'_l = \{ p_n, \dots, p_1 \} \quad (4.30)$$

which is flipped for convention. Repeating the above search for all $e \in E_L$ leads to the set of all paths P_L . Storing also path's meta data

$$\delta x_{p_l} = \{ \delta x_{1,2}, \delta x_{2,3}, \dots, \delta x_{n,s} \} \quad (4.31)$$

and

$$\delta U_{p_l}^* = \{ \delta U_{1,2}^*, \delta U_{2,3}^*, \dots, \delta U_{n,s}^* \} \quad (4.32)$$

finishes the algorithm. A path, that is assumed to be the ideal load path incorporating the restrictions, is found.

4.2.3 Create Patches from Paths

In this section, paths and their information are used to create patches. Figure 4.7 shows the general process.

A path, while in the discrete case being a selection of elements, is for now just a line going through Ω . To get a patch from it, a width along the path is to be added. There is no need for the width to be uniform as it can be used as a parameter reacting to local conditions. Width is then a function of location along the path, as figure 4.8 suggests.

Width should account for three properties. Firstly, a design factor describing the broadness of patches in general, as one may wish for generally wide or narrow patches depending on preference and patch bonding strength. It may be expected that narrower but ultimately thicker patches lead to better results as they do not span so far away from the ideal load paths. However, shear load transfer from base to patch would then go over a relatively small contact area, requiring high shear strength of the patch bond. Also, thicker patches protrude further outwards which may lead to design space issues.

This general width factor will be denoted

$$k_w = \text{const.} \forall p_l \in P_l. \quad (4.33)$$

As will be explained in chapter 4.2.4, different values for k_w may be tried, modifying it to a set of n values as

$$k_w = \{1 \dots n\} k_{w0} \forall p_l \in P_l. \quad (4.34)$$

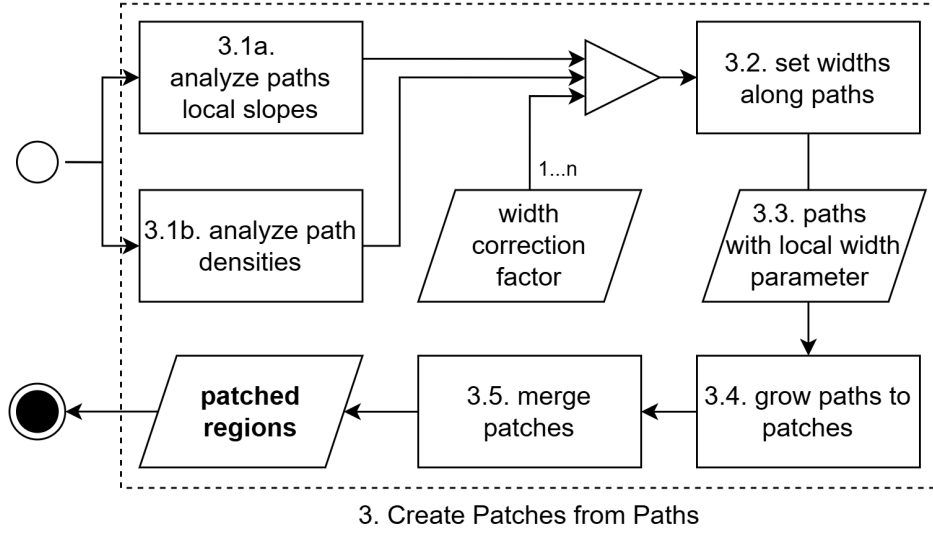


Figure 4.7: Function structure for patch optimizer module 3, *Create Patches from Paths*. The paths determined before get discretized into patched regions based upon path properties width, density and a correction factor.

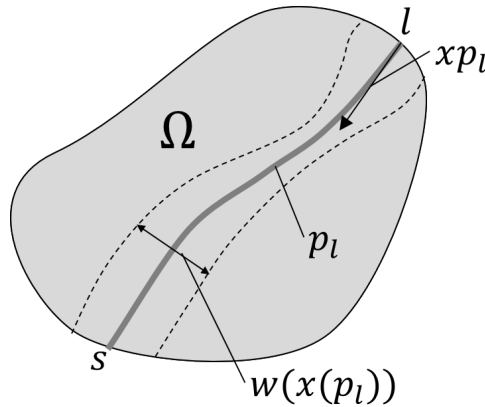


Figure 4.8: Width $w(x(p_l))$ being a function of location x on a path p_l . This allows the translation of a path into a patch, with width perpendicular to path direction.

Secondly, the path finder searched stiff routes with relative stiffness (gradient ∇) evaluated. Still, this stiffness will not be consistent throughout a path. Some sections of it may be adding more to it than others. See figure 4.9b, U^* may not decrease linearly along any path p_l , indicating varying stiffness. To account for this, patch width is corrected

by local gradient. Here, unpenalized values and the actual gradient ∇ , see (4.24), can be applied. Because also areas of high positive gradient (i.e. areas of U^* increase) are expected to be stiff, absolute gradient is used:

$$\nabla_w(x(p_l)) = \left| \frac{\delta U_{p_l}^*(x(p_l))}{\delta x_{p_l}(x(p_l))} \right|. \quad (4.35)$$

Thirdly, as numerous paths $p_l \in P_L$ are being evaluated, they could overlap in some places or generally be closer to each other than elsewhere. If this were not accounted for, those areas, where a lot of load flowing through is to be expected, would be undervalued. A metric of path density is thus introduced by

$$\varrho_w(\Omega) = n(P_l \in \partial\Omega) \quad (4.36)$$

with n being the number of paths going through some part of Ω . Figure 4.9c shows how path density may vary along some section perpendicular to the paths. Finally, all combines into

$$w(x(p_l)) = k_w \nabla_w(x(p_l)) \varrho_w(x(p_l)). \quad (4.37)$$

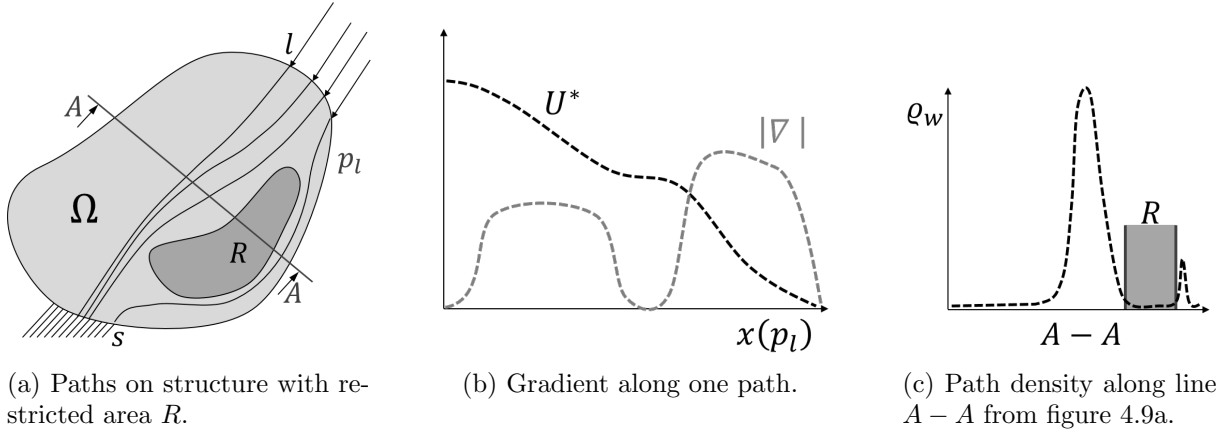


Figure 4.9: Qualitative description of influence factors on patch width: Gradient and path density. Together, these factors control patch width along paths.

Growth direction is perpendicular to path direction. As Ω may be curved, the width is to be evaluated not linear but as the distance over Ω , as figure 4.10a shows. In the discrete case, this is finding elements closest to a plane normal to direction vector $\vec{x}(p_l)$ which are neighbors of the respective element and evaluating their plane-projected distance. Note that in element selection, one may prohibit using restricted elements, leading to asymmetrical growth if bordering a restricted region. For each path, element selection leads to a set of patch elements denoted $E(p_l)$.

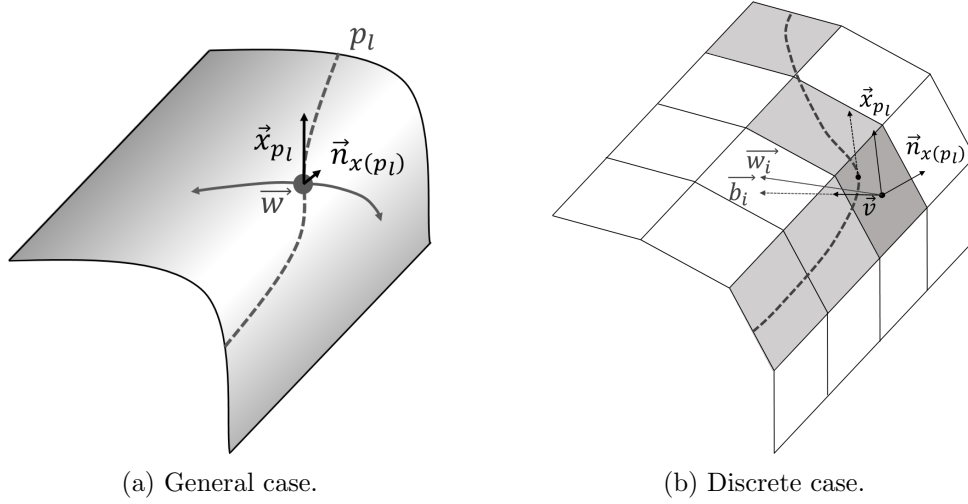


Figure 4.10: Orientation of grow direction for path width. Grow is perpendicular to the path direction. Curvature of the surface is accounted for by growing in a intersecting plane perpendicular to path direction and having the patch width be the arc length.

As the last step within this sub-function, all paths are merged, creating one (not necessarily connected) region which is

$$E_p = E(p_l \in P_l). \quad (4.38)$$

Overlapping path elements are only added once. This is acceptable, as path density has been accounted for earlier in (4.36).

4.2.4 Tune Thickness

With patches now being two-dimensional with respect to Ω , next, one can add patch thickness and tune base thickness accordingly. Total patch region area A_p , general area A_Ω and respective unpatched area $A_b = A_\Omega - A_p$ with patch thickness t_p , base thickness t_b and patched region thickness t_{bp} are introduced. From this, geometry volume and thus mass can be determined and optimized. Figure 4.11 shows the function structure.

First, a design parameter g is to be selected, which in the following will be maximum equivalent stress, while being interchangeable with other properties such as maximum deflection, strain, strain energy, principal stress or some specific node deflection as a metric for component stiffness under specific load. Starting point is the base's initial thickness $t_{b0} = t_0$ of the unpatched state for which finite element analysis leads to baseline performance g_0 . Next, select desired base part thickness t_{b1} which is either fixed or within a range

$$t_b = \{1 \dots m\} \frac{a - b}{1 - m} + \frac{b - ma}{1 - m} \quad (4.39)$$

with resolution m , lower limit a and upper limit b being $0 < a < b \leq t_{b0}$.

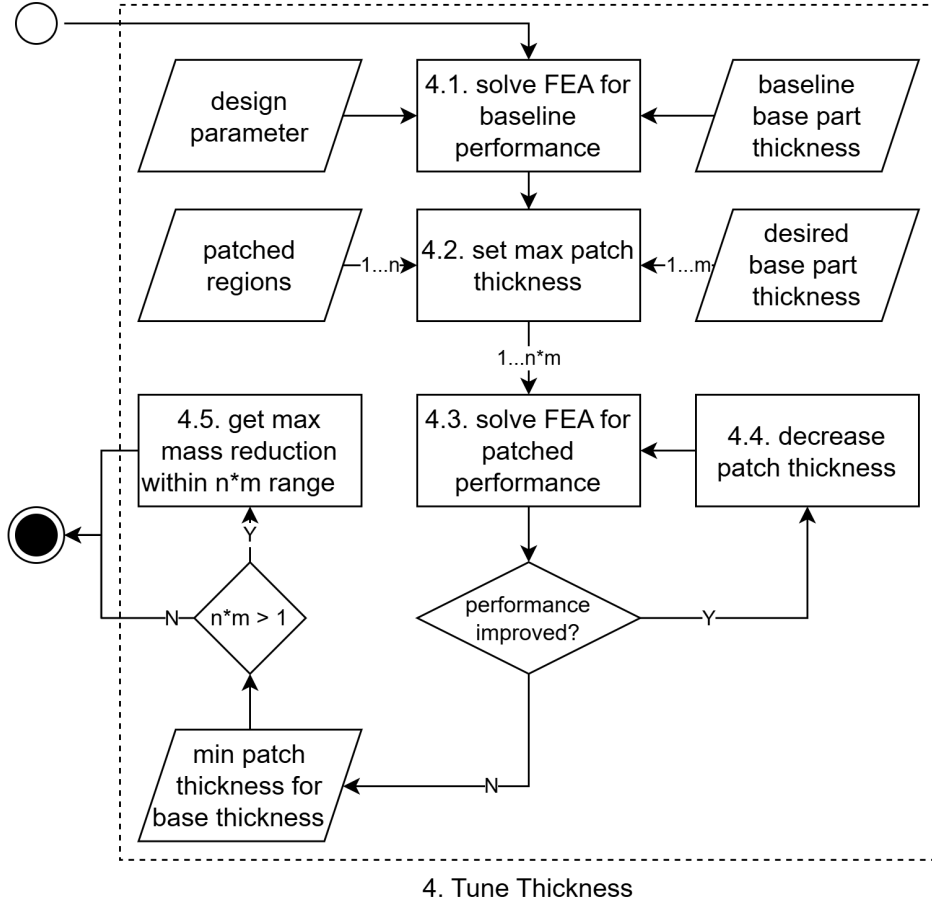


Figure 4.11: Function structure for patch optimizer module 4, Tune Thickness. From the patches made in module 3, thicknesses get processed to achieve minimal overall mass but maintaining component performance, i.e. maximum stress, to the initial design for the given load case, the baseline performance. Thickness is iteratively decreased and evaluated until performance matches the baseline performance.

From initial volume $V_0 = t_0 A_\Omega$ a maximum patch thickness results by checking leftover volume when decreasing base thickness to t_b :

$$t_p = \frac{V_0 - t_{b1} A_\Omega}{A_p}. \quad (4.40)$$

Finite element analysis including patches then leads to performance g which is generally expected to be better than initially, so $g < g_0$. This is, for example, maximum stress. From here, one can iteratively decrease patch thickness until $g' \approx g_0$. More specifically, due to the approximate linear dependency of g on t in linear elastic analysis where there is plane stress, as typically being the case in shell geometry, one can have patch thickness decrease proportional to performance ratio Q_g , converging quickly without risking instability. This is done by

$$Q_g = \frac{g}{g_0} \quad (4.41)$$

$$t'_p = t_p Q_g. \quad (4.42)$$

Note that since only t_p and not also t_b is updated, relative volume decrease actually is lower than Q_g , thus approximation is on the safe side and monotonous convergence to g_0 expected. At convergence, defined by some threshold, minimal patch thickness t_{p1} is found. From this, an overall volume reduction as ratio

$$Q_V = \frac{V_1}{V_0} \quad (4.43)$$

can be evaluated. This ratio will be highly dependent on selected base material thickness and patch dimensioning. Earlier, the possibilities of using ranges for width factor k_w , see (4.34), and base t_b , see (4.39), were introduced. Any combination of those may be the best. Repeating the process of patch creation, see figure 4.7, and subsequent thickness tuning leads to $m \times n$ data points as visualized in figure 4.12. By surface interpolation one has then found the minimum within k_w and t_b ranges also for intermediate values which is $Q_{V,min}$ at optimal $k_{w,opt}$ and $t_{b,opt}$.

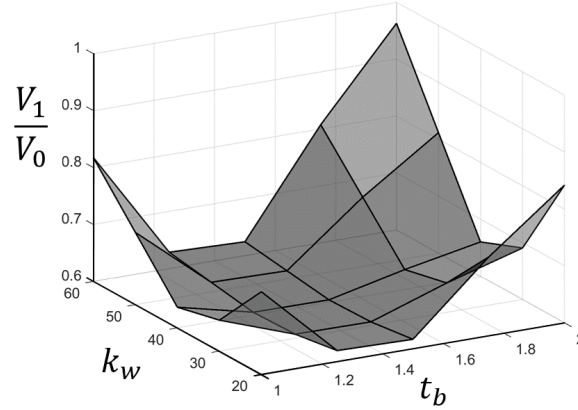


Figure 4.12: Plot of resulting optimal volume ratio $Q_V = V_1/V_0$ over base thickness t_b and width factor k_w . the optimal combination of parameters can be observed by interpolation of the response surface and determining the minimum.

Chapter 5

Software Implementation

In the following, the implementation of the aforementioned patch optimizer method as a software is presented with respect to platforms used, functional elements, and interfaces. Except for standard in-built functions and some segments within U^* computation which are referenced accordingly, all was developed and coded from scratch within this project. The software follows the logic shown earlier closely, adding some elements for pre- and postprocessing and interfacing. The software is coded in *MATLAB R2021a* interfacing with *ANSYS Mechanical APDL 2021 R1* for solving finite element equations. Also, some manual preparatory steps are to be performed, which were done in *ANSYS Workbench 2021 R2*.

MATLAB is a commercial software for performing numerical computation where the focus lies within matrix operations [Har20]. While being code-based, it is not aimed at developing stand-alone software but for working from within the MATLAB environment. MATLAB uses a unique programming language. There are numerous functions implemented in MATLAB already or to be accessed through toolboxes such as algorithms, interfaces and data visualization functions. In this project, only core functionality without additional toolboxes was used.

ANSYS is a commercial software package for performing finite element analysis [Rad22]. Its core component is the *ANSYS Mechanical Solver*, which is capable of solving large-scale finite element equations. To communicate with it, there is a script language called *APDL*, short for *Ansys Parametric Design Language*, based upon the FORTRAN language [MAA⁺24]. In its early versions, APDL was the only way to interact with ANSYS by entering commands in the console of the *Mechanical APDL (MAPDL)* interface. Later, a more sophisticated graphical user interface (GUI), *ANSYS Workbench*, was introduced. It provides tools for pre- and post-processing as well as additional applications, such as *DesignModeler* for geometry preparation. Nevertheless, the underlying workflow has remained unchanged: the Workbench GUI still generates an APDL input file, named *ds.dat*, which is passed to the *ANSYS Mechanical Solver*. After execution, the solver returns the results to the Workbench GUI. Modern versions of ANSYS also continue to include the

MAPDL interface. This enables straightforward automation by generating APDL input files externally and executing them through MAPDL, as it can be launched directly from the operating system without opening the MAPDL GUI.

5.1 Software Architecture

Figure 5.1 shows the software architecture, highlighting components newly developed within this project. At user level, geometry and data preparation is performed by the user within CAD and a meshing software. In theory, meshing could be performed in any software of choice, although mesh data export format needs to comply with ANSYS convention which will be described in section 5.2.1. From there, geometry, constraints and some analysis settings are entered in a text-based interface within MATLAB. Upon running the MATLAB code, the optimizer proceeds automatically while updating a geometry plot, visualizing intermediate results such as U^* data, paths and patches until finally, result data is presented. Some export functionality of the resulting patches mesh is given as output to subsequent CAD geometry implementation by a specifically developed export module.

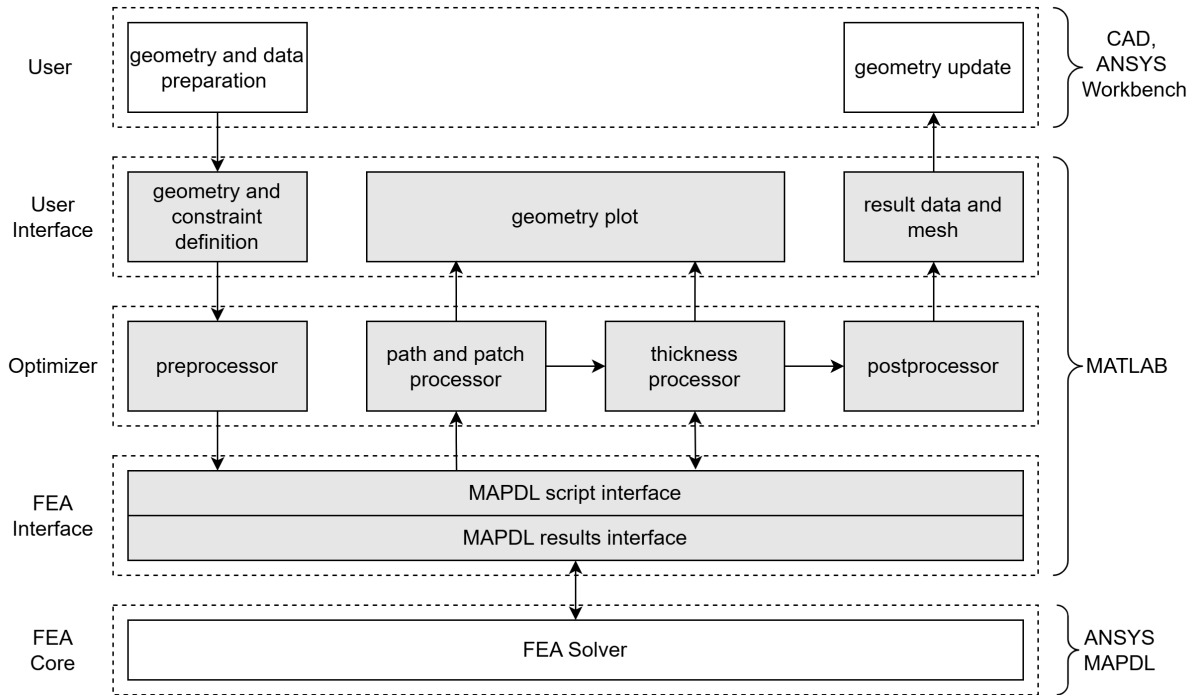


Figure 5.1: Patch optimizer software architecture. Highlighted in grey is developed within this project. Left: Function group. Middle: Functional blocks. Right: Environment used.

Within optimizer level, first, data is preprocessed where geometry is analyzed and information for FEA prepared, such as U^* data point selection. From FEA results, path and patch optimization is performed within MATLAB environment. Thickness optimiza-

tion is performed again with MAPDL interfacing. Finally, results are postprocessed and given back to the user.

On FEA interface level, within MATLAB environment, tasks are translated into ANSYS APDL code format creating MAPDL input files such as geometry data and operation commands. Also from within MATLAB, the path optimizer then calls MAPDL for working through the input file which includes commands for solution and desired output creation. The outputs are then imported with a results interface and fed back into optimizer.

The FEA core itself is not part of this project. ANSYS uses very efficient ways to solve finite element equations, so using an interface and external solving is fast, making incorporating FEA within MATLAB environment not worthwhile at this stage of development.

5.2 Function Structure

Figure 5.2 shows the overall function structure, for the most resembling the method structure introduced before in figure 4.2. From an initial geometry and load case, the import data is manually prepared for preprocessing. Then, U^* FEA is performed, paths found and patches created. Finally, thicknesses are adjusted, the optimum found and data exported, leading to an optimized geometry. In the following, the function structure will be described more in detail.

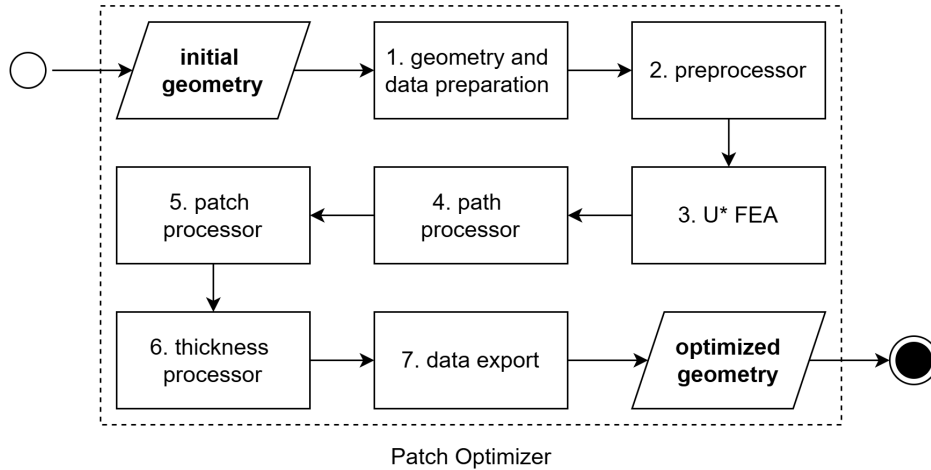


Figure 5.2: Function structure of the patch optimizer software. It follows the logic from the general method function structure, see figure 4.2.

5.2.1 Geometry and Constraint Definition

On user level, data is to be prepared as shown in figure 5.3. The main task here is to create a set of input data, as the right column shows, for the patch optimizer to start. The required data formats will be explained in section 5.3.

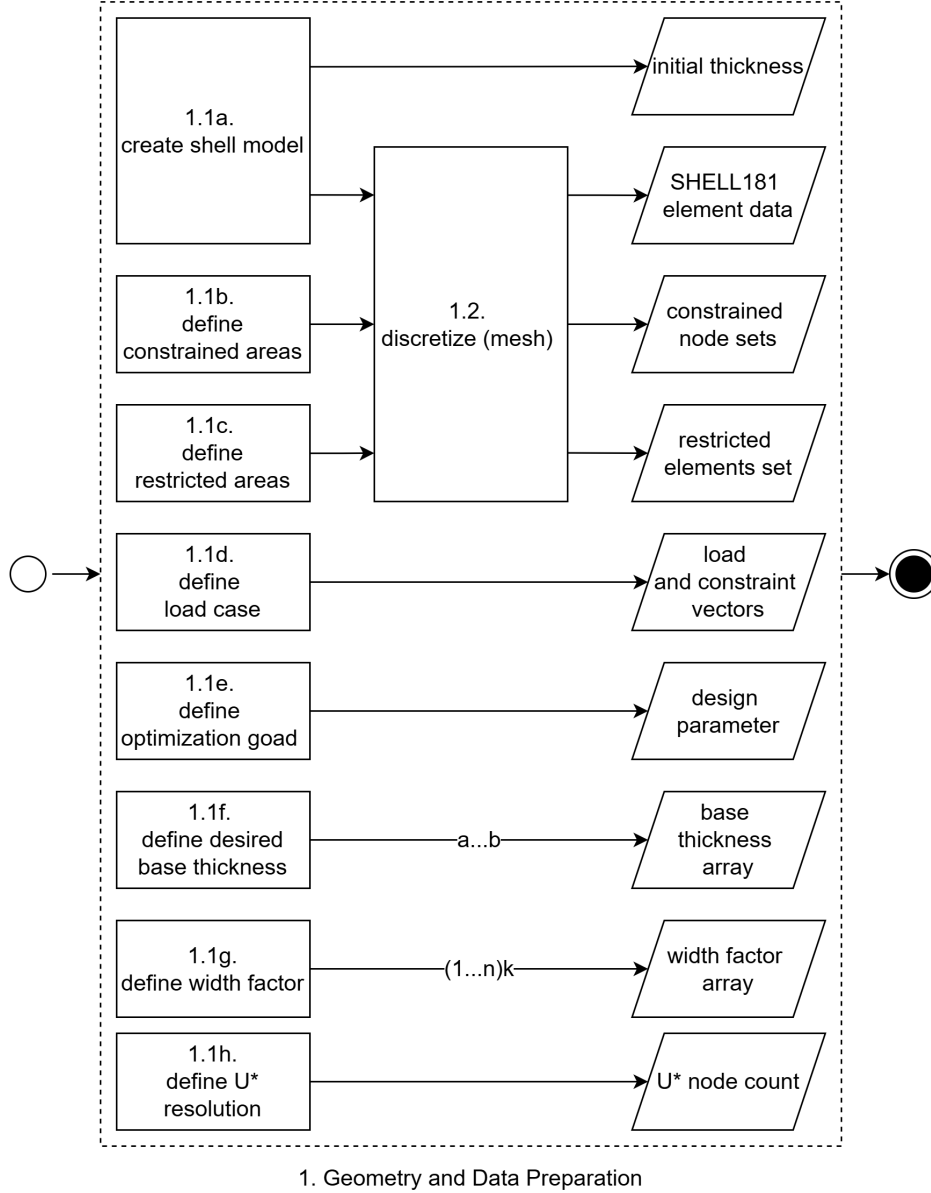


Figure 5.3: Function structure patch optimizer software module 1, Geometry and Data Preparation. Here, the geometry is meshed and all user input data is transferred into the system.

Either from existing solid geometry or from scratch, a shell model is to be created in CAD. It should form a single connected surface with uniform thickness. Material thickness should point in the direction of where the patch should lay. On hollow components, this would be selecting the inner surface with material growth outwards. Thickness is provided as a separate parameter. The geometry is discretized in a meshing software, conveniently this could be done in ANSYS Workbench. The output of it should be quadrilateral 4-node

elements to keep node count low for computational efficiency. This is achieved by ANSYS SHELL181 data format, which is used here [AnE21]. Any other mesher that can output 4-node quadrilateral shell elements would work as well.

For the given model, there exist constraints, i.e. geometric or mechanical boundary conditions. They may be applied to faces, edges or vertices on the CAD geometry or directly to nodes and elements in the meshed geometry. The optimizer requires node sets for each constraint, as well as an element set for restricted areas. An ergonomic way is to transform a CAD selection to mesh items in by using ANSYS Mechanical for selection and export as node and element lists, accordingly.

Next, one needs to define the load case. Each constrained node set is controlled by an application node with three coordinates, having 6 degrees of freedom (DoF) and 6 load vector components. Also, the connection to the application node can be set as either rigid in itself or flexible, which is ANSYS *CERIG* or *RBE3*, see [AnC21].

Additionally to the base case definition, some information is to be given for optimization. First is the optimization goal being either maximum equivalent or principal stress, strain energy or deflection, given as design parameter *SEQV*, *PRIN1*, *SENE* or *MDEF*. Then, a desired base thickness is to be chosen. As this may not be fixed, as described before, a minimum / maximum thickness and a number of increments can be given instead. The same is valid for the patch width factor, describing the average patch width (in mm) the software tries to find if there are no other restrictions such as restricted areas or an edge. Either a fixed factor or a range can be given. Finally, the desired resolution of U^* calculation must be set. U^* FEA takes up a considerable amount of the overall computation time and scales linearly with the number of calculated nodes. For compact components, a range of 200 - 500 nodes leads to sufficiently accurate data for interpolation. Larger, complex components may require up to 1500 nodes.

To summarize, the starting geometry, load data and some optimization parameters must be given. These are all manual inputs required for the patch optimizer to start.

5.2.2 Preprocessor

The preprocessor, whose function structure is shown in figure 5.4, imports the user-generated data and prepares the first computation, which is U^* .

Firstly, all elements are imported from the element dataset. It includes element nodes and their respective locations. Connectivity can be derived by shared nodes - for some element, the software searches all others for common node IDs to identify this. As element connectivity will be used frequently in the optimizer, it is useful to increase efficiency for this search. Thus, the first function block after import is finding each element's neighbors and storing all neighbor IDs in an element-specific cell array. Next, U^* nodes are selected based upon the desired node count specified by the user. The method introduced in

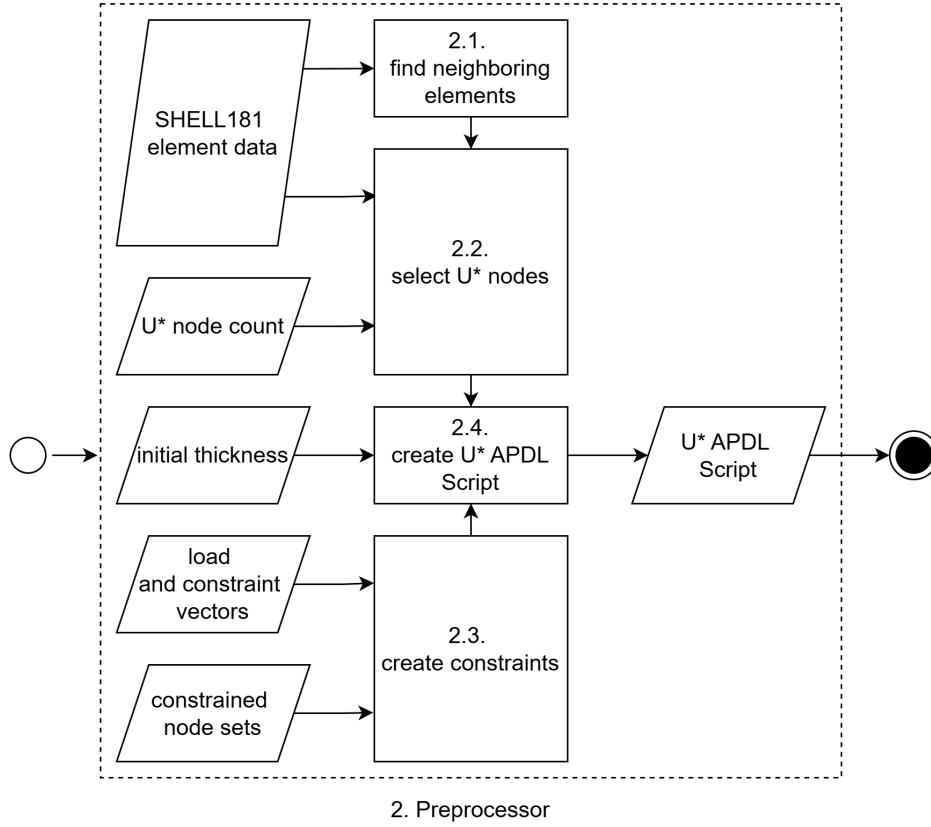


Figure 5.4: Function structure patch optimizer software module 2, Preprocessor. Here, from the input geometry, the FEA-Model for U^* -computation is developed. This includes mapping of neighboring elements, grid-based computation node selection and constraint creation.

section 4.2.1 is used iteratively until the found nodes are close to the desired count by some tolerance, which is chosen as 5 % for a maximum of 8 iterations, otherwise accepting the found amount.

Next, load and constraint data is used to create constraints. These, together with the element data, thickness and U^* nodes are used to create the U^* APDL input script. One may notice that material data has not been taken into account yet. As FEA is done only linear elastic, generic steel material properties with elastic modulus $ex = 2e11$ and Poisson ratio $prxy = 0.3$ are used as standard. If other metals or non-metals may be used, it is possible to change those values accordingly. The interface layer between base and patch, when analyzing the patched geometry later on in the process, is not accounted for as it is modeled monolithic. This is valid for bonded patches as described earlier in sections 2.1.3 and 3.1.

5.2.3 U^* FEA

With the APDL input file / script created, MAPDL can be called for performing the finite element analysis. The script describes the steps to perform. Part of the default APDL code-blocks are taken from [JM22], which are the solve set-up and output generation.

These blocks do not change between different computations. The process within MAPDL can be divided into preprocessor, solver for initial state, solver for inspection loads and postprocessor as figure 5.5 suggests.

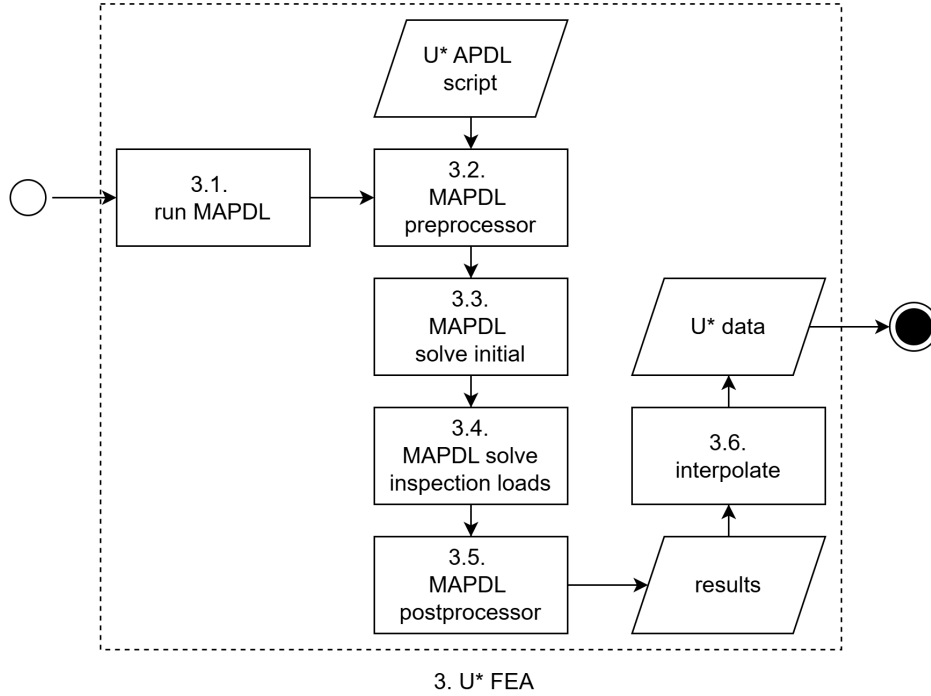


Figure 5.5: Function structure patch optimizer software module 3, U^* FEA. This part of the process runs mainly in MAPDL, until the point where results for the computation nodes are finished. They get re-imported into MATLAB and there interpolated to all elements.

Initially, MAPDL imports the elements and nodes. Constraints are created as defined. Then, the design load case is solved once to acquire strain energy of the initial state. Afterwards, load nodes are fixed and inspection loads applied instead. For doing so, three load steps for each node are created, where inspection loads in coordinate directions are applied individually. After solving all load steps, data is exported back for MATLAB to be imported. Required information are the load application node's deflection under design load, node deflections under inspection load and reaction forces. After re-import of the results, interpolation is performed as described in 4.2.2. For each element, closest neighboring U^* nodes are searched and a distance-weighted average is taken to interpolate from the few U^* nodes onto the whole geometry. The results can be displayed with the 3D geometry plot function. Figure 5.6 shows an example with U^* nodes highlighted in red and interpolated results plotted onto the elements by a color map.

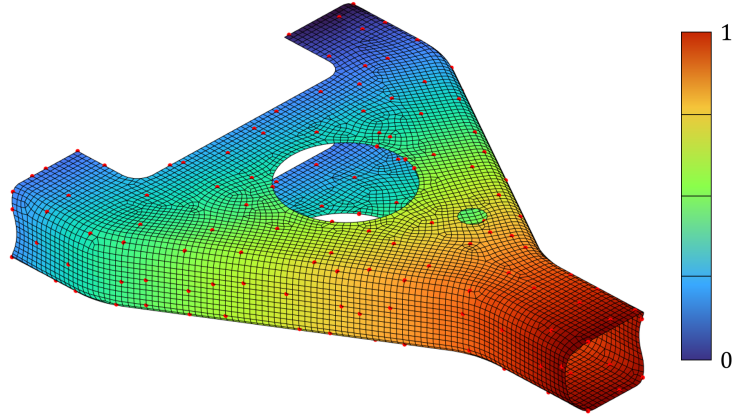


Figure 5.6: 198 U-Star-Nodes selected on an exemplary geometry. Calculated nodes are highlighted in red, while the subsequent interpolation of their results to all elements is mapped by the element color.

5.2.4 Path Processor

From U^* now, paths can be found. Figure 5.7 shows the general function structure and figure 5.8 shows Dijkstra's algorithm.

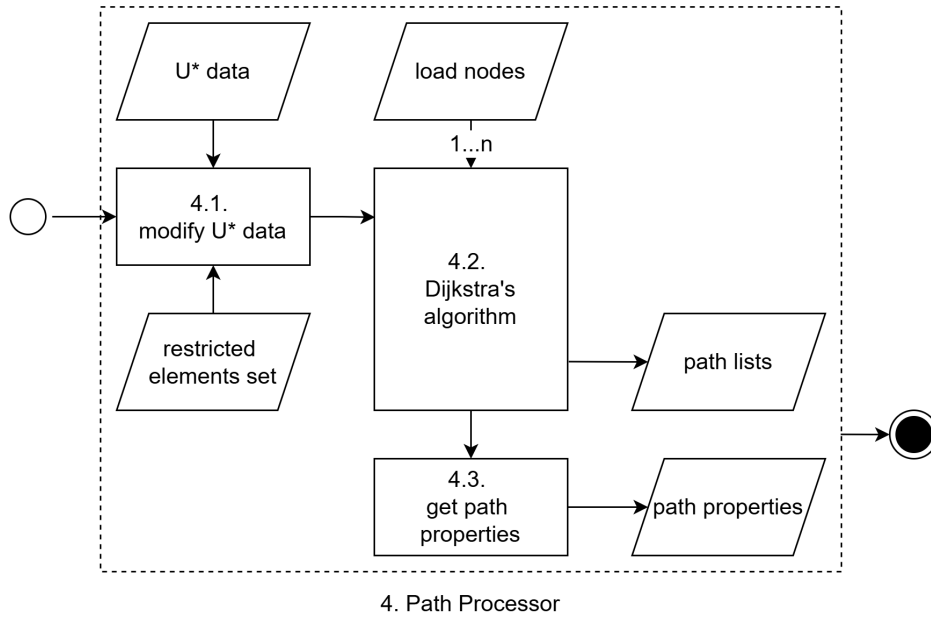


Figure 5.7: Function structure patch optimizer software module 4, Path Processor. Here, from U^* -data, restricted elements and boundary conditions, efficient main load paths can be determined.

First, U^* values are modified to account for restricted elements. If restricted elements shall not be used in any case, a preset penalty factor of 10^9 is used. Otherwise, a moderate value between 1 and 10 is suggested with a penalty of 10 decreasing the favorability of a restricted path by around 85 % as follows from equation (4.25).

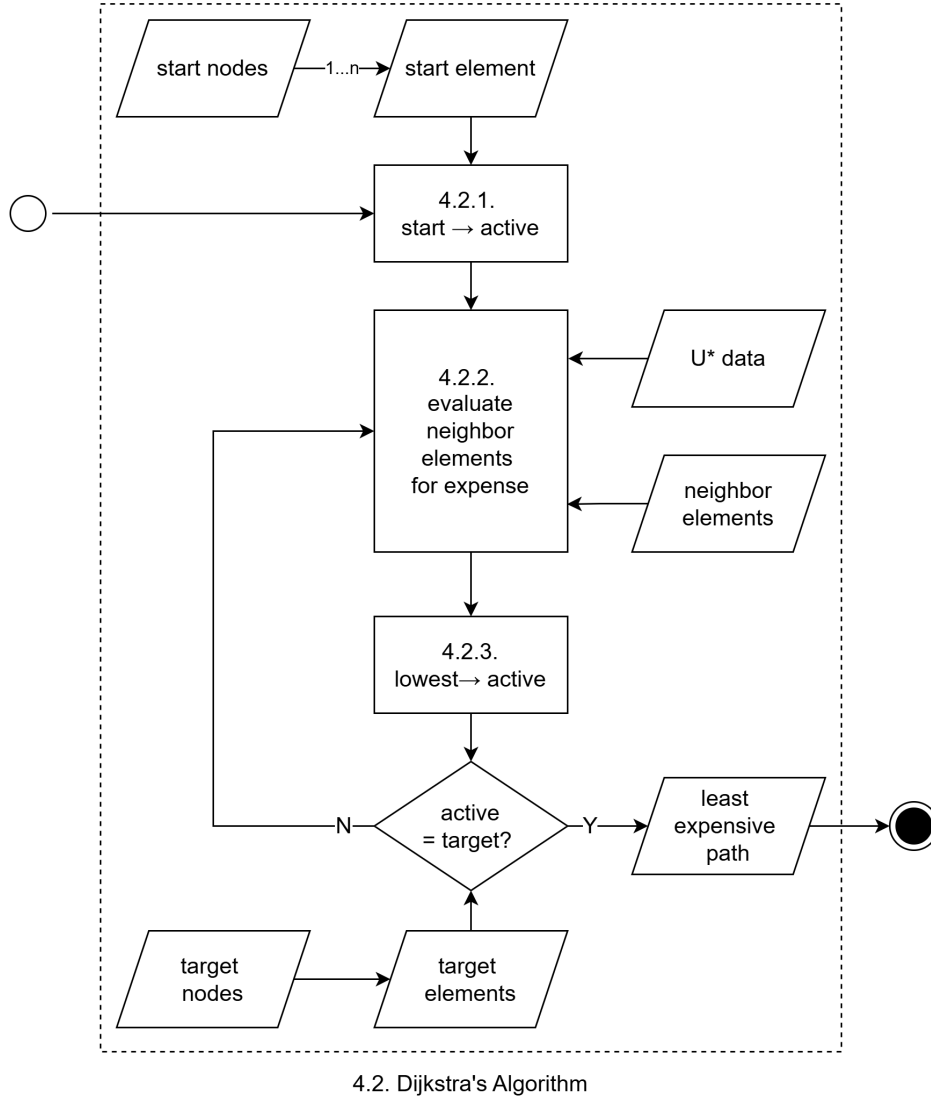


Figure 5.8: Function structure patch optimizer software submodule 4.2, Dijkstra's Algorithm. Beginning from start elements, expense to reach neighboring elements is evaluated, until the least expensive path to a target is found.

Then, for all elements connected to load nodes, Dijkstra's algorithm is performed. Paths are searched not between nodes but elements, as U^* data is interpolated to elements and restricted areas are defined by elements as well. From a start element, neighboring elements are evaluated as described in equations (4.22) to (4.28). The active element is updated accordingly. If it is one of the target elements, the iteration aborts as the shortest path is found. From here, path properties are evaluated and stored. With paths found for all load elements, this section is complete.

5.2.5 Patch Processor

In the patch processor, first, width factors are evaluated. Then, patches are grown from the paths and finally merged. See figure 5.9 for the structure.

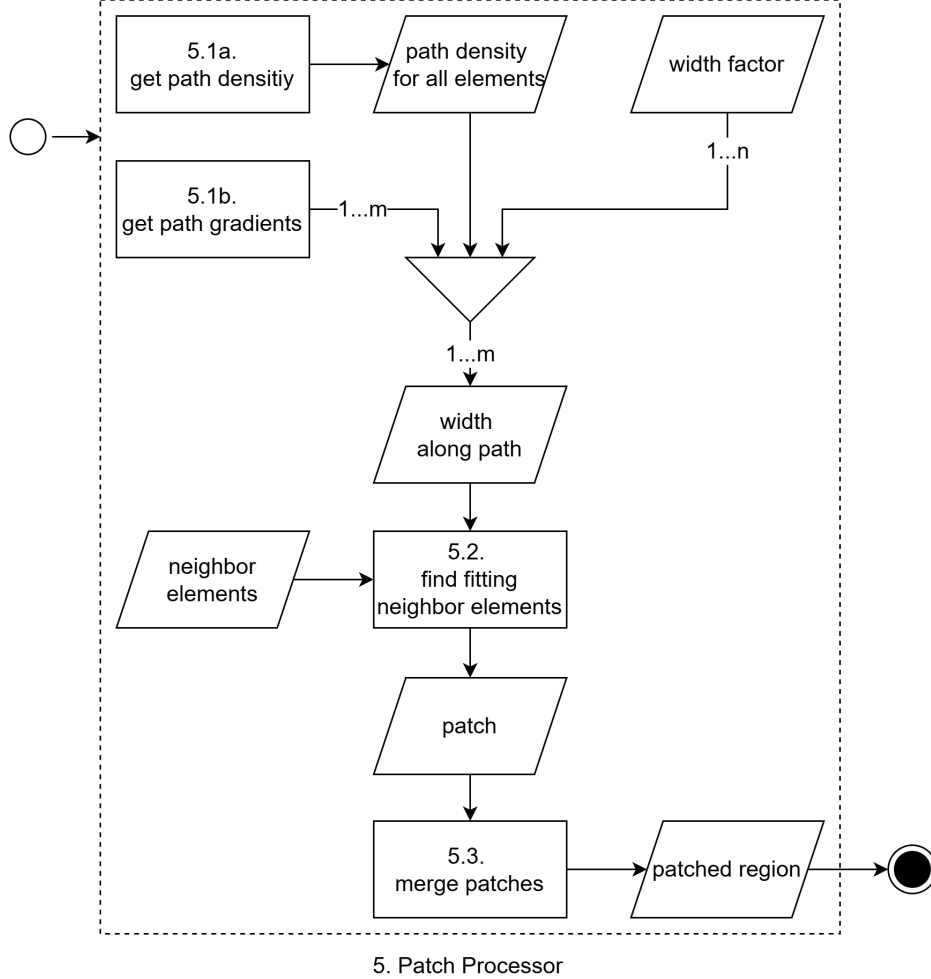


Figure 5.9: Function structure patch optimizer software module 5, Patch Processor. Paths get analyzed w.r.t density and gradient to determine width. With that, paths are grown to become patches, and merged.

For acquiring path density, number of path crossings and paths passing by in close proximity are counted with diminishing effect as distance to the evaluated element increases. To keep the code efficient, only three steps are evaluated for each element: Unique paths on the element itself, in direct neighborhood and on the next set of neighbors of those. Counting is done in a weighted ratio of 1 : 0.5 : 0.25, respectively.

Path slope is evaluated as in equation (4.35). With this, widths along paths can be created for all paths and all width factors, getting a desired width for each path element. To now get to patches, fitting neighbor elements need to be found. As width should grow perpendicular to the path, first, one has to determine path direction. Due to the paths leading through discrete elements, they generally are not very smooth. This would

occasionally lead to unreasonable direction vectors. To counteract, the path is smoothed first which is done by Gaussian smoothing over 5 elements. Figure 5.10 shows the effect of this.

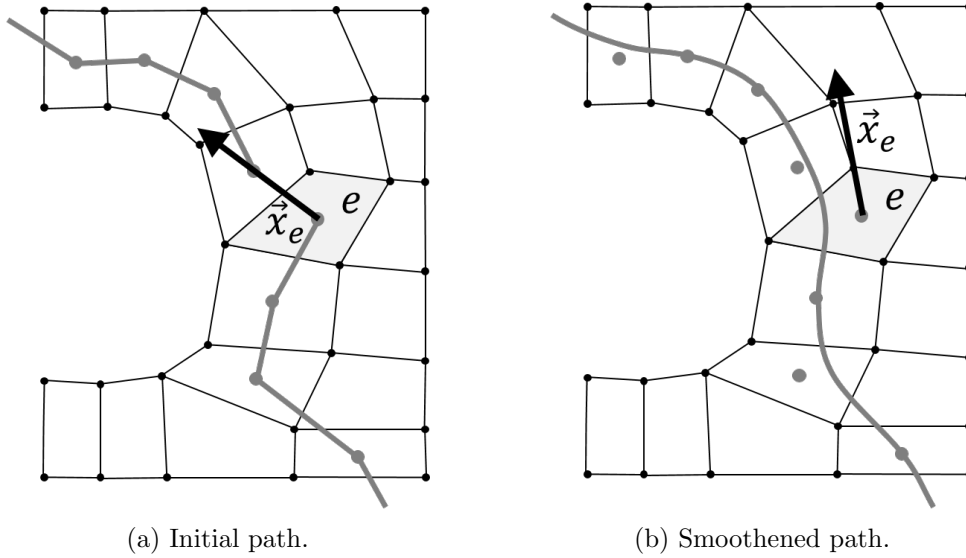


Figure 5.10: Smoothing of a path leading across discrete elements centers. The smoothing leads to less jitter in the path direction and decreases mesh dependency.

For each element, a plane through the center and perpendicular to the direction vector can now be defined. Neighboring elements are evaluated in terms of proximity to this plane, choosing the nearest. Distance is the projection of element center distance onto the plane. The chosen elements neighbors are then used to grow until desired width is reached. Finally, all paths are merged to create the patched region by adding up all unique patch elements.

5.2.6 Thickness Processor

To now get mass reduction potential from the patches, minimal thickness while maintaining performance needs evaluation. Figure 5.11 shows the thickness function structure, while figures 5.12 to 5.14 show its subfunctions.

First, initial performance for basic design without patches is evaluated by FEA. Then, for some desired base thickness and patch geometry, patch thickness is determined by leftover volume at decreased base thickness. Its performance is compared to the base case. If, as expected, performance is better, the patch thickness is decreased until performance is equal to the initial state. Result is the optimized geometry with decreased overall volume and mass, respectively. In the case of multiple base thicknesses and patch widths, this process is repeated for every combination leading to a matrix of optimizations. Cubic interpolation of the resulting 3D-surface then yields the approximate optimum within the corresponding ranges as well as the respective geometry.

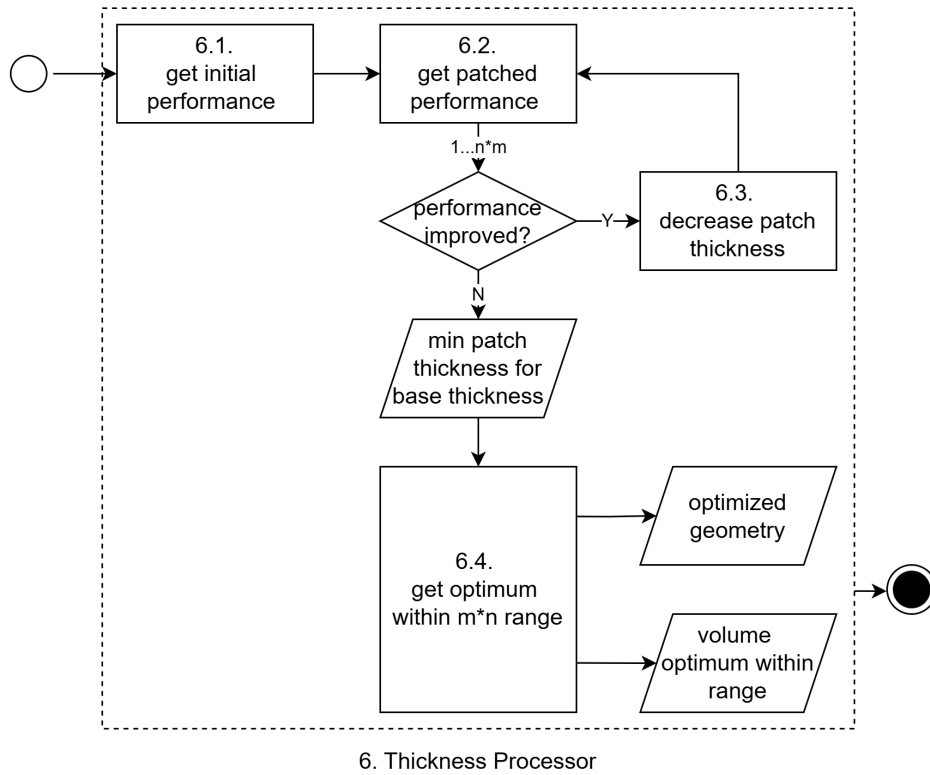


Figure 5.11: Function structure patch optimizer software module 6, Thickness Processor. Performance of patched and unpatched state is compared and minimal required patch thicknesses evaluated.

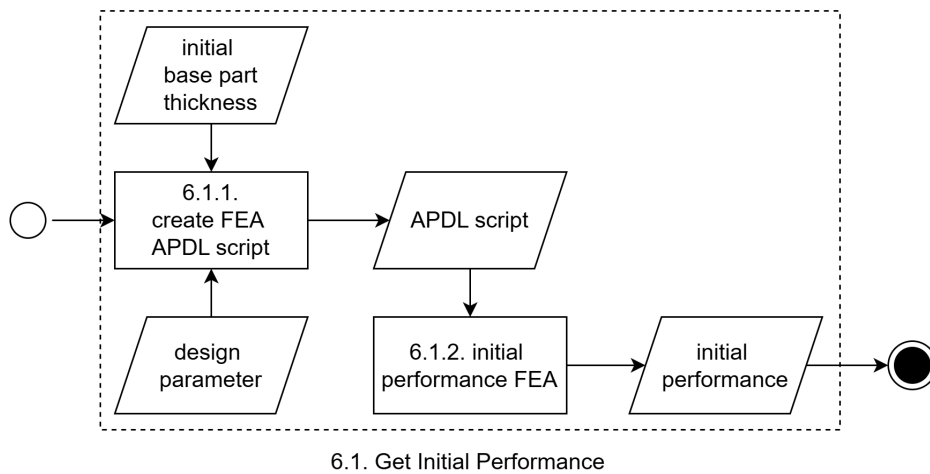


Figure 5.12: Function structure patch optimizer software submodule 6.1, Get Initial Performance. Performance for the initial, unpatched state, is evaluated by finite element analysis.

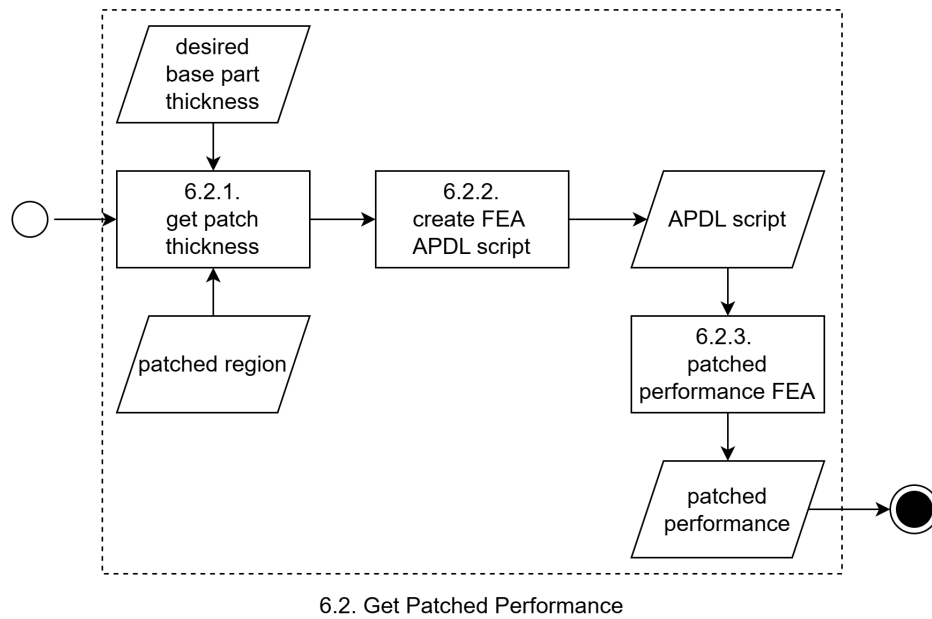


Figure 5.13: Function structure patch optimizer software submodule 6.2, Get Patched Performance. Performance for the patched state is evaluated by finite element analysis.

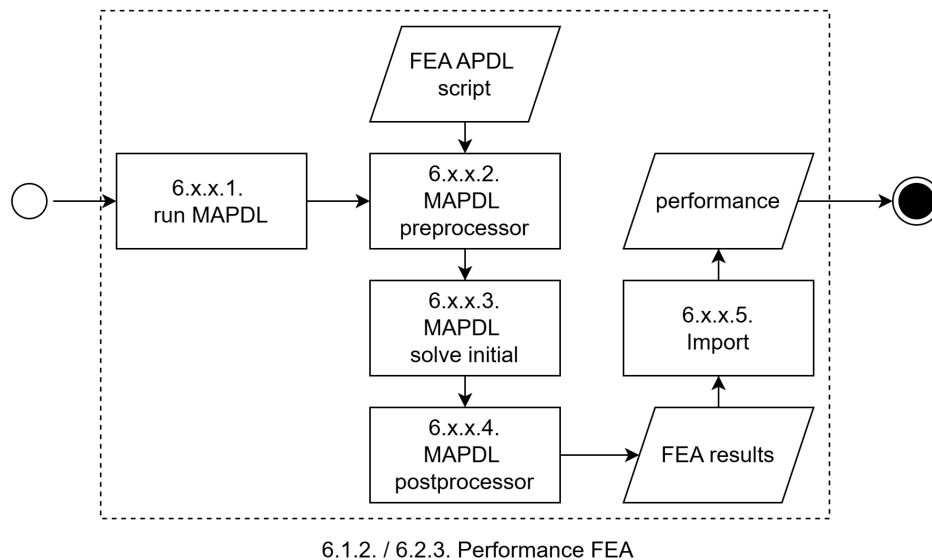


Figure 5.14: Function structure patch optimizer software submodules 6.1.2 / 6.2.3, Performance FEA. This process is the same for the unpatched (6.1) and patched state (6.2). Finite element analysis is performed in MAPDL and the results reimported.

5.2.7 Data Export

Finally, data is to be exported by a specifically developed export module as figure 5.15 shows. The optimized geometry is transformed into a volumetric mesh that shows the patches at their thickness. From there, a CAD model can be derived. Its data format will be described in the next section 5.3. Also, the optimization data is exported, which is the performance, volume ratio and thicknesses for base and patch.

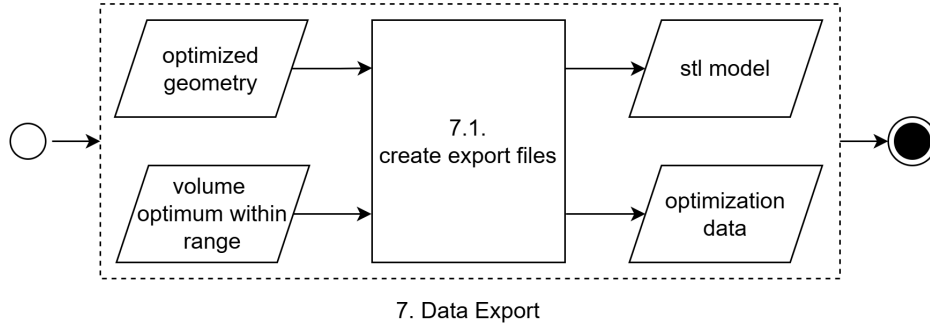


Figure 5.15: Function structure patch optimizer software module 7, Data Export. The optimized geometry is exported as stl model and achieved volume decrease is given.

5.3 Interfaces

Within the software, there are several interfaces as figure 5.1 suggested. There is user interaction at the input, during calculations by showing a model update and at the output. Within, MATLAB and ANSYS communicate through ADPL script language. In the following, structure and data formats will be explained.

5.3.1 Input Interface

As shown in figure 5.3, the user has to give several information. Many of it are to be input directly within the MATLAB GUI. Table 5.1 shows all parameters and their formats, see FORTRAN 77 format code [Ora10], tables 5.2 and 5.3 show node and element data list format for SHELL181 elements. Sets must be given as integer lists in format 8i10. When doing manual preprocessing with ANSYS Mechanical, all this can be found correctly formatted within the *ds.dat* APDL solver script that is created. The user, however, needs to extract those lists and save them separately as this process step is not automated yet at the state of submission.

Table 5.1: Patch optimizer input data formats. When following the suggested process of using ANSYS Mechanical for data preparation, these formats are used automatically. If other environments for data preparation may be used, the formats must match accordingly.

parameter	type	format
initial thickness in mm	number	double
element data	list	19i9
node data	list	1i9,3e20
constrained node sets	list	8i10
restricted elements set	list	8i10
load vectors in N/Nm	array	[fx fy fz mx my mz]
load point	array	[x,y,z]
constraint vectors in mm/rad	array	[dx dy dz rotx roty rotz]
constraint point	array	[x,y,z]
optimization method	string	{SEQV,PRIN1,SENE,MDEF}
base thickness array in mm	array	[lower,upper,res]
width factor array	array	[res,scale]
U^* node count	number	integer

Table 5.2: Patch optimizer node data format 1i9,3e20. As stated before, this comes standard in ANSYS Mechanical but must be maintained when using other software for manual preparation.

field	1	2	3	4
content	ID	X	Y	Z

Table 5.3: Patch optimizer element data format 19i9. As stated before, this comes standard in ANSYS Mechanical but must be maintained when using other software for manual preparation.

field	1	2	3	4	5 - 10	11	12	13	14	15	16 - 19
content	MAT	TYPE	REAL	SECT	–	ID	N1	N2	N3	N4	–

5.3.2 Graphical Interface

While computing, the software can display intermediate results. MATLABs built-in 3D plot function can show any set of surfaces in 3D-space and colorize them based upon a property parameter. For each element, a six-sided cube with a uniform color parameter is created. That way, thickness and some result data such as U^* -value can be shown. Also, patched regions can be visualized with their respective thickness. This is incorporated in the `plotElems` function. Additional information such as points or vectors can be plotted within the `plotElemsU` function. For example, figure 7.13 shows U^* points and interpolated U^* data using `plotElemsU`. Also, intermediate result values and progress are displayed in the MATLAB command window.

5.3.3 FEA Interface

Within itself, the patch optimizer interfaces MATLAB with ANSYS, using the APDL as introduced earlier. The interface was developed from scratch within this project. APDL scripts are built up from blocks, which are created by custom MATLAB functions. Table 5.4 shows the structure and respective commands to create the elements. MAPDL is launched by MATLAB through *system* command executing an OS command launching MAPDL in batch mode reading from the specified ds.dat file. ANSYS outputs are created in a list containing all element (or, if chosen, nodal) results, ordered by ID without any additional fields.

Table 5.4: Structure of the ds.dat ANSYS APDL batch file and respective MATLAB functions

command block	content	function
header	analysis method	init2dat.m
material properties	element type, elastic modulus, poisson ratio	
section data	thickness and offset for each element	sec2dat.m
nodes	calling nodes + coordinates from separate node list	elem2dat.m
elements	calling elements from separate element list	
constraints	creating rigid or flexible constraints on nodes	concerig2dat.m / conrbe32dat.m
loads	creating nodal loads	
solve	calling the solve	solve2dat.m
result export	defining desired results and exporting them to file	results2dat.m
footer	finish the run	

5.3.4 Output Interface

After successful optimization, the resulting geometry is output for remodeling in CAD. The input mesh used a format specific to ANSYS, since the FEA is done in ANSYS as well. Here though, a neutral format can be used to give the user freedom of choice for CAD environment. The output creates an STL-File, which is the most common format for mesh files and can be read by most CAD programs. However, STL only supports triangular elements, so the quadrilaterals from SHELL181 are split alongside their diagonal. The STL is a text file with a header "solid", then a simple list of elements in the format shown in table 5.5 including the node coordinates and the elements normal vector and finally a footer "endsolid". Finally, optimization results are output as well, giving the volume decrease and respective thicknesses, finalizing the patch optimizer process.

Table 5.5: Structure of a single STL element block, format 1a20,3e20. This format is readable by most CAD software. If using some software that cannot process STL, the data may be manually processed based on this structure.

field	1	2	3	4
	"facet normal"	X	Y	Z
	"outer loop"			
	"vertex"	X	Y	Z
content	"vertex"	X	Y	Z
	"vertex"	X	Y	Z
	"endloop"			
	"endfacet"			

Chapter 6

Methods for Patch Bonding

So far, a method and software for optimizing sheet metal components by decreasing base part thickness and adding a second layer, a patch, were introduced. Adding the patch in the right position can strengthen the component by carrying forces more efficiently than with a uniform thickness design. However, from force application points to support points, the force transfer into the patch must be ensured in a safe and rigid manner – otherwise, the patch may fail or not enough stiffness is provided to take load from unpatched regions. As was discussed in section 3, adhesive bonding of steel plates shows potential to solve this as it exhibits good shear strength and due to the uniform connection behaves monolithic after a short transition zone for large bonding areas. Adhesive bonding of steel plates is thus further investigated. As there are numerous different adhesives available, first, a selection must be made. Main focus are static and fatigue strength within the scope of operation for automotive components, ease of application and low possible bond thickness combined with high shear modulus to get a stiff connection.

6.1 Adhesive Selection

As introduced earlier, for bonding steel plates, types of structural adhesive come into question which are chemically bonding due to them being generally much stronger than only physically bonding ones. Within the automotive industry, for structural applications, thermosetting epoxy-based 1K-adhesives find widespread use. They can be hardened at the same temperatures and duration as car body coating cures which is 180 °C for 30 minutes. These adhesives exhibit the highest shear strengths and their application can be conveniently incorporated into the automotive production process. Some common ones are *Dupont Betamate 1630* and *Henkel Loctite EA 9514*, which were selected for further analysis as they are one component, further easing application, have a high shear modulus and shear strength and some tolerance for surface oil contamination which may be present in automotive components. Both of them are fluid at room temperature and cure at the before-mentioned temperature. As an alternative, structural tapes were investigated

as well. Being rather new developments, they only start to get track in automotive application. One of them is *3M SAT1100S*, which is a non-woven fiber embedded into a 1K-epoxy adhesive, creating a tape of uniform thickness. Structural tapes like this have some definite advantages: easier application, uniform bond thickness and some initial stickiness. In the following, these three selected adhesives are investigated with respect to strength and usability.

6.2 Adhesive Testing

For evaluating suitability to the components within scope of this thesis, one must know the static strength, fatigue strength, stiffness and manufacturability of glued joints with the above-mentioned adhesives. There is some information available through the suppliers: Static data for shear strength, tensile strength and respective moduli, which are summarized in table 6.1. This data represents ideal samples at standard tests. For example, tensile strength is evaluated on a cast adhesive probe according to DIN EN ISO 527 [DIN19] and represents the adhesive’s maximal mechanical properties. Lap shear testing is performed to DIN EN 1465 [DIN09] which has defined mating sheets and application conditions. In a real use case, strength may be different due to application and mating component influences. Also, there is no fatigue data available through the suppliers. Hence, application was analyzed and static testing performed to give realistic properties for bonding steel patches. After a preselection on that basis, fatigue behavior was investigated.

Table 6.1: Supplier data for selected structural adhesives [DuP19] [Hen14] [3M20].

^a derived from tensile modulus at poisson ratio $\nu = 0.3$.

adhesive	shear strength [MPa]	shear modulus [MPa]	tensile strength [MPa]	tensile modulus [MPa]
Dupont Betamate 1630	≥ 30	577 ^a	29	1500
Henkel Loctite EA 9514	45	562 ^a	44	1460
3M SAT 1100S	25	1385 ^a	–	3600

6.2.1 Application

All adhesive testing was performed with 2.2 mm S500MC steel samples, as this material will also be used for prototypes, see next chapter 7. Surfaces were cleaned with acetone and then lubricated to 1 g/m² *Fuchs Anticorit RP 4107 S* which is within specification of surface lubrication for all tested adhesives. Bonding area was 12.5 mm × 25 mm in accordance with DIN EN 1465.

Liquid Adhesives

Liquid adhesives were manually applied to the bonding area at room temperature. To define bond thickness, the sheets were carefully stacked with spacers next to the bond to not interfere with the adhesive layer. Any spill-out was wiped off prior to curing at $180\text{ }^{\circ}\text{C}$ sample temperature for 30 minutes.

Tape Adhesive

For the 3M tape adhesive, patches in the size of bonding area were cut out. Here, bond thickness is not defined by spacers but the tape itself. Initial thickness is 0.4 mm and by clamping force while baking it can be reduced to up to 0.1 mm . Preliminary testing showed a target thickness of 0.2 mm to be desirable, as this allows to thoroughly fill the gap consistently and give some acceptable overflow on the edges while not exposing the non-woven fiber. That gap width was achievable by a downwards pressure of 7 kPa onto the specimen while curing, see figure 6.1. Curing was conducted at $180\text{ }^{\circ}\text{C}$ sample temperature for 30 minutes.

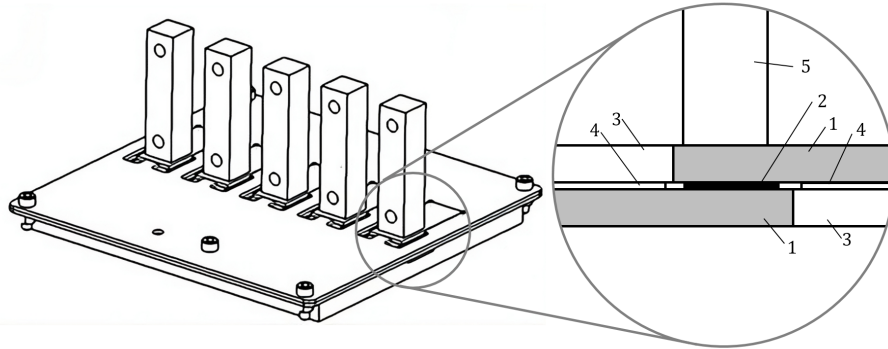


Figure 6.1: Curing fixture for adhesive tape. Specimens (1) and tape (2) are aligned in the fixture (3) with parallelism maintained by spacers (4). Clamping force is applied by weights (5).

6.2.2 Static Lap Shear Testing

On lap shear testing, Betamate adhesive achieved average shear strengths above 30 MPa for bonding thicknesses 0.1 mm , 0.2 mm and 0.3 mm . There is no readily apparent effect of thickness to shear strength. Loctite adhesive achieved strengths above 41 MPa , respectively. Although supplier specification was lowest, 3M tape adhesive performed with the highest shear strength of 44 MPa . Variance for Loctite is very high, for 3M tape it is lowest. Figure 6.2 summarizes the results. For Loctite, no consistent adhesion could be achieved, figure 6.3b shows partially adhesive failure. This is significantly better for Be-

tamate, see figure 6.3a. However Loctite still achieves higher shear strengths throughout. For 3M tape, no adhesive failure occurred, see figure 6.3c.

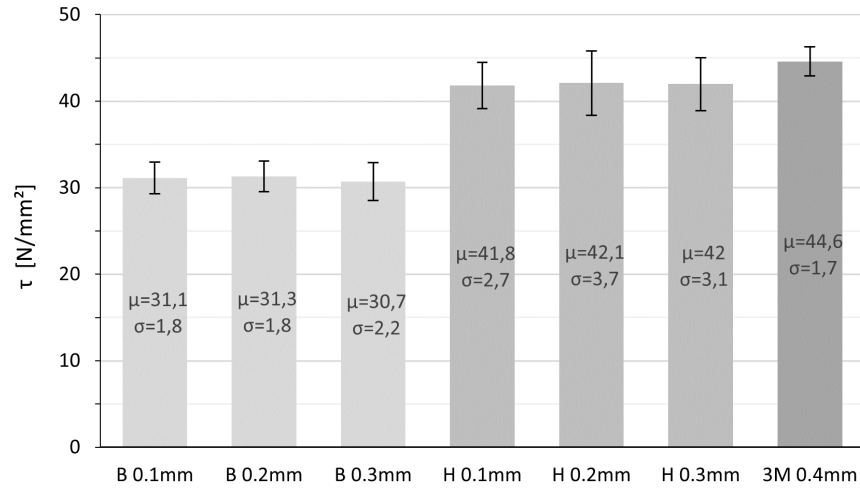


Figure 6.2: Achieved shear strength τ for adhesives (B) Dupont Betamate 1630, (H) Henkel Loctite EA 9514 for bond thicknesses 0.1, 0.2 and 0.3 mm and 3M SAT 1100S tape with 0.4 mm tape thickness. Mean μ and standard deviation σ for 5 samples each.

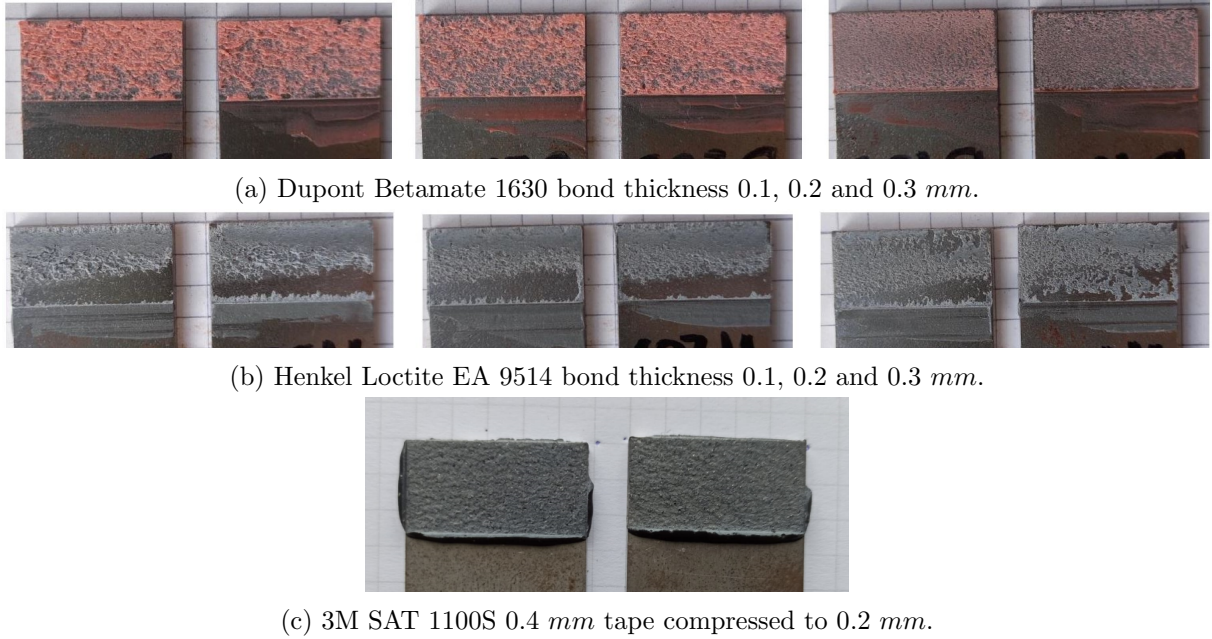


Figure 6.3: Specimen faces after lap shear test. Betamate 1630 (a) shows little adhesive failure indicated by the metal surface visible in the joint area. Loctite EA 9514 (b) shows adhesive failure of up to one third of the bonded area. 3M SAT 1100S tape shows no adhesive failure.

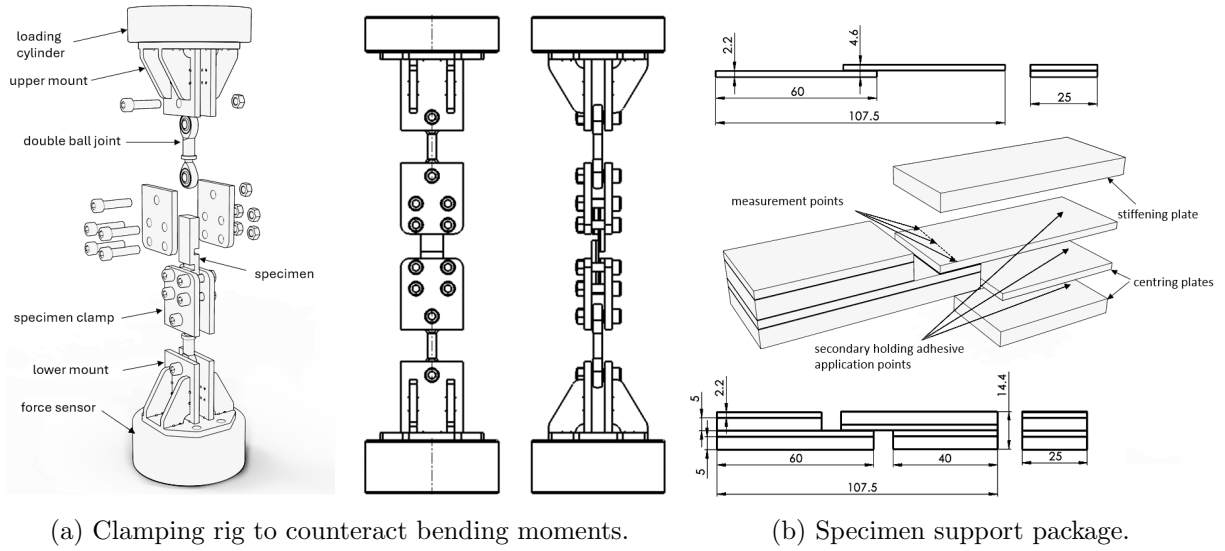


Figure 6.4: Setup for fatigue testing 3M SAT 1100S tape. The specimens are held in a way to not induce any undesired moments and achieve pure shear in the adhesive. This is achieved by universal joints above and below the specimens, and support plates on the specimen package to counteract peeling stress from specimen bending.

6.2.3 Fatigue Lap Shear Testing

Due to generally no promising results on the static test for Betamate 1630, fatigue testing was only conducted for the 3M adhesive tape and Loctite EA 9514. As fatigue strength is highly affected by stress concentration, a fixture was made to counteract any moments induced, see figure 6.4a. Also, specimens were preselected to only use ones with consistent bond thickness and supported on the backside to minimize peeling effect, see figure 6.4b. Fatigue testing was conducted on a universal hydraulic test bench shown in figure 6.5. As the clamping rig allows only for tensile loading, testing was done with a pulsating load inducing pulsating shear stress in the ratio $R_\tau = 0.1$. This keeps some safety margin to zero, preventing any shock loads due to test rig and fixture slackness. Often, loads are not pulsating, as tested here, but alternating. To get an equivalent alternating stress amplitude, one can translate the results by either Goodman or Gerber approach, as suggested for example in [ASMCdS22]. This translation and which approach is fitting to the adhesives is not further investigated here as a pulsating load case is tested only, also for the prototype build.

Table 6.2 summarizes the fatigue test results for 3M SAT 1100S tape adhesive. The respective SN-curve is shown in figure 6.6. At 10^6 cycles, expected shear stress amplitude is $\tau_{amp} = 7.73 \text{ MPa}$ for $R_\tau = 0.1$. This relates to an upper stress limit of

$$\tau_{max,50}(n = 10^6) = \frac{2\tau_{amp}}{1 - R_\tau} = 17.19 \text{ MPa} \quad (6.1)$$

which is 38.5% of static strength. The results show moderate variation on each load level.



Figure 6.5: Hydraulic universal test bench for performing the pulsating fatigue testing of adhesive specimens with the specimen clamping rig mounted to it. Force is exerted by the vertical hydraulic piston above the rig and measured by a pressure sensor below it.

Table 6.3 summarizes the fatigue test results for Henkel Loctite EA 9514 liquid adhesive at 0.1 mm thickness. The respective SN-curve is shown in figure 6.7. At 10^6 cycles, expected shear stress amplitude is $\tau_{amp} = 7.16 \text{ MPa}$ for $R_\tau = 0.1$. This relates to an upper stress limit of

$$\tau_{max,50}(n = 10^6) = 15.92 \text{ MPa} \quad (6.2)$$

which is 38 % of static strength. The results show a much higher variation than the tape, and worse predictability of the SN-Curve.

Table 6.2: Fatigue results for 3M SAT 1100S 0.4 mm tape.

Sample ID	Load Level	Number of Cycles [–]	N 50 %
2.1.3	Level 1: $\tau_{\max} = 24 \text{ MPa}$, $\tau_a = 10.72 \text{ MPa}$ ($L_1 = 7500 \text{ N}$; $L_2 = 800 \text{ N}$; $R_\tau = 0.107$)	9 684	9 523
2.1.5		16 954	
2.2.4		6 053	
2.3.1		8 308	
2.4.5	Level 2: $\tau_{\max} = 22.08 \text{ MPa}$, $\tau_a = 9.92 \text{ MPa}$ ($L_1 = 6900 \text{ N}$; $L_2 = 700 \text{ N}$; $R_\tau = 0.101$)	9 936	23 356
2.1.2		16 779	
2.3.4		68 331	
2.4.2		26 122	
2.2.5	Level 3: $\tau_{\max} = 19.84 \text{ MPa}$, $\tau_a = 8.96 \text{ MPa}$ ($L_1 = 6200 \text{ N}$; $L_2 = 600 \text{ N}$; $R_\tau = 0.097$)	138 300	100 909
2.1.4		58 818	
2.4.3		149 017	
2.2.3		85 537	
2.6.4	Level 4: $\tau_{\max} = 17.92 \text{ MPa}$, $\tau_a = 8 \text{ MPa}$ ($L_1 = 5600 \text{ N}$; $L_2 = 600 \text{ N}$; $R_\tau = 0.107$)	581 059	668 846
2.7.4		1 389 288	
2.5.3		698 922	
2.7.3		354 701	

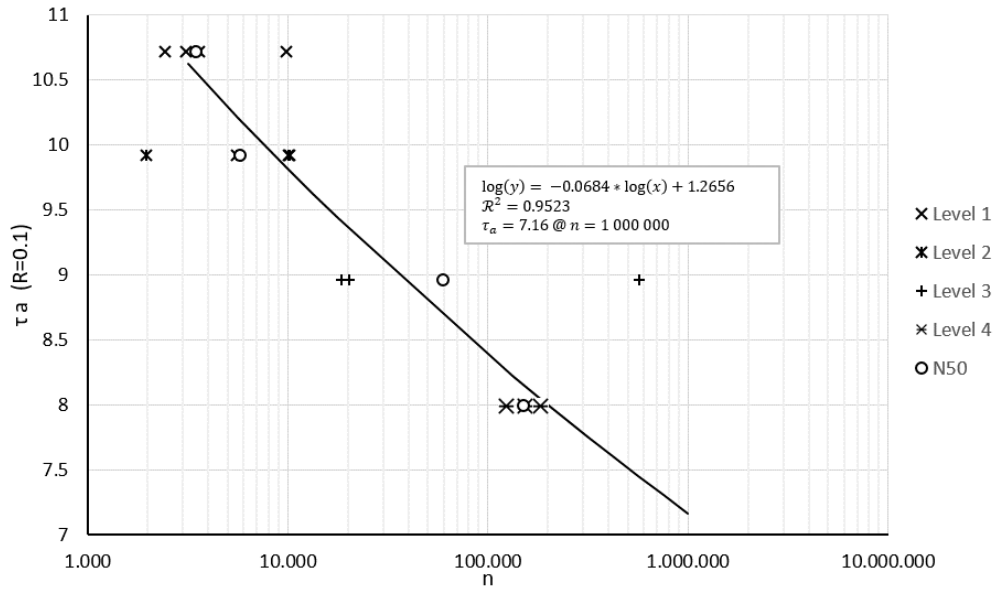

Figure 6.6: SN-Curve for 3M SAT 1100S 0.4 mm tape from fatigue testing. Shear stress amplitude at $R_\tau = 0.1$. Moderate spread in the results and a good predictability are observed.

Table 6.3: Fatigue results for Henkel Loctite EA 9514 liquid adhesive.

Sample ID	Load Level	Number of Cycles [–]	N 50 %
2.0.1	Level 1: $\tau_{\max} = 24 \text{ MPa}$, $\tau_a = 10.72 \text{ MPa}$ ($L_1 = 7500 \text{ N}$; $L_2 = 800 \text{ N}$; $R_\tau = 0.107$)	9 830	3 494
3.0.2		2 440	
4.0.5		3 603	
5.0.2		3 086	
7.0.1		3 758	
7.0.5		1 816	
6.0.1	Level 2: $\tau_{\max} = 22.08 \text{ MPa}$, $\tau_a = 9.92 \text{ MPa}$ ($L_1 = 6900 \text{ N}$; $L_2 = 700 \text{ N}$; $R_\tau = 0.101$)	5 582	5 798
1.0.3		1 971	
5.0.5		10 089	
5.0.1		10 176	
5.0.3	Level 3: $\tau_{\max} = 19.84 \text{ MPa}$, $\tau_a = 8.96 \text{ MPa}$ ($L_1 = 6200 \text{ N}$; $L_2 = 600 \text{ N}$; $R_\tau = 0.097$)	20 383	59 746
3.0.5		18 461	
2.0.4		566 771	
3.0.3	Level 4: $\tau_{\max} = 17.92 \text{ MPa}$, $\tau_a = 8 \text{ MPa}$ ($L_1 = 5600 \text{ N}$; $L_2 = 600 \text{ N}$; $R_\tau = 0.107$)	153 206	150 416
7.0.9		122 660	
2.0.2		181 095	

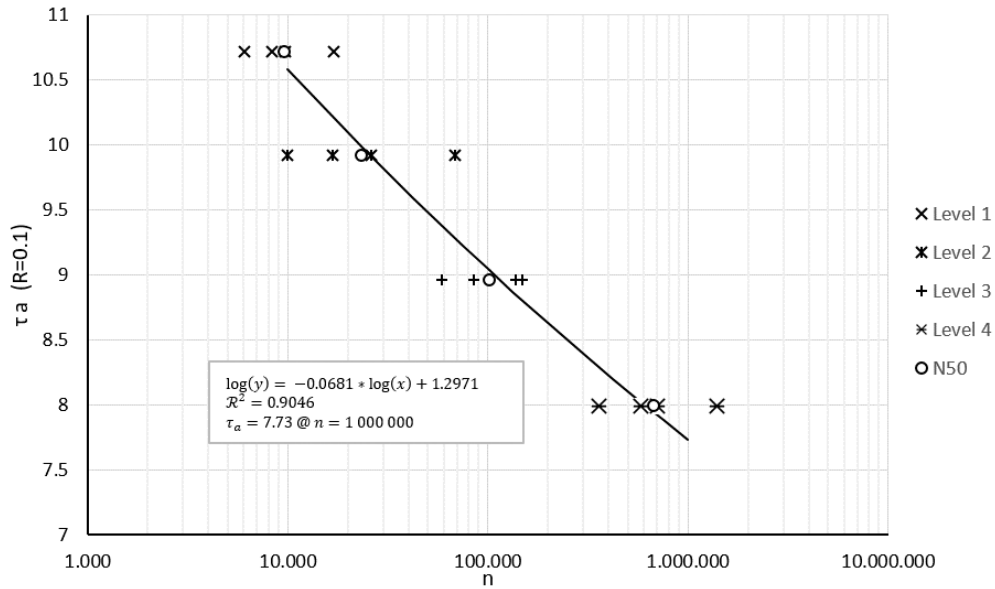


Figure 6.7: SN-Curve for Henkel Loctite EA 9514 0.1 mm from fatigue testing. Shear stress amplitude at $R_\tau = 0.1$. High spread in the results and a slightly worse predictability compared to the tape are observed.

6.2.4 Conclusions for Adhesive Selection

Both liquid adhesives Betamate 1630 and Henkel Loctite EA 9514 showed high sensibility to any inconsistencies. A defined bond thickness in the tight range of 0.1 *mm* was not achieved due to effect of warp in the bonded sheets or positioning errors. Especially for Loctite, this led to high variation in shear strength and adhesive failure mode. Application was much more consistent with the adhesive tape. Uniform bond thicknesses and thorough coverage of the bonding area could be achieved, resulting in very high static shear strength, far exceeding the data sheet value. Loctite liquid adhesive and 3M tape both seem to be suitable for high static load-bearing applications. For liquid adhesive, particular attention to the application process is advised, to ensure a uniform bond thickness.

Fatigue strength for the adhesive tape and Loctite liquid adhesive show a fatigue shear strength of $\tau_{a,3M} = 7.73 \text{ MPa}$ and $\tau_{a,loctite} = 7.16 \text{ MPa}$. This relates to relative fatigue strength of approximately 17 % with respect to the static strength for both adhesives for a pulsating load case. Considering the variety of possible inconsistencies at the application, non-uniformity of the shear stress in an overlap joint with stress concentration on the edges (see figure 2.5) and the neglect of potentially existing additional stress components due to peel effects or misalignment, the result for both adhesives is deemed acceptable. Both adhesives thus also show suitability for dynamic applications, while again, the liquid adhesive shows higher deviations in the achieved number of cycles, emphasizing the requirement of a precise application process. The tape adhesive overall shows higher consistency.

Based on these findings, an informed selection of adhesive for a prototype build process can be made. 3M SAT 1100S tape adhesive showed the highest static shear strength, highest fatigue strength and least variance in the results. Henkel Loctite EA 9514 followed closely, but with high variance. Still, selection was made for the liquid adhesive due to the following reasons. Firstly, considerably higher material cost of the tape adhesive. Secondly, the prototype selected, as will be seen in the next chapters, exhibits highly distorted surfaces at the regions of patch application. The advantage of the tape observed in the lap shear test, where the tape was able to achieve a uniform bond over the whole mating surface, may not translate onto complex shaped geometry. There, a liquid adhesive may be better able to fill gaps between the mating sheets by a variation in adhesive layer thickness.

Generally, it is advised to investigate further into the root causes for the strength deviations. The dependence on adhesive thickness and skewness, surface preparation and methods for gap preservation should be studied to achieve a robust gluing process for possible serial application. Also, one must consider a sufficient safety margin when using adhesive for fatigue load cases.

Chapter 7

Validation

To validate the method presented, a real-world serial part is optimized, built and tested. The aim is to show mass reduction potential and check that the part's strength can in fact be maintained at lower mass using this method. While the method is applicable for any sheet metal components with known load case, some part with high functional requirements but also high desire for lightweighting should be used. Then, after optimizing, the manufactured patched design should be tested and its performance – in comparison to a serial part – be evaluated in terms of mechanical behavior under realistic load.

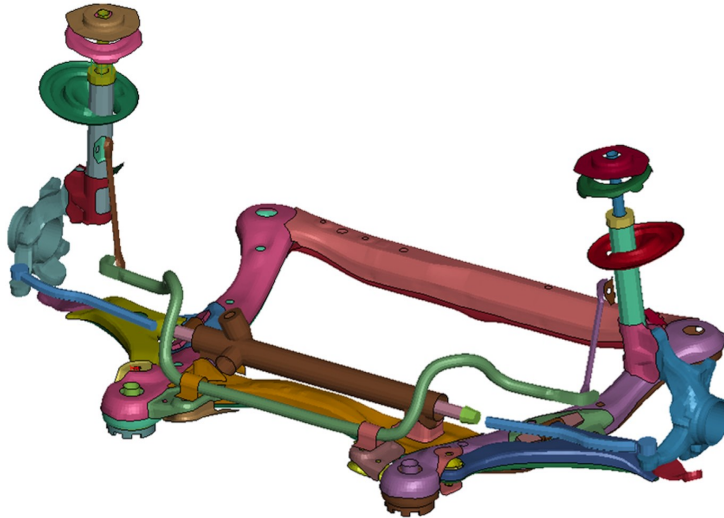
7.1 Component Selection

In a car, there are plenty of sheet metal components that could be suitable for patch optimization. A lot of lightweighting methods such as tailored blanks are found in the automotive body [MJLK14]. For most cars, the body is a welded unity and critical load cases are overall body stiffness or crash scenarios. Optimization of a single component can be hard to do, since often any part needs to be evaluated within the whole system, complicating simulations and testing. In automotive chassis, mass reduction is as important since unsprung masses have an over-proportionally high effect on driving safety and handling [Bas00]. While playing a role in crash load cases, chassis components are generally designed to sustain operating loads and behave stiff to keep a predictable driving behavior. For this project, a chassis part is selected since it can be handled more efficiently as an individual component with a well-defined load case. Fatigue load cases, which are critical in chassis design, make a particularly challenging and interesting area of application for adhesives which are not that widely used for fatigue load cases, so far. As the component selection is driven by being representative for real-world applications, a wishbone link for McPherson front suspension is chosen. It is the most common design found in modern cars [SRTFS20] and components often feature complex, highly deformed geometries. It can be assumed that serial production wishbone links are already highly mass-optimized, so being able to optimize these highly sophisticated components even

further would emphasize the method’s capabilities. The wishbone link of a Toyota Camry 2012 model (XV50) is selected as the Camry is among the most sold cars worldwide and a model of the whole car, including the suspension, is publicly available through George Mason University [MBP⁺14]. An image of the car and front axle model can be seen in figures 7.1a and 7.1b.



(a) Toyota Camry 2012 model.



(b) Toyota Camry front suspension.

Figure 7.1: Toyota Camry 2012 model and its front suspension, based on [MBP⁺14].

7.2 Reverse Engineering

The wishbone link model provided by GMU is very coarse and includes only general material data. Reverse engineering is required to achieve a more precise geometry, material data and welds for the optimization and accurate prototype build. For doing so, the model was 3D-scanned and a fully parametric CAD model created within this project.

Figure 7.2 shows the recreated wishbone link CAD model, consisting of two sheet metal shells, labeled *ET1* and *ET2* and two bushings labeled *ET3* and *ET4*. The two shells are welded to each other by MAG-welds. It's main shell's material was determined to be steel with a thickness of 3 *mm* and grade S500MC. This was derived from spark spectrometry and tensile testing of specimens taken from the part itself. S500MC is a microalloyed thermomechanically rolled high-strength steel for cold forming with high available thicknesses up to 12.7 *mm* [Fla14]. Properties are summarized in table 7.1.

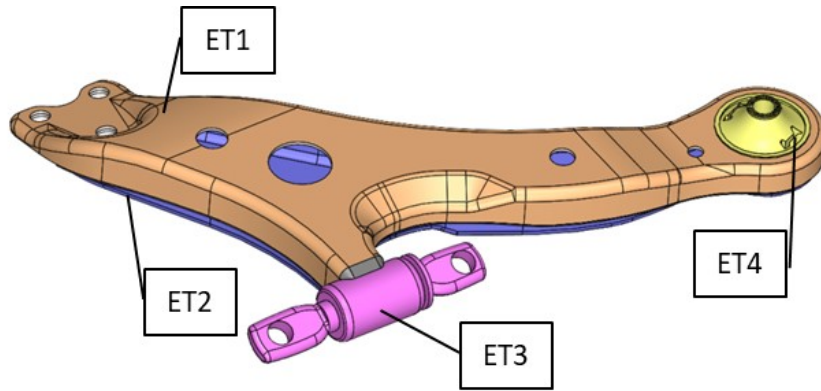


Figure 7.2: Toyota Camry wishbone link consisting of two main shells (*ET1* and *ET2*), cylindrical bushing (*ET3*) and spherical bushing (*ET4*).

Table 7.1: S500MC $t = 3$ *mm* mechanical properties [Fla14]

yield strength R_e	≥ 500 <i>MPa</i>
tensile strength R_m	550 – 700 <i>MPa</i>
total elongation A_{50}	≥ 14 %
bending radius	1.5 t

The welds are found to be one-sided a3 lap welds, bridging a gap of approximately 1 *mm* throughout the part's circumference. This was determined by metallurgical analysis on cross-section cuts as well as CT-scans, as can be seen in figure 7.3. In a student project [BKPN21] the required weld quality was evaluated to be of class C according to DIN EN ISO 5817, see norm [NN223].

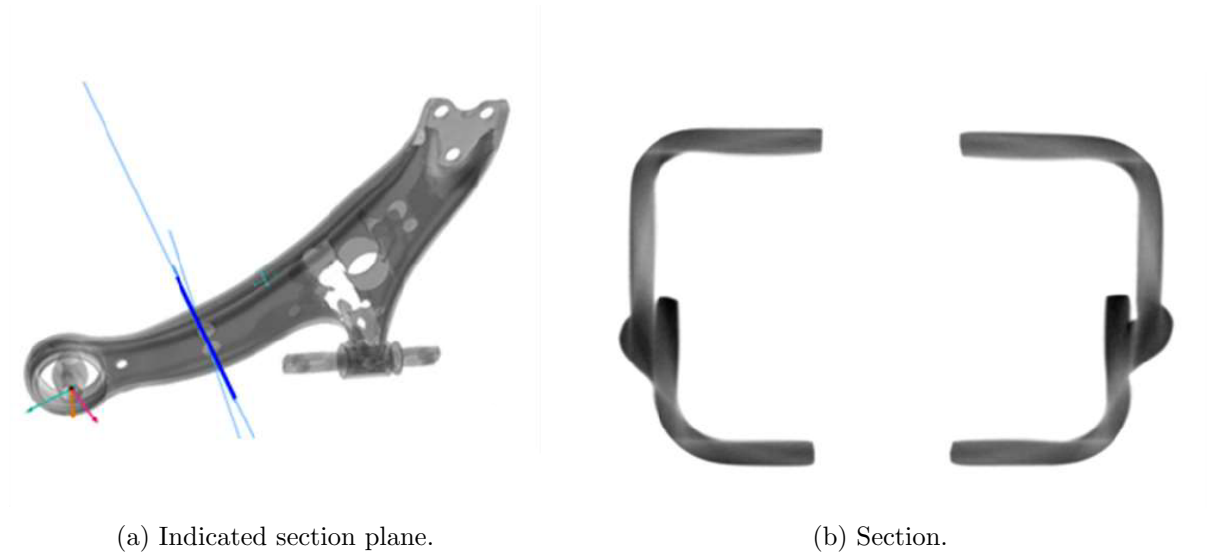


Figure 7.3: CT scan of Toyota Camry wishbone link [BKPN21]. The clearance between ET1 and ET2, as well as the weld geometry, can be observed.

7.3 Loading

To do sensible optimization, a realistic load case for the component must be known. Here, a fatigue load case is of main interest, as fatigue strength of the adhesive bonded patch is to be evaluated. As there is no specific information available for the Toyota Camry, this was based upon CARLOS-Multi standard [SKH94], which describes average loads acting upon a car when driving. The component loading was derived by the author and within student projects [HKN22],[HKN24]. CARLOS-Multi summarizes test results from eight different vehicles traveling 5400 km on public roads while measuring wheel forces, resulting in an average expected load collective for 40000 km, normalized to axle weight separately for front and rear axles and for front and rear-wheel drive cars. Table 7.2 shows the composition of different road types driven and their extrapolation factors. Figure 7.4 shows level crossing frequencies of the resulting load collectives. Lateral and longitudinal events overlap which is described by a joint density distribution. Level crossings are also given in tabular form.

To get component loading from this, multibody analysis of the Toyota Camry front suspension shows that the wishbone link takes the main forces in lateral and longitudinal direction, while playing no significant role in vertical direction which is consequently disregarded further on. Longitudinal forces are mainly braking and accelerating load cases while lateral forces arise from cornering. Vertical forces from road bumps are absorbed by the suspension strut. See figures 7.5 and 7.6 for orientation.

Table 7.2: CARLOS Multi recorded distance, extrapolation factors and total distance [HKN24]
based on [MBP⁺14].

Road type	Recorded distance [km]	Extrapolation factor [-]	Extrapolated distance [km]	Total return period [km]
rough road	151	10.60	1600	
country road, good	2420	2.31	5600	
country road, bad	1409	8.23	11600	40000
city road	618	14.89	9200	
highway	840	14.29	12000	

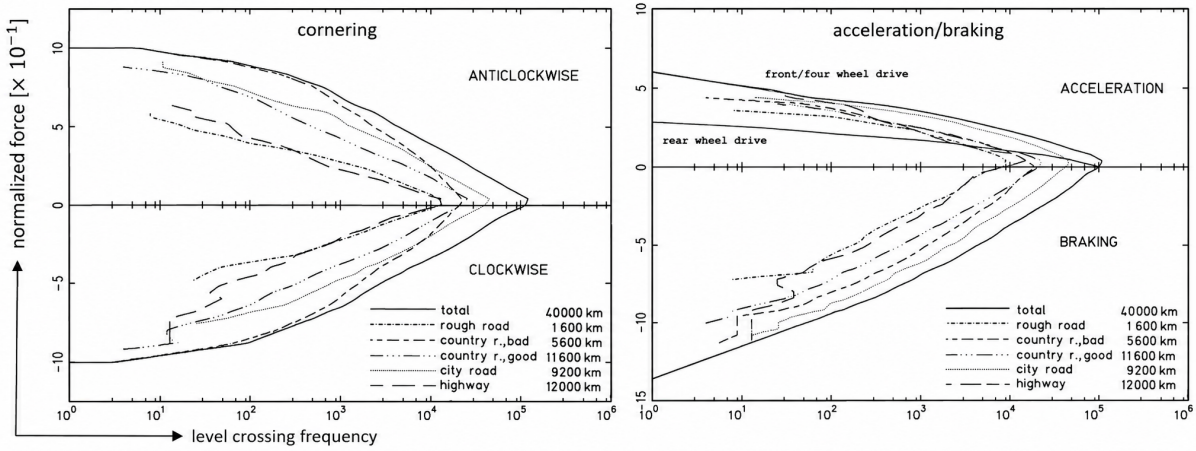


Figure 7.4: CARLOS-Multi level crossing frequencies for lateral and longitudinal forces, [HKN24] based on [SKH94].

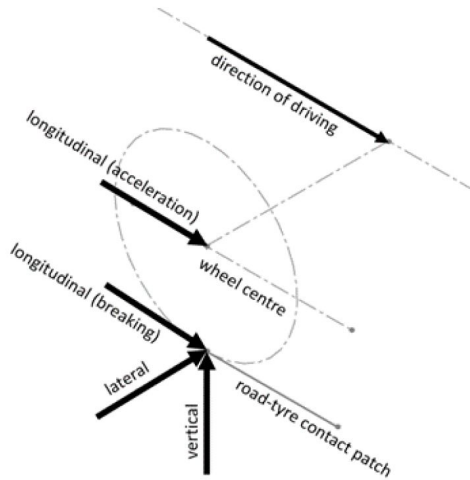


Figure 7.5: Wheel forces directions [HKN24].

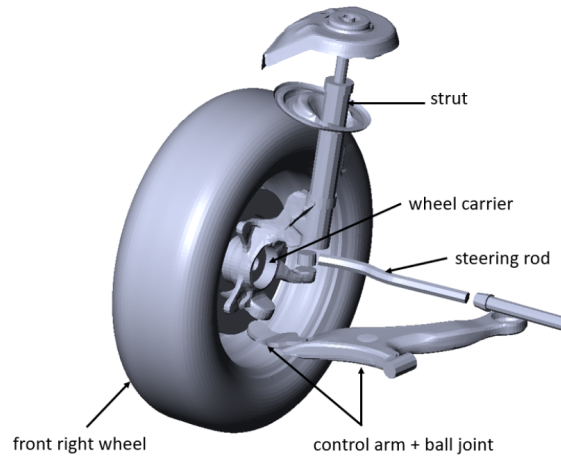


Figure 7.6: Toyota Camry McPherson suspension detail, [HKN24] based on [MBP⁺14].

Lateral forces, as well as braking, act on the road-tire contact patch. Longitudinal events of acceleration and engine braking act on the wheel center as the moment induced by leverage to the tire contact patch is counteracted by a moment in the driveshaft. In the following, these are handled as separate load cases, namely braking (+/-), longitudinal (+/-) and lateral (+/-). Relative to the wishbone link, load direction changes when the suspension is compressed, which, apart from road bumps, also happens due to weight shift. However, the effect is small and only decreases the wishbone links load share and was thus neglected for further analysis. For each load case, from the given load collectives from CARLOS-Multi, fatigue assessment according to FKM-Guideline [Ren20] is performed for a extrapolation to 200 000 km component lifetime. Table 7.3 shows the degrees of utilization.

Table 7.3: Degree of utilization for wishbone link at CARLOS-Multi load cases, according to FKM-Guideline, for 200 000 km [HKN24].

No.	load case	degree of utilization
1	lateral (+)	0.188
2	lateral (-)	0.082
3	longitudinal (+)	0.241
4	longitudinal (-)	0.145
5	braking (+)	0.144
6	braking (-)	0.760
-	combined	0.849

As stress hotspots and principle stress direction are comparable among load cases, they can be combined as the table summarizes. A comparison of principal stress vectors can be seen in figure 7.7 and the stress hotspots for all load cases are depicted in figure 7.8.

It is therefore a reasonable abstraction to optimize the component with respect to a single representative load case, since the expected error due to minor deviations in load direction is small and maximum stresses occur at the same locations. However, this assumption may no longer hold after geometric modifications, such as patching, where stress hotspots could appear in different locations and the validity of this abstraction may be compromised.

The design load by linear equivalent damage is

$$F = \begin{bmatrix} -4850 \\ 424 \\ 0 \end{bmatrix} N \quad (7.1)$$

which represents all loading on the wishbone link and is expected to hold 50 000 cycles at a 50 % survival probability (representing 200 000 km). This load can be used for patch optimization and also for later validation.

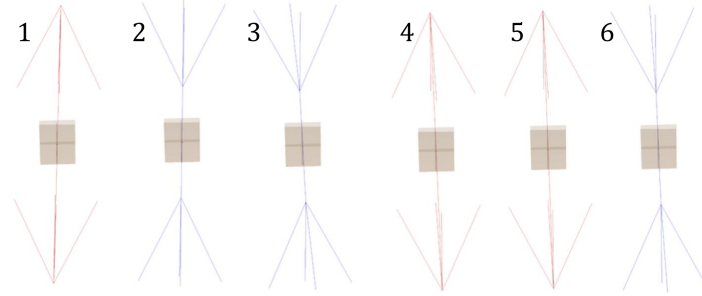


Figure 7.7: Principal stress vectors for load cases as in table 7.3 from left to right [HKN24]. While some load cases show compressive stress and others tensile stress, the direction shows only minor deviations.

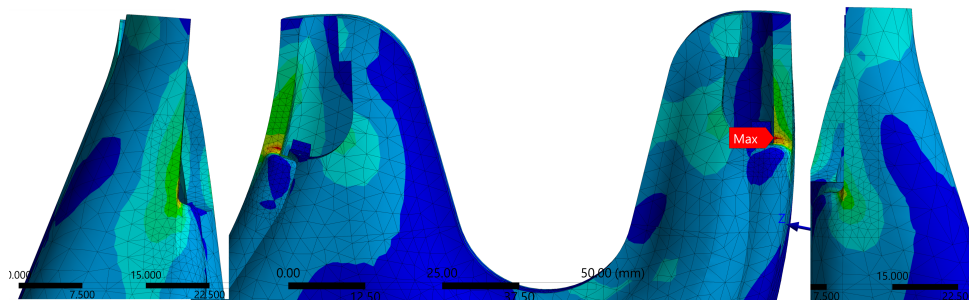


Figure 7.8: Maximum equivalent stress locations for all load cases as listed in table 7.3 [HKN24]. Depending on the load case, the maximum stress occurs either at the right or left shoulder of B. In this figure, mounting region B is not shown. In both cases, the stress concentration is located at the weld ends.

7.4 Mass Optimization by Patch Optimizer

7.4.1 Geometry Definition

For applying the patch optimizer software, the wishbone link is abstracted to a single shell component. Attachment pieces ET3 and ET4 are excluded. In reality, the main part is welded from two overlapping shells ET1 and ET2, see figure 7.9a. To form a single shell, these are merged, while their overlap and weld thickness is disregarded – as is their width difference, see figure 7.9b– setting $w = w_1$ to keep outer dimensions consistent. The shell is meshed into 26 549 quadrilateral elements with a maximum size of 3 mm.

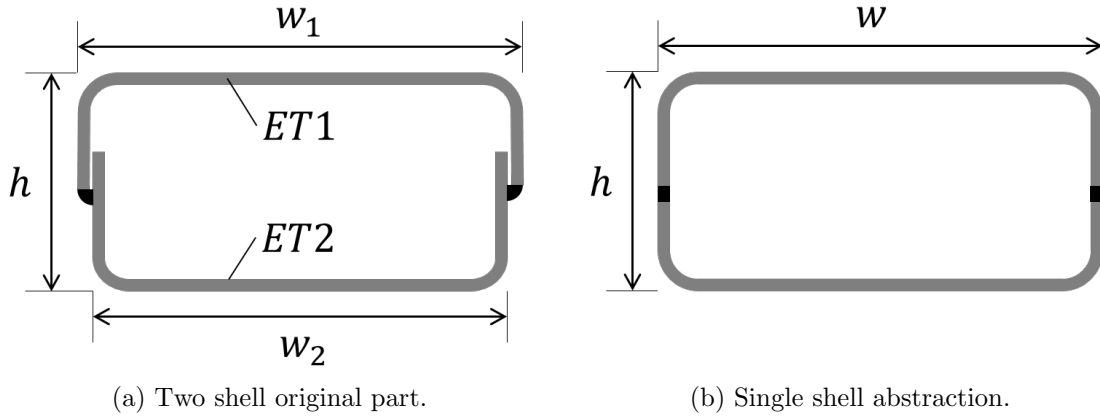


Figure 7.9: Wishbone link section with shells and weld indicated. The real geometry is abstracted to a singular shell for the optimization software.

7.4.2 Loads and Constraints

The main shell is constrained along its contact edges to the respective components ET3 and ET4 as well as the mounting holes at B. Figure 7.10 shows the meshed shell and the constrained edges highlighted. The coordinate system is not shown here, as the car's global coordinate system is used with origin in the center of the front axle and the x-axis pointing in driving direction. In the following, the constraint regions will be named A, B and C, where A represents the spherical rubber bushing ET4, B is the mounting region for the wheel carrier and C represents the cylindrical rubber bushing ET3.

The bushing at A is replaced by a remote support at its center with fixed translation and free rotation, resembling a spherical bearing. There is no loading at A, thus, with P, D, F being remote point location, displacement and force, respectively:

$$P_A = \begin{bmatrix} -1226.6207 \\ -366 \\ 226.5198 \end{bmatrix}, \quad D_A = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \infty \\ \infty \\ \infty \end{bmatrix}, \quad F_A = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}. \quad (7.2)$$

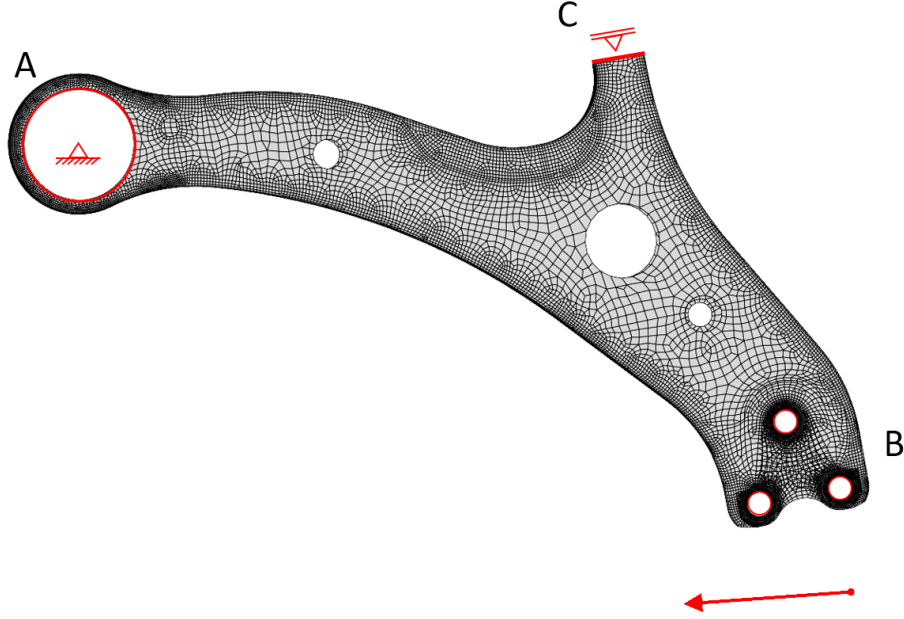


Figure 7.10: Wishbone link mesh and constraints. At A, the spherical bushing is represented by fixing the indicated circle's center. At C, the cylindrical bushing is represented by allowing movement along the axis of C-A. At B, the load is applied.

The bushing at C is replaced by a remote support at its center with free translation along its axis, which goes through P_A as figure 7.10 shows, resembling a linearly sliding spherical bearing, with

$$P_C = \begin{bmatrix} -940.212 \\ -423.5328 \\ 211.813 \end{bmatrix}, \quad D_C = \begin{bmatrix} \infty \\ 0 \\ 0 \\ \infty \\ \infty \\ \infty \end{bmatrix}, \quad F_C = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}. \quad (7.3)$$

Finally, load application point B is defined. Here, a ball joint piece is attached between wheel carrier and wishbone link, see figure 7.11. The ball's center defines load application point P_B . To prevent rigid-body rotation around the axis $P_A - P_C$, vertical deflection of the link is prohibited:

$$P_B = \begin{bmatrix} -936.51 \\ -750 \\ 210 \end{bmatrix}, \quad D_B = \begin{bmatrix} \infty \\ \infty \\ 0 \\ \infty \\ \infty \\ \infty \end{bmatrix}, \quad F_B = \begin{bmatrix} -4850 \\ 424 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}. \quad (7.4)$$

From this, the system is statically determined. In reality, as supports at A and C are rubber bushings, some rigid-body movement would be allowed, leading to a much higher overall deflection of B compared to this model. Preparatory finite element analysis showed that the abstraction to rigid supports did not significantly influence the links stress and

strain state, which is the only concern for optimization, i.e. rigid body movement can be disregarded.

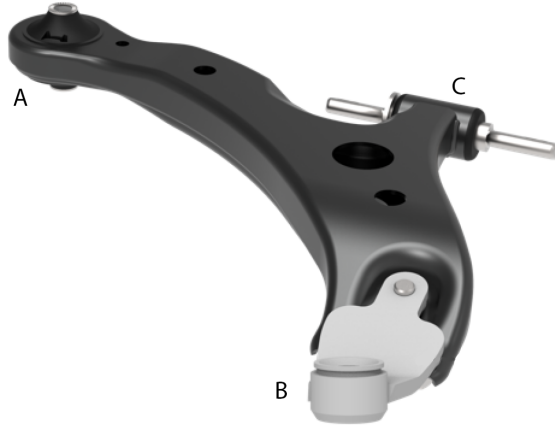


Figure 7.11: Wishbone link with ball joint attachment at B [HKN24].

7.4.3 Restricted Areas

Restricted areas are those where adding a patch would be unfavorable with respect to manufacturability, strength or outer dimensions. As seen in chapter 6, glued connections are sensible to mating surface quality. The link was abstracted along the weld line to a single shell. In reality, the weld's and close-by component's surface are uneven and hard to define. To account for this, the weld line is restricted by selection of an uninterrupted chain of 653 elements along the split line and penalization with a high factor of 10^9 . See figure 7.12 for the selected elements. For comparison, also an unrestricted version where patching on the weld line would be allowed is computed. These two cases are subsequently named the *restricted case* and the *unrestricted case*.

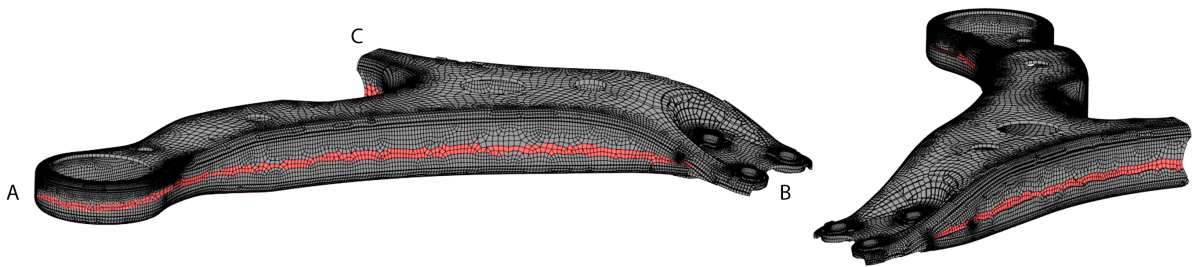


Figure 7.12: Wishbone link weld seam elements for restricted element set.

7.4.4 U^* results

U^* computation was performed with 504 points, the results can be seen in figure 7.13. As introduced in section 2.3.2, the U^* -Index describes the degree of connectivity to the load application points, normalized to a scale of 0 to 1. Elements close to load application point B show a very high value close to 1. Elements close to supports A and C get close to 0 - this is expected, as proximity to a support will generally lead to a relatively stiffer connection to those instead of the load application point. It can be seen, that the region between A and C shows moderately high U^* -values, indicating no particularly stiff connection. The region between A and B shows higher U^* -values, indicating high degree of connectivity and thus a, stiff area. The region close to B, between B and the center hole, shows very low values, an indication of very little stiffness.

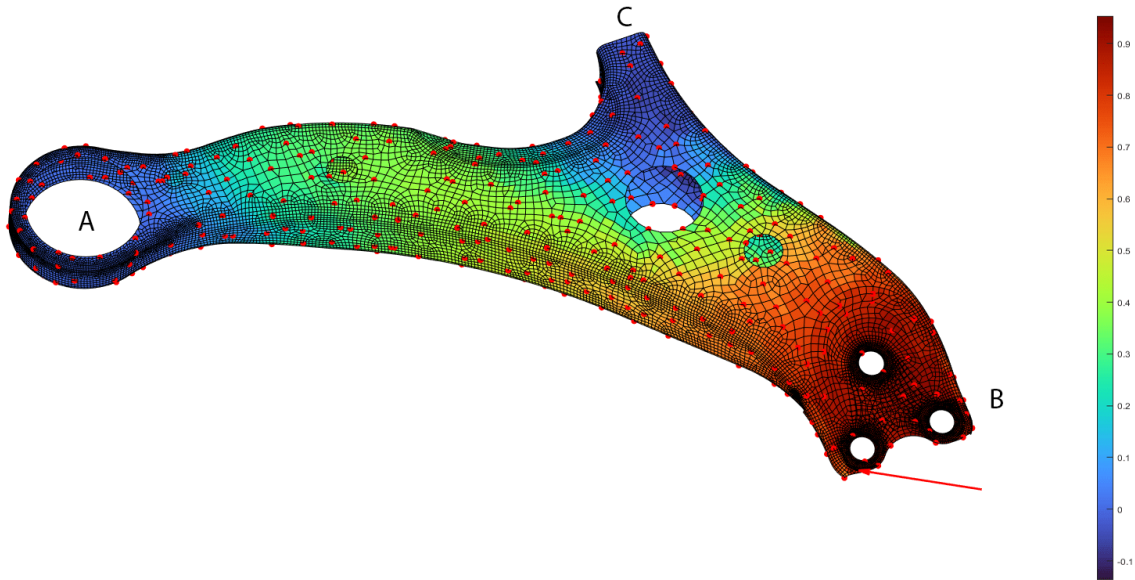


Figure 7.13: U^* results on wishbone link with 504 computed nodes. Red dots show the computed nodes and the coloring shows their interpolation to all elements. As would be expected, elements closer to B are more connected to it, and thus, show a high value close to 1. Elements further away decrease in U^* -index with respect to B.

7.4.5 Path Results

Based on U^* , paths were searched. They originate from 256 start elements on B and can find 205 target elements in A and C. Figures 7.14 and 7.15 show paths found for the restricted and unrestricted case. For both cases, from no start element the stiffest path found leads all the way up to A, instead all go up to C. The reason is, that the connection to C is much shorter and stiffer for all start elements. A lots of paths eventually follow the same route from midway between B and C to C. There, path density will be very high,

so it is expected to see a wide patch forming there in the next step. The restriction does not seem to interfere a lot with the paths, as they mostly run alongside the restricted weld line. Some differences can be observed close to B, where the restriction hinders paths to go over the shoulder and shifts them down instead, as can be seen in the figures' rightmost images. The restrictions in this example seem to not be interfering a lot with finding efficient paths. In other cases, differences to an unrestricted design may be more severe.

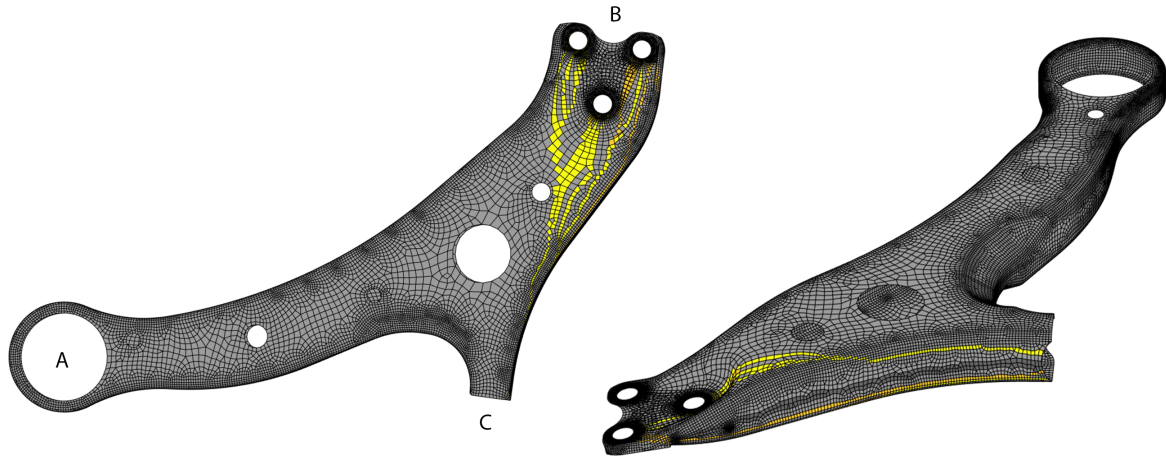


Figure 7.14: Wishbone link with paths highlighted for the restricted case. Paths do not cross the weld seam, as that region is restricted, but go alongside it.

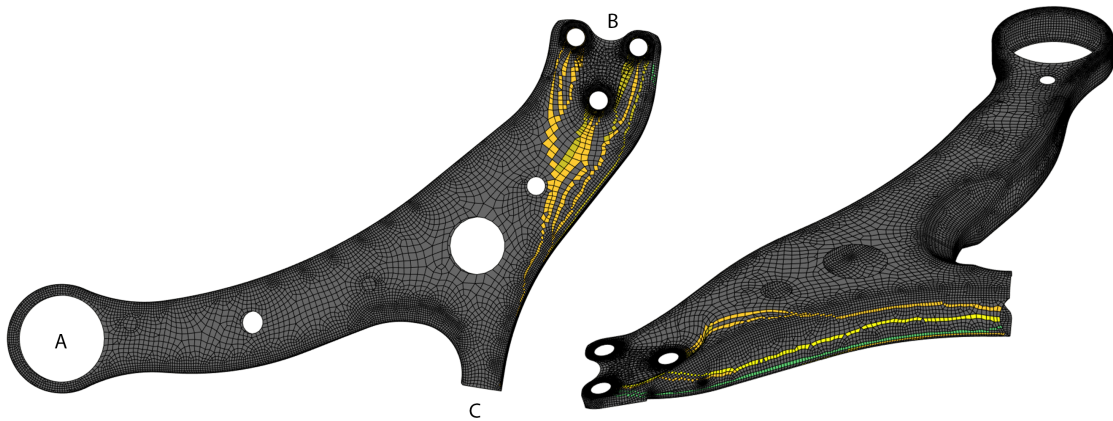


Figure 7.15: Wishbone link paths highlighted for unrestricted case. Some paths cross the weld seam, as it is not restricted here. Still, most paths go alongside it.

7.4.6 Patch Results

From the paths, patches were created. The width factors analyzed were

$$k_w = [20, 40, 60, 80, 100, 120]. \quad (7.5)$$

See figures 7.16 and 7.17 for width $k_w = 20$ in the restricted and unrestricted case. Due to high U^* gradient ∇ close to B, almost all of the area is patched. Also midway from B to C, path density is high, creating a wide patch. Most interesting is the difference in patches from the restricted to the unrestricted case. Patches are not allowed to grow over the restricted region, so even though some paths lie in close proximity to it, they grow asymmetrically into the other direction. While in the unrestricted case, patches from the top and bottom half of B eventually merge midway between B and C, in the restricted case the two branches stay separate, so also two separate patches result. The unrestricted case results not only in a patch that would lie over the weld seam, which could cause issues in manufacturing the bond (which is why the restriction was chosen that way in the first place) but also create a highly deformed shape, wrapping around the whole right flank of the base. The restricted case yields two separate patches which in themselves seem to be much less complex, being relatively straight and without mayor deformation. Whether this simplification also creates a comparably stiff result will be checked in the next step.

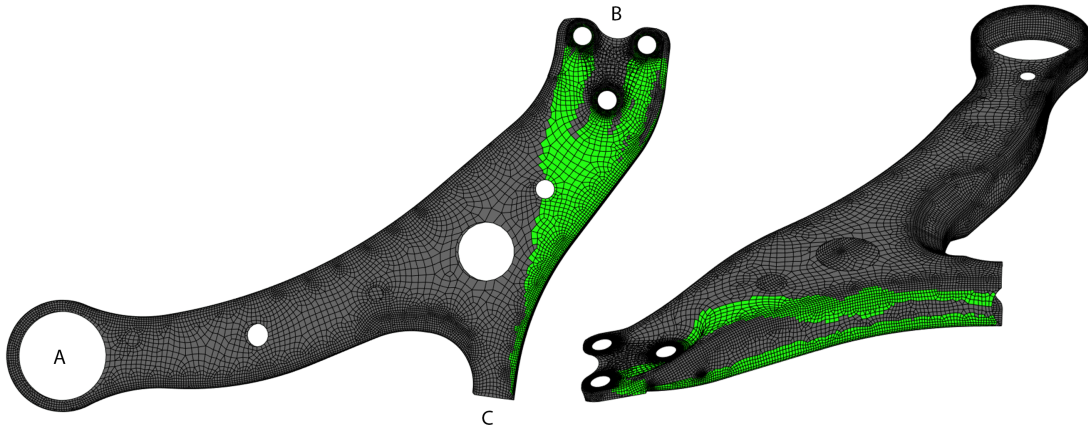


Figure 7.16: Wishbone link patches restricted case at width $k_w = 20$. The patch does not cross the restricted elements. Instead, patches grow away from it. Two separate patches form, as they cannot connect to each other due to the restricted weld seam. Both patches exhibit a rather simple and straight shape.

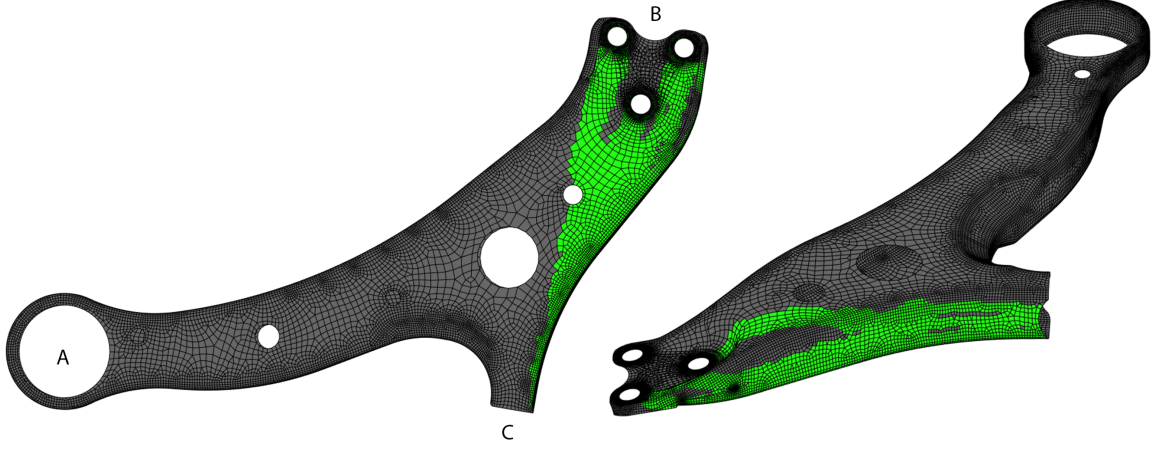


Figure 7.17: Wishbone link patches unrestricted case at width $k_w = 20$. One singular patch forms, that connects over the weld seam halfway from B to C. Apart from issues with bonding this patch above the irregular surface of the weld seam, this shape is also more complex to form as it grabs around the geometry.

7.4.7 Thickness Tuning Results and Discussion

The initial base material thickness is 3 mm. Optimization was performed with respect to maximum equivalent stress, which for the initial state is $g_0 = 324 \text{ MPa}$ under the given loads, see equation (7.1). This is comparable to the real non-simplified model, which showed a maximum 359 MPa equivalent stress in the same location. The negative deviation for the simplified model can be explained with the simplification of geometric discontinuities in the weld region, which on the real part lead to stress concentration. Figure 7.18 shows equivalent stress for the unpatched state, with stress concentration on the left shoulder at B.

The range of investigated base thicknesses is

$$t_b = [1.50, 1.75, 2.00, 2.25, 2.50, 2.75] \text{ mm}. \quad (7.6)$$

Lower base thicknesses showed to not be stable, and optimization unable to converge. For each, patch thickness is reduced until equivalent stress of the patched design converges with a maximum deviation of $Q_g = 1 \%$ to the unpatched initial design. Tables 7.4a and 7.4b give the resulting patch thickness for the restricted and unrestricted case. Tables 7.5a, 7.5b and corresponding graphs 7.19a and 7.19b show the resulting overall volume reduction for best k_w with respective patch thicknesses for each analyzed base thickness.

Maximum found volume reduction for the restricted case is 31.2 % and 31.8 % for the unrestricted case, respectively. It lies on the very edge of the investigated range at $t_b = 1.5 \text{ mm}$ and $t_p = 2.56 \text{ mm}$ ($k_w = 120$) for both cases. This is on the very limit of base thickness decrease before the models behavior gets unstable and the FEA solver unable to converge, likely due to nonlinearities such as buckling occurring.

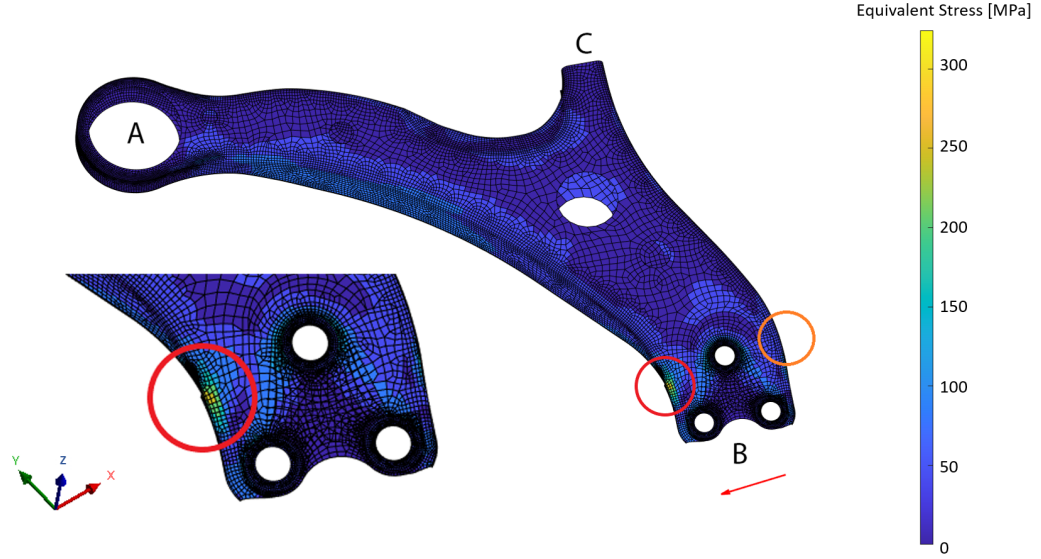


Figure 7.18: Wishbone link equivalent stress for initial, unpatched state. The maximum stress hotspot can be seen on the left shoulder at B (red circle) and has a magnitude of $g_0 = 324 \text{ MPa}$. Another hotspot lies on the right shoulder of B (orange circle) with a magnitude of approximately 275 MPa .

Table 7.4: Wishbone link optimized patch thickness t_p results for base thickness t_b and width factor k_w combinations.

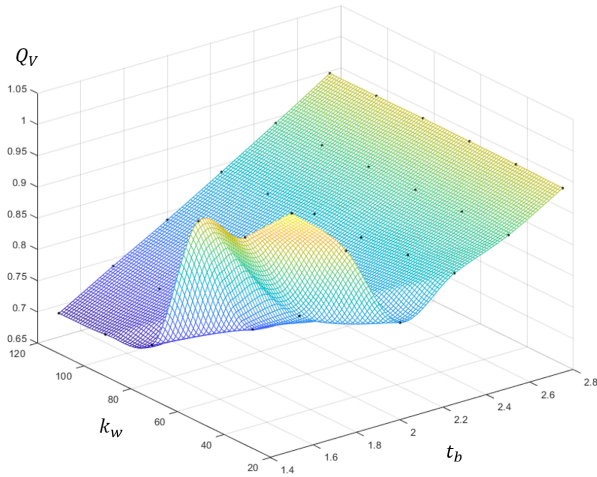
(a) t_p for restricted case.							(b) t_p for unrestricted case.						
	base thickness t_b [mm]							base thickness t_b [mm]					
k_w	1.50	1.75	2.00	2.25	2.50	2.75	k_w	1.50	1.75	2.00	2.25	2.50	2.75
20	10.34	7.07	2.71	2.13	1.21	0.56	20	10.35	7.60	2.78	2.08	0.96	0.48
40	7.94	3.86	2.53	1.78	1.10	0.53	40	8.32	4.25	2.44	1.72	0.90	0.45
60	7.10	2.61	2.07	1.49	0.97	0.49	60	7.54	2.51	1.66	1.23	0.81	0.42
80	3.15	2.43	1.90	1.41	0.92	0.46	80	3.22	2.02	1.60	1.19	0.79	0.40
100	2.70	2.19	1.72	1.27	0.84	0.42	100	2.73	1.86	1.48	1.10	0.73	0.38
120	2.56	2.11	1.64	1.21	0.79	0.40	120	2.49	1.74	1.39	1.03	0.69	0.35

For the restriction on weld seams that prevented patches from entering the welded region a decrease in mass reduction potential would be expected. Remarkably, the restricted model shows almost no difference to the unrestricted case, as efficient paths generally go along the weld and crossing it does not give a huge advantage, see figures 7.14 and 7.15.

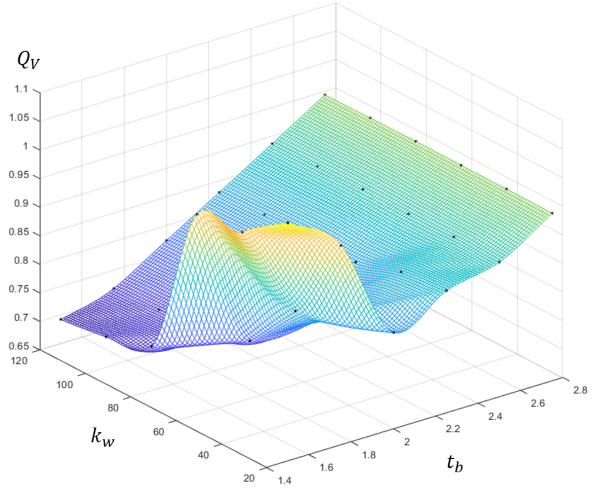
Only on patch growth, there is a minor disadvantage from not growing over the weld and thus connecting the two parallel patches from midway between B and C up to C, as can be seen in figures 7.16 and 7.17.

Table 7.5: Wishbone link performance and relative volume results for optimal base thickness and patch thickness combinations. The restricted case achieves almost the same volume reduction Q_V as the unrestricted case. A combination of low base thickness t_b and high patch thickness t_p yield the lowest volumes. The results from which were selected for prototype building are highlighted in bold in table (a).

(a) Restricted case.					(b) Unrestricted case.				
t_b [mm]	t_p [mm]	k_w [-]	g [MPa]	Q_v [-]	t_b [mm]	t_p [mm]	k_w [-]	g [MPa]	Q_v [-]
1.50	2.56	120	322	0.688	1.50	2.49	120	321	0.682
1.75	2.19	100	322	0.738	1.75	1.74	120	321	0.718
2.00	1.64	120	322	0.787	2.00	1.39	120	322	0.774
2.25	1.27	100	322	0.836	2.25	1.03	120	322	0.830
2.50	0.79	120	322	0.891	2.50	0.95	20	322	0.885
2.75	0.56	20	322	0.946	2.75	0.47	20	321	0.943
3.00	—	—	324	1	3.00	—	—	324	1



(a) Restricted case.



(b) Unrestricted case.

Figure 7.19: Response Surface for the wishbone link volume reduction Q_V over base thickness t_b and width factor k_w . Both variants behave very similar. Although heavily constrained, the restricted case shows a comparable maximum mass reduction potential, highlighting the method's ability to handle restrictions.

Comparing the equivalent stress for the patched state, see figures 7.20 and 7.21, with the unpatched state as in figure 7.18, it also shows that even though the stress hotspot lies outside the patched region, stress intensity there can be maintained at lower base thickness by redirecting the load. It can be concluded that for this component, path-based optimization finds a significant volume reduction even for restricted geometry. Remarkably, the volume reduction deviates only negligibly when considering the restriction on weld seams.

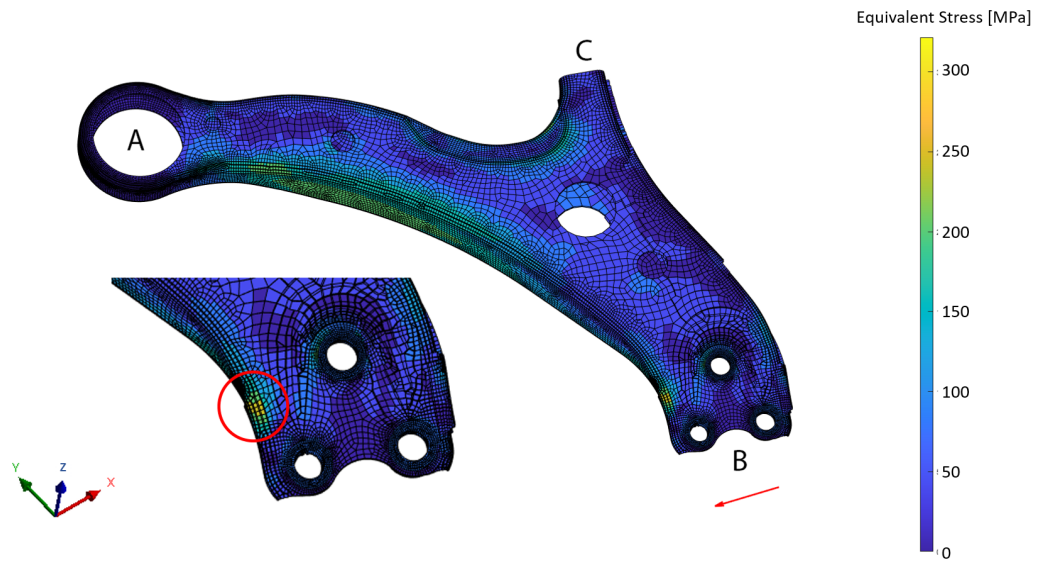


Figure 7.20: Wishbone link equivalent stress for restricted state. The stress hotspot did not change from the initial, unpatched state, but also did not increase in magnitude even though the base thickness at that spot decreased and no patching on that very spot was performed.

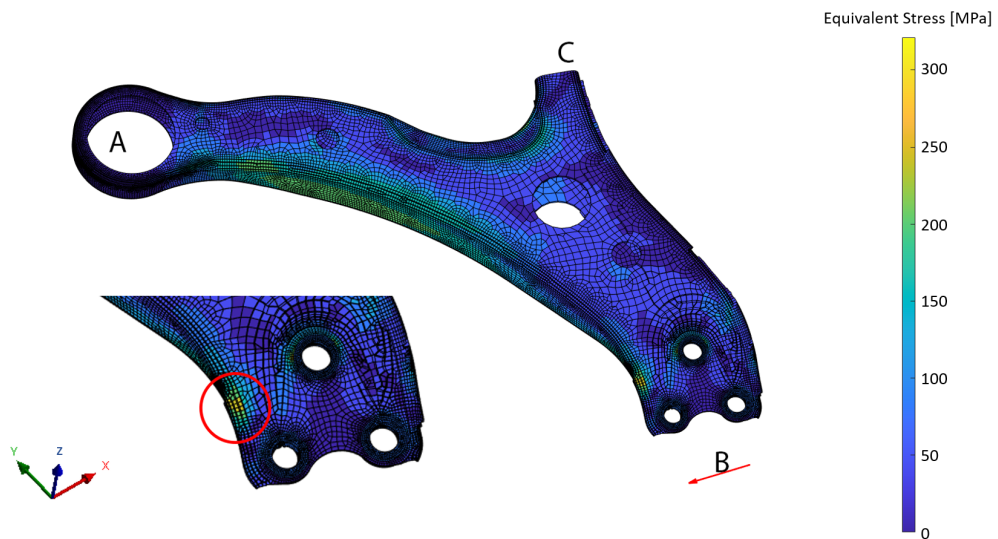


Figure 7.21: Wishbone link equivalent stress for unrestricted state. Likewise, the stress hotspot did not change from the initial, unpatched state, but also did not increase in magnitude even though the base thickness at that spot decreased and no patching on that very spot was performed.

7.5 Derivation of Prototypes from Optimizer Results

To verify the patch geometries obtained from the optimizer, representative prototypes of the complete wishbone link were defined and designed. The selected designs were first validated virtually by finite element simulation and subsequently manufactured and physically tested.

Three configurations were considered. The first corresponds to the serial design, consisting of a 3 mm thick wishbone link without any patch. This configuration is denoted as *type A* in the following and serves as the reference for comparison with the patched variants.

Based on the simulation results summarized in table 7.5a, two mass-reduced variants were selected. Specimens of *type B* use a base sheet thickness of 2.2 mm and a patch thickness of 1.2 mm for both patches. Due to material availability, these values were rounded down from the calculated required thicknesses of $t_b = 2.25$ mm and $t_p = 1.27$ mm. Between base and patch, an adhesive layer over the whole patch area is considered with a thickness of $t_a = 0.2$ mm.

In addition, a second mass-reduced variant, denoted as *type C*, was defined with a base thickness of $t_b = 1.8$ mm and a patch thickness of $t_p = 2.2$ mm. This configuration represents the simulation result of $t_b = 1.75$ mm and $t_p = 2.19$ mm and is therefore slightly rounded up from the calculated values. Between base and patch, the adhesive layer considered also with a thickness of $t_a = 0.2$ mm.

As the patch optimizer generates its output directly from the discretized mesh, the resulting patch contours are not smooth. The contours may contain small irregularities or holes where individual elements were excluded. For prototype manufacturing, the patch geometry was therefore manually refined. As there were only minor discrepancies in the type B and C patch contours, they were unified for further simplification in the patch manufacturing. Finally, the mounting region at B was filled completely to allow proper attachment of the load-transferring ball joint. The resulting patch outline is shown in figure 7.22.

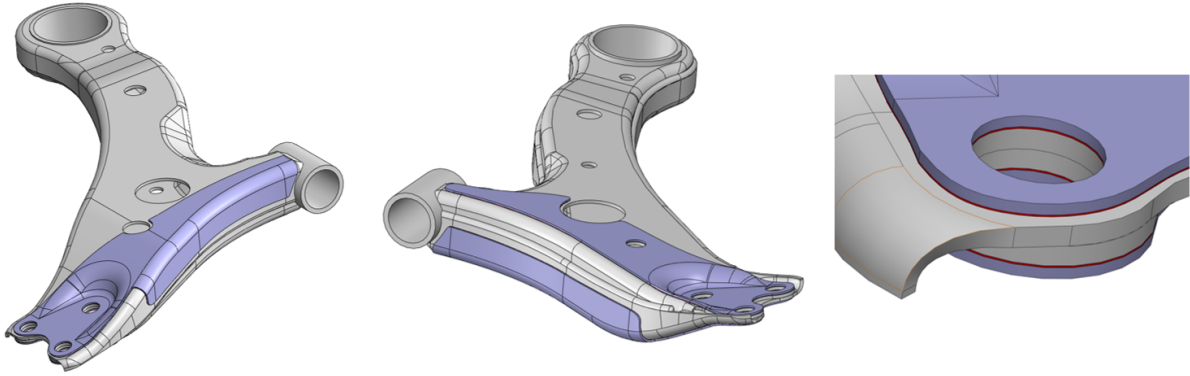


Figure 7.22: Wishbone link 2.2 mm variant patched. The patch follows the computed design with some smoothing on edges and fill of the contact area to the later added ball joint at B. The adhesive is modeled as a separate solid with 0.2 mm thickness.

7.6 Virtual Validation

The optimizer operates on a simplified structural model with a uniform mesh resolution. It is configured to generate patched designs that achieve the same maximum equivalent stress as the non-patched reference. The integrated finite element solver uses shell elements and increases their thickness in patched regions without modeling multiple layers. Thus, the lower-stiffness adhesive layer between base sheet and patch is neglected. The geometrical simplifications were introduced to limit computational effort. Although an explicit modeling of multiple layers is technically possible, it was not implemented in the optimizer at its current stage of development.

The finally selected designs were subsequently analyzed manually using a more sophisticated finite element model. This refined model incorporates weld geometry, an explicit adhesive layer, the manually refined patch contour and adjusted thicknesses, local mesh refinement in critical regions, and volumetric elements. The objective is to quantify the difference between the optimizer simplification and the derived manufacturable geometry. Also, it gives more information such as deflection behavior and shear stresses in the adhesive layer, which are not available from the optimizer output.

All simulations were performed linear-elastic and evaluated at a defined load level consistent with the settings used in the optimizer.

7.6.1 Simulation Setup

Simulation was performed in *ANSYS Mechanical 2021 R1*. The models were discretized using volumetric tetrahedral elements with a global element size of 3 mm and a growth rate of 1.2. The areas of maximum stress (two shoulders in region B) were locally refined to an element size of 0.3 mm. For the patched designs, the adhesive and patch were refined to a maximum element size of 1.2 mm due to lower shell thickness of the patches.

The weld ends were rounded with a 2 mm fillet radius to better represent realistic weld geometry and to avoid artificial stress concentrations caused by sharp edges.

Boundary conditions were applied at A, B, and C in accordance with the optimizer setup described in section 7.4.2. The load was introduced at B accordingly.

The materials were modeled as linear-elastic representations of S500MC steel and Loctite EA 9514 adhesive. The contact between base sheet, the adhesive layers, as well as between adhesive layers and patches was defined as bonded. In addition, self-contact was defined for the base component in region B, where the two halves meet.

Each model consisted of approximately 1.2×10^6 elements. The simulation output included equivalent (Von Mises) stress, shear stress within the adhesive layers, total deformation, and the deflection of the remote point at B where the load was applied.

7.6.2 Simulation Results

The finite element simulations yielded very high maximum equivalent stresses for all three investigated configurations. Figures A.1a, A.1b, and A.1c in appendix A show the equivalent stress distributions for the three specimen types, including the respective locations of maximum stress. These peak values can be attributed to artificial stress concentrations at sharp geometric transitions and at the load application points.

For a more meaningful comparison, a stress probe was defined in the base material near the weld end in region B at a consistent location for all variants. Although other regions exhibit high equivalent stress as well, these are predominantly compressive or showing steep stress gradients. It was evaluated in the student thesis [HKN24] that the most likely failure location is at the right shoulder of the component near region B, as was introduced in section 7.3. This matches the region where failure occurred in the physical tests, as will be discussed in section 7.8.3. The probe location, exemplary shown for type B, is depicted in figure 7.23.

The computed stresses for all types and their deviation from the reference design (type A) are listed in table 7.6, rows 1 and 2.

The three configurations exhibit different deflection behavior under identical loading. The total deflection and the corresponding relative deviation from the reference design are presented in table 7.6, rows 3 and 4. Figure 7.24 compares the load–displacement behavior of all three types graphically. The patched designs show a considerably lower

stiffness than the unpatched version. The total deformation of all specimens can be found in figure A.3 in appendix A.

The adhesive layer is required to transfer the applied load without failure. The maximum shear stresses within the adhesive are shown in table 7.6, row 5. The maximum shear stress occurs locally at the very edges of the patches. It rapidly decreases moving inwards from the patch edge. The corresponding shear stress distributions for types B and C are shown in figures A.2a and A.2b in appendix A.

Table 7.6: Summary of finite element simulation results for the three specimen types A (3 mm base without patch), B (2.2 mm base + 1.2 mm patch) and C (1.8 mm base + 2.2 mm patch). Von Mises equivalent stress for selected location, adhesive shear stress and total load application point deflection.

Specimen type	A	B	C
max. equivalent stress at critical location	194 MPa	183 MPa	193 MPa
deviation from A	-	-6%	-1%
maximum shear stress in the adhesive	-	18.9 MPa	22.6 MPa
Total deflection of load application point	0.72 mm	0.91 mm	1.06 mm
deviation from A	-	+26 %	+48 %

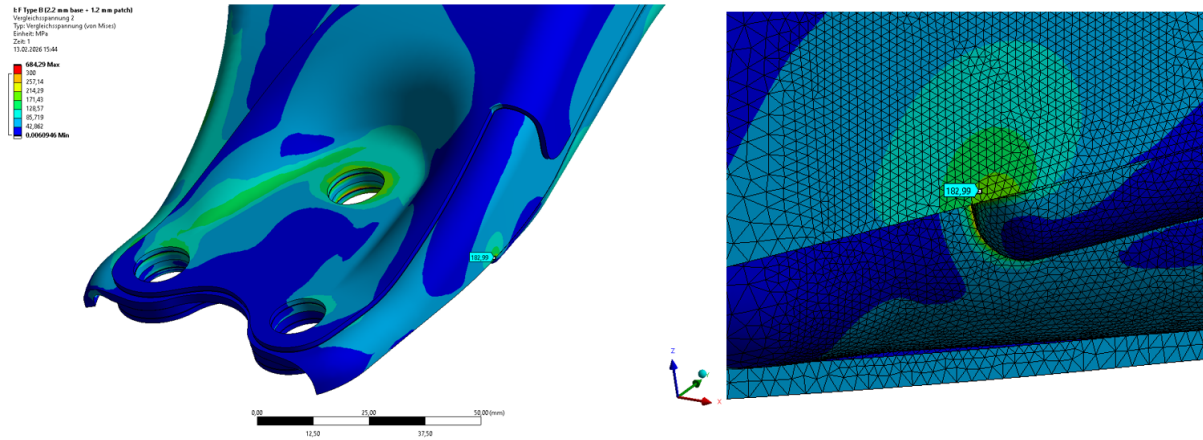


Figure 7.23: Probe on the solved model for type B design. The probe is located on the right shoulder at B, close to the weld end. Equivalent stress depicted. The probe is selected as the local maximum in that region.

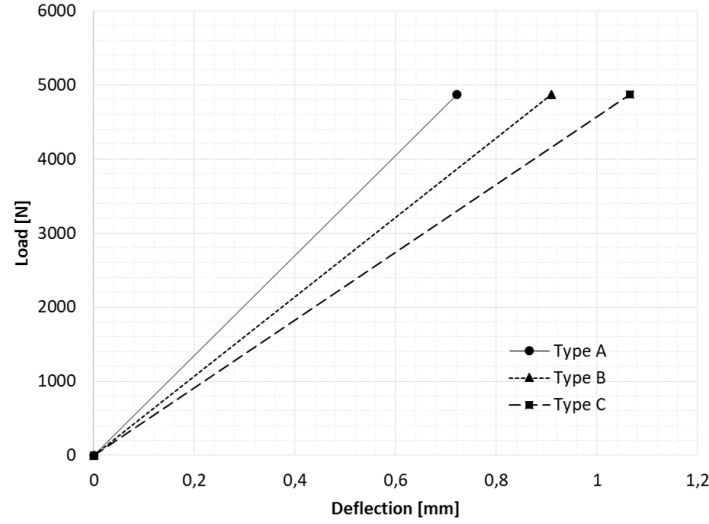


Figure 7.24: Load-deflection behavior of types A, B and C under same load. Specimens type B and C show considerably lower stiffness.

7.6.3 Discussion on Simulation Results

One objective of the simulation was to evaluate the accuracy of the optimizers simplification in comparison to a more sophisticated volumetric model. The solid model shows several local stress concentrations that were not present in the uniformly meshed shell model used within the optimizer. However, the locations of the stress hotspots are consistent between variants. Since the optimizer aims for overall design improvement and is not directly addressing stress concentration anyways, the simplified representation of the reference design using a shell model is considered sufficiently accurate for its intended purpose.

For the patched designs, which exhibited the same maximum stress as the unpatched reference in the optimizer model, the detailed results are only partially conclusive. The volumetric model yields higher maximum equivalent stresses than the reference design, whereas the stress at the defined probe location is similar or slightly reduced. In the optimizer, all calculations were performed on an identical mesh, varying only the element stiffness. In contrast, the volumetric model includes geometric details such as the weld seam, whose geometry depends on the base sheet thickness. This difference in modeling approach can lead to increased variability in local stress results. It therefore remains uncertain whether the agreement at the probe location is coincidental and to what extent the chosen level of abstraction can be considered generally valid also for patched design. The modeling of the adhesive layer did not show a considerable decrease of component stiffness. Shear strain is found to be very low, so the abstraction to monolithic behavior is considered sufficiently accurate. The very thin layer sees low overall shear stresses, with local concentrations still below the permitted value for the selected adhesive.

Another objective of the simulation was the comparison of deflection behavior. It

should be noted that the optimization targeted equal equivalent stress rather than equal stiffness. The detailed analysis shows that the optimized designs differ substantially from the reference in terms of stiffness as they exhibit a noticeable reduction. This can possibly be attributed to unpatched regions, such as the span toward support A, where bending stiffness is reduced due to lower sheet thickness. An optimization constrained to maintain overall component stiffness would likely result in different patch configurations and corresponding changes in stress distribution.

Finally, the simulations were intended to evaluate whether the selected designs are suitable for further experimental investigation. Design variants B and C show performance comparable to reference variant A and are therefore considered suitable for prototype manufacturing.

7.7 Prototype Manufacturing

All three specimen types were manufactured. For serial representation type A (3 mm), rather than using an actual serial component, a dedicated reproduction was made to ensure full control and comparability between all variants with respect to material, welds, and geometry, and to allow mounting to the test fixture given the geometric modifications required at bearings A and C. The specimens type B and C include reduced-thickness base parts and fully bonded adhesive patches on both sides. Figure 7.25 shows the prototype manufacturing process with sub-processes explained in the following.

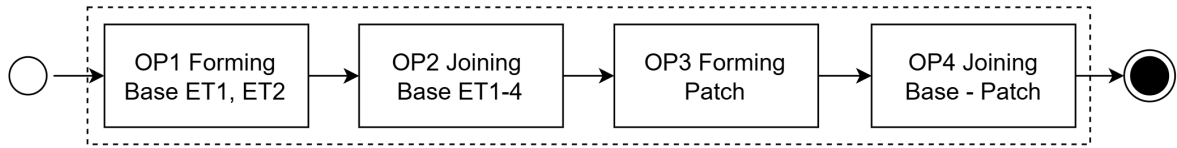


Figure 7.25: Wishbone link prototype manufacturing process. First, the main parts ET1 and ET2 are formed. They are joined with the bushings ET3 and ET4. Then, after forming it, the patch is attached to the base part.

7.7.1 Forming of the Main Parts

The main part consists of the two cold-formed shells, ET1 and ET2. From the CAD geometry, their forming process is derived by incremental forming simulation with *STAMPACK Advanced Software*. Different to most other forming simulation programs, STAMPACK can use volumetric elements to more accurately predict formability of thick shells or complex forming operations where sheet thickness may play a significant role such as domes or sharp bends. It uses not the commonly known FLD, as introduced earlier in section 2.1.2, that works only for plain strain but a more sophisticated model for arbitrary strain, the triaxial failure diagram (TFD), see [RSFO16]. Also within STAMPACK, simultaneous forming of several sheet layers can be analyzed. For the prototype build process, only very few prototypes are required. Focus for manufacturing are few and simple manufacturing tools, whereas material used or manufacturing effort are of lower interest. It was possible to simulate forming operations for different sheet thicknesses with one tool only at 3 mm gap width by using spacer sheets, which was proven to work as expected. Consequently, the forming and intermediate trimming operations could be performed in the same manner for all specimen types A, B and C. See figure B.1 in appendix B for TFDs after complete two-step forming for ET1 and ET2.

Figure 7.26 shows the forming process steps. The process is identical for both halves, ET1 and ET2.

Initially in OP1.0, from rectangular metal sheets, blanks in 3 mm, 2.2 mm and 1.8 mm as well as 0.8 mm and 1.2 mm spacers were laser cut. All blanks obtain the same outer

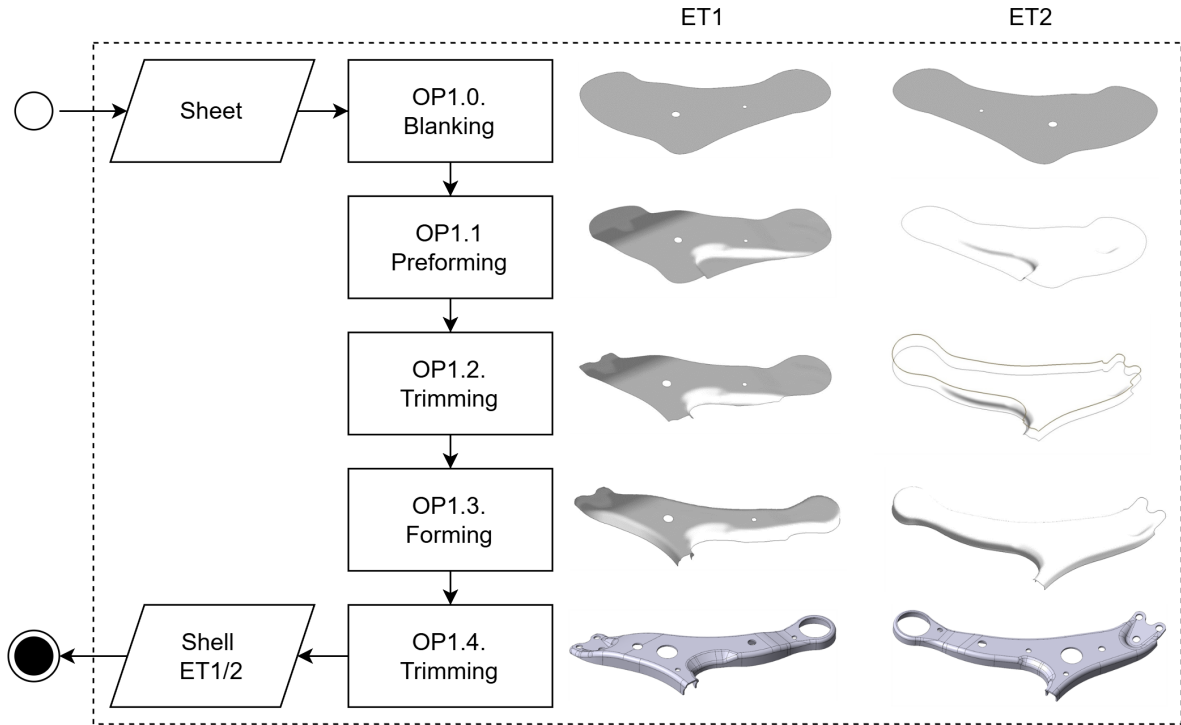


Figure 7.26: Wishbone link manufacturing operation OP1, forming base ET1 and ET2. The sub-steps are the same for both shells, consisting of forming and trimming operations.

contour and two positioning holes placed in regions which will later be cut off. Then, in OP1.1, the blanks were preformed, see figure 7.27. As testing showed, forming without any blankholder force, so-called crashforming, showed the best results. An oversized blank, much larger than the formed region, is required to prevent wrinkling by restricting mass flow into the die. The excess material would, however, lead to excessive thinning in the next forming operation as it hinders material flow, so it was manually trimmed off by tracing a 3D-printed template and rough cutting on the band saw in OP1.2.

This step showed to be critical, as often, notches were cut into the sheet's edge on inner radii. This led to several failed parts in the next forming step and should have been done by milling or laser cutting instead.

In OP1.3, the final forming step was performed, see figure 7.28. Here, the whole outer contour was formed down, requiring high press force and creating a lot of discarded specimens due to excess thinning or edge cracking. While this was acceptable for prototype production, a serial process would require additional intermediate forming steps to ensure process robustness.

Subsequently in OP1.4, the final contour and holes were trimmed by milling. For doing so, a milling fixture forced the parts into shape, as springback from OP1.3 led to high deviations even after springback compensation. See appendix B, figure B.2 for the springback after forming, and figure B.3 for the milling fixture.

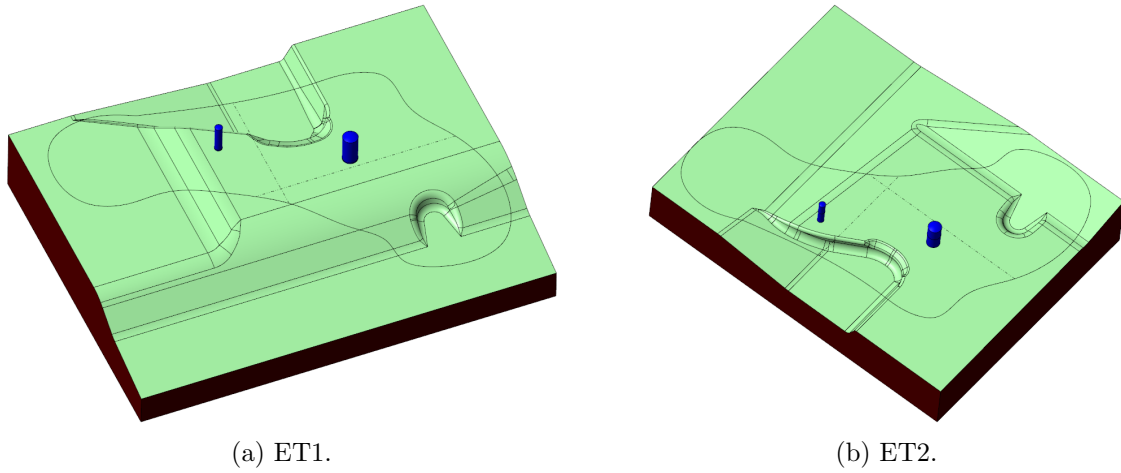


Figure 7.27: Wishbone link preforming step OP1.1 lower die with blank outline. Here, the overall curvature, contour at region B, and the shoulder between A and C are pre-formed on a large blank to achieve uniform thinning in those regions and counteract wrinkling.

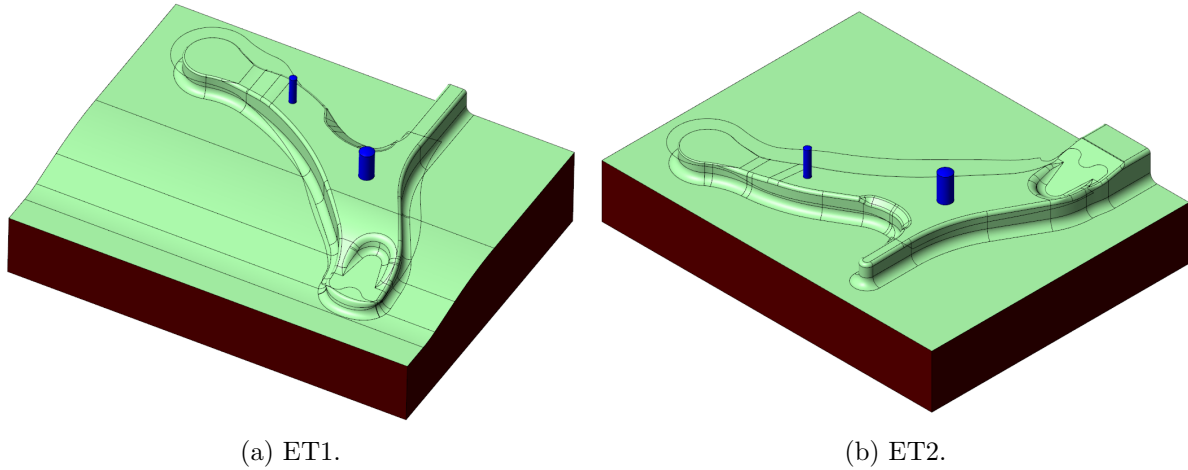


Figure 7.28: Wishbone link forming step OP1.3 lower die with blank outline. After trimming the parts, here they get formed to their final shape. The preformed contours from OP1.1 help align the sheets. The whole edge is formed down.

7.7.2 Joining of the Main Parts and Bushings

In step OP2, the two main shells ET1 and ET2 are welded to each other and ET4's sleeve by MAG-Welding. ET3 and ET4 are Toyota serial parts. For the prototypes, a welding fixture was specifically made to keep shape consistent, see figure 7.29. Parts ET1, ET2 and ET4 sleeve were positioned on the fixture. Due to springback, they had to be forced into a defined location. Then while fixed, they were tack-welded (step OP2.1). Subsequently, they could be taken out and be welded thorough on a table (step OP2.2). Figure 7.30 shows a resulting part in 2.2 mm thickness variant.

Finally, in step OP2.2, bushing ET3 was pressed into the opening at A and bushing ET4 into its sleeve.

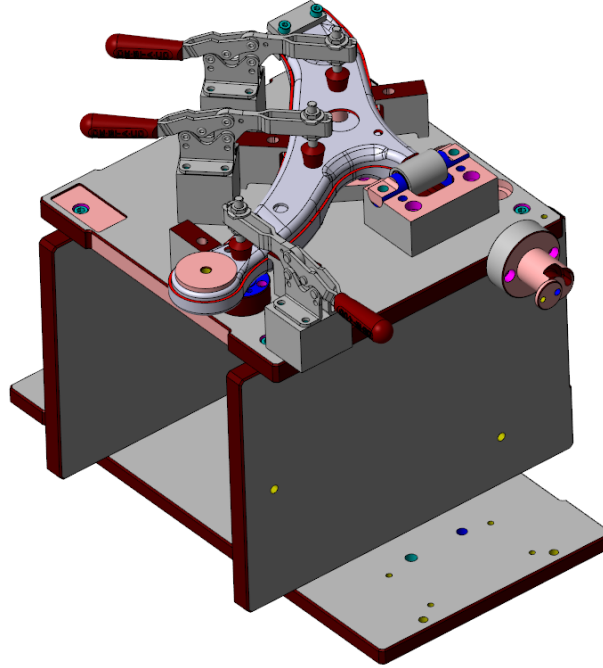


Figure 7.29: OP2.1 Wishbone link welding fixture. Again, the two halves are forced to target position, counteracting any shape deviations. That way, the wishbone link may spring back slightly after welding, but most deviations are canceled out from permanently joining the two parts and the cross sectional area of all prototypes is consistent.



Figure 7.30: Wishbone link after OP2.2 welding, 2.2 mm variant.

7.7.3 Patch Manufacturing

For specimens type B and C (base thicknesses of 2.2 mm and 1.8 mm), the patches are manufactured following the design simplifications as discussed in the introduction to this section. Both patch variants share the same outer contour, while their shapes follow the respective base parts, which differ slightly due to the base thickness variation. The patches were produced through laser cutting (OP3.0), forming (OP3.1), and trimming to the final

shape (OP3.2). Forming was done with a simple die. Figure 7.31 shows one half of the forming tool with the final patch shape. Due to relatively low forming complexity and low material thickness, only one forming step was required, in contrast to the two-step process for the base part. Trimming was done manually by use of a 3D-printed template.

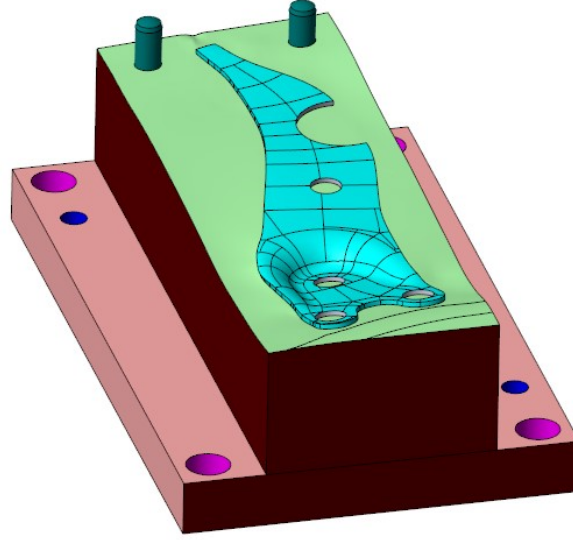


Figure 7.31: Wishbone link 2.2 mm variant patch forming tool. Only a single-step forming is required.

7.7.4 Patch Joining

As the last prototype manufacturing step, patches are joined to the 2.2 mm and 1.8 mm base components. Liquid adhesive Henkel Loctite EA 9514 was used as selected in chapter 6.2.4. The desired gap width is $t_a = 0.2 \text{ mm}$ as it leads to the highest shear strength, see figure 6.2. To maintain gap width, steel spacers with the same thickness were embedded in the adhesive layer. They decrease the overall bonding area, so they were placed in locations of overall low shear stresses expected, which is in a safe distance to the border and in low-stress regions, see figure 7.32. The amount of glue added was double the required volume to cover the whole surface at a constant 0.2 mm gap thickness. This was to fill any possible voids between the two mating pieces, which may exist for example in tightly bent corners. Any excess adhesive was squeezed out of the gap when pressure was applied.

The patches covered in adhesive were placed on the base component. Pressure was required to ensure proper bonding across the gap maintained by the embedded spacers. To achieve uniform pressure on the whole component, vacuum pressing was used. The base part with adhesive and patch was sealed in a vacuum bag and vacuumized to $\leq 0.9 \text{ bar}(g)$. Vacuum bags used were fluoropolymer bagging film which can be used up to $260 \text{ }^\circ\text{C}$.



Figure 7.32: Adhesive applied to the lower 1.2 mm patch for specimen B1. 0.2 mm steel spacers can be seen embedded within. The amount of glue is 4 ml, corresponding to an overall bonding thickness of 0.2 mm with a 100 % surplus to fill any possible gaps larger than 0.2 mm, e.g. in tight radii.

Curing was performed at 180 °C for 60 minutes, following the adhesive manufacturers recommendation. The vacuum was maintained throughout the whole curing process, to prevent any leakages of the bag or expanding gases within the bag from decreasing the vacuum pressure. Figure 7.33 shows the curing set-up and 7.34 shows the resulting component in 2.2 mm variant.



Figure 7.33: Wishbone link adhesive curing set-up.

The joining for the 2.2 mm variant went largely successful. Excessive adhesive was squeezed out the gap around both patches perimeter, indicating a sufficient amount of adhesive and successful push down with the vacuum setup. One patch has a shoulder running alongside the wishbone link from B to A, as was depicted in figure 7.22. Only there, the gap was not consistent and insufficient bonding was expected.

For the 1.8 mm variant with 2.2 mm patches, larger deviations in the geometry of the mating surfaces led to a more inconsistent gap, even after applying the vacuum.



Figure 7.34: Wishbone link prototype after curing.

Additional clamps fixated the patch overall, but some gaps were still visible after curing. Larger areas of insufficient bonding or locally very high adhesive thickness were expected.

7.7.5 Prototype Build Results

The prototypes built are to be used for a comparison of different designs under load with the focus on mass reduction. In theory, the lightweight versions should weigh 83.6 % for the 2.2 mm base + 1.2 mm patch type B and 73.8 % for the 1.8 mm base+ 2.2 mm patch type B with respect to the unpatched 3 mm type A, considering the mass of the main shell without bushing ET3 added. Due to rounding of the material thickness, smoothing of the patch shape and mass of the adhesive itself, the real weights came to 2.68 kg for type A, 2.27 kg for type B (84.7 % of A) and 2.12 kg for type C (79.1 % of A).

7.8 Experimental Validation

For validation, both the rebuilt serial components and the patched variants were tested under design load conditions. For doing so, a component test rig was developed to fit onto the same general-purpose hydraulic test bench previously used for adhesive fatigue testing, as shown in figure 6.5.

The validation started with static testing to compare the load/deflection behavior of the three specimen types, both within the elastic range and beyond it, representing a misuse load case. This is to evaluate the differences in deformation behavior between patched and unpatched specimens, the load transfer capability of the adhesive, and the associated failure mechanisms.

Subsequently, fatigue testing at the design load was performed to assess the suitability of the patched design for cyclic loading. Due to time constraints and limited specimen

availability, the fatigue tests represent an initial experimental campaign and do not yet permit a statistical evaluation.

7.8.1 Test Rig Design

The test rig was designed so that it can transfer loading at point B in various directions in the XY-plane to simulate lateral, longitudinal or combined load cases relative to the wishbone link's mounting orientation in the vehicle. Support is designed to recreate the simulated constraints. Instead of using the serial parts rubber bushings, to keep rigidity movement low for increasing test efficiency and accuracy, they are swapped for stiffer bearing supports, similar to how it was done in the patch optimizer. Figure 7.35 shows the fixture and coordinate directions referred to in the following.

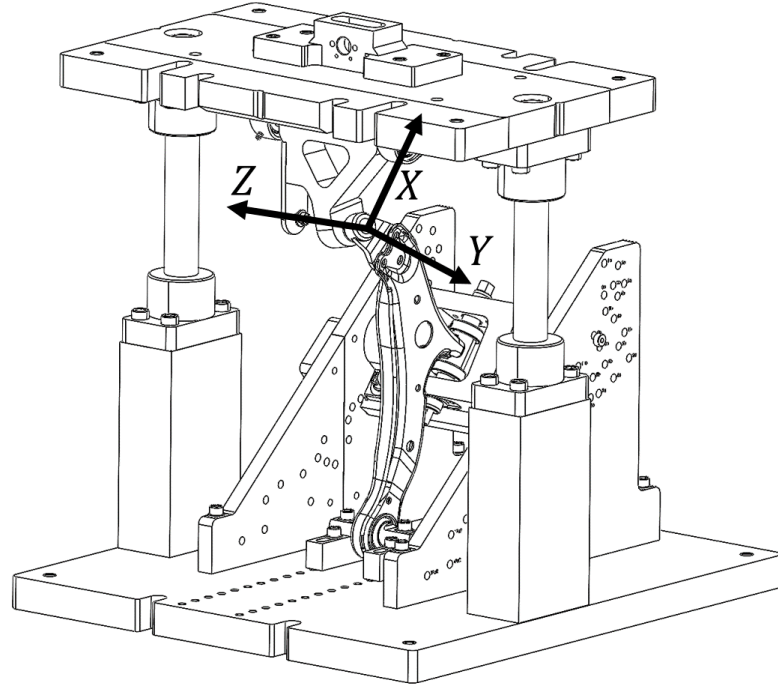


Figure 7.35: Wishbone link test rig. The link can be mounted on the bushings at A and C. Load is applied on B. The constraints are congruent with the finite element simulation performed. To be able to change direction of load application, the whole link can be panned around Z at load application point B.

The test rig is built into a frame with linear guiding pillars, only allowing vertical movement in force direction for the upper crossbeam to prevent any lateral force damaging the test bench's drive. To transfer force to point B while keeping deflection perpendicular to it free and restricting any buckling, a swing is attached to B. That way, B is free to translate in the XY-plane, supported in Z and all rotational DoFs are free. Deviations in force direction if the swing deflects are neglected as they are below 1° for the expected link deformation. At A, a spherical bearing is inserted into the component, fixing all translatory DoFs. At C, the bushing sleeve can move along its axis via a shaft free to

slide and rotate in outer bushings. They again are sitting in a swing that can rotate around Z at block bearings.

The inner shaft allows translation and rotation with respect to X. The outer bearings allow rotation around Z. Ideally, also rotation around Y would be allowed to completely match the simulation case. However, moment around Y is negligible as this constraint showed no difference in FEA.

Point B stays fix for different angles of force application. Instead, the wishbone link, including ball joint attachment, can rotate in XY-plane around B and bearings A and C can be fixed in different angles. A can be bolted onto the base plate in different locations, while B moves upwards into various mounting positions. Angles can be set over a range of 90° at a resolution of 5° , see figure 7.36.

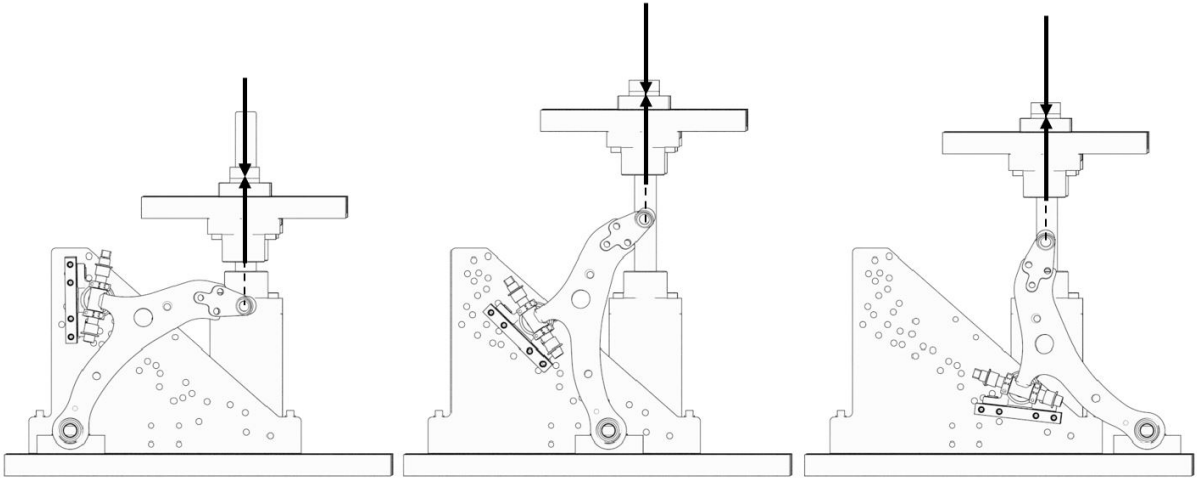


Figure 7.36: Test rig 0° , 45° and 90° testing orientations [HKN24]. 0° refers to a load case acting against driving direction of the car (in negative X direction). 90° refers to a lateral load case. Intermediate values can be used for combined load cases.

Some modifications to the fixture had to be made after initial try-out. The upper part of the fixture had a tendency to wedge itself in its guides. The modification uses only the right-hand guide and a centering of the piston above the loadline. See figure C.1 in appendix C for the final design.

7.8.2 Static Test Setup

The tests conducted aim to investigate the behavior of the wishbone link at a realistic load case to see, if the 2.2 mm and 1.8 mm patched variants show a comparable behavior to the 3 mm unpatched serial part - within the simulated linear elastic range for comparison to the theoretical model, but also beyond plastic deformation to investigate general deformation behavior and strength. The representative force vector as derived in section 7.3, see equation (7.1), is applied to the specimens and the resistance to deformation and specimen behavior is recorded.

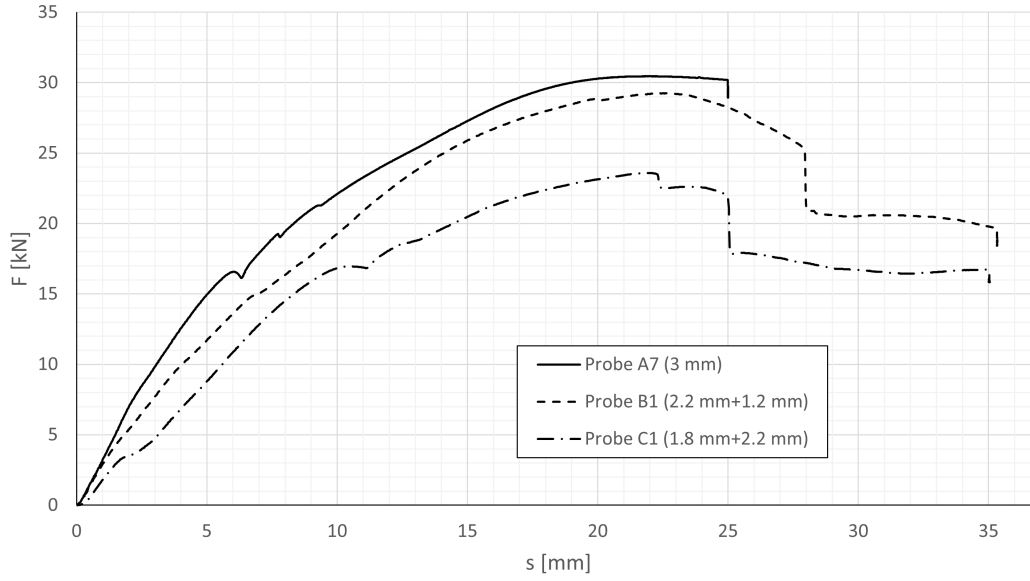
For the fixture, the closest setting to the simulated force vector is selected. The latter is angled 3° from the component's X-axis. On the fixture, the selected setting is at position 2 being 5° counter-clockwise rotation around Z.

A ball bearing is attached to the wishbone link at B with three bolts. A spherical bearing is inserted into the large hole at A, the ball joint attached at B and a cylindrical bearing is pressed into the bushing at C. See figure reffig:TestSpecimen in appendix B) for a specimen with bearing elements attached. The fixture is aligned in vertical direction so that stress-free mounting of the specimens is possible. The specimen is first attached at C and the ball bearing at B is inserted into the swing. Then, the vertical position is finely adjusted to allow stress free attachment of the bearing at A to the fixture's frame. With all mountings secured, the specimen is ready for testing.

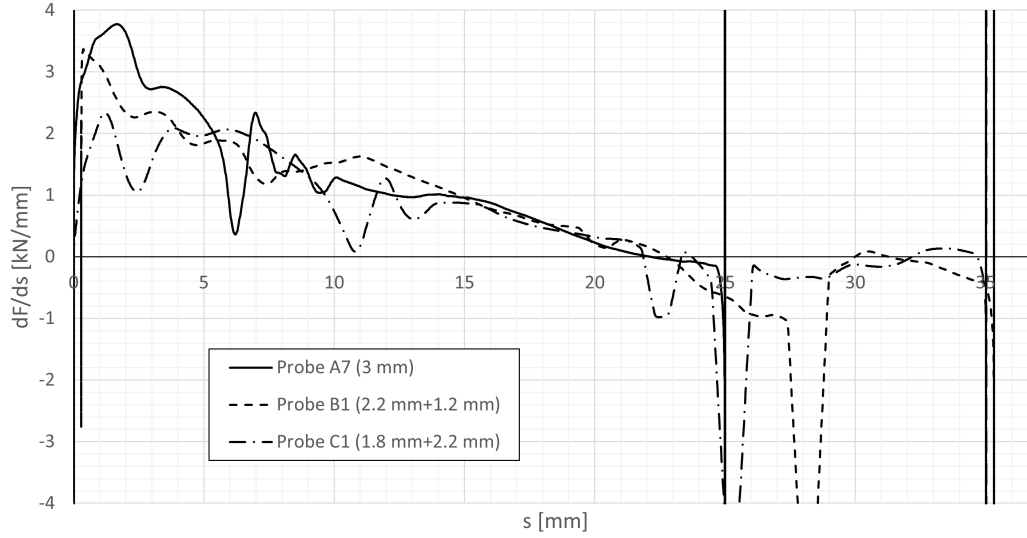
The test bench is capable of applying 50 kN of downwards pressure by a hydraulic piston. It can run either force-controlled, recording displacement, or displacement controlled, recording the force required. In this test, displacement is controlled as plastic deformation and a decrease in required force when the part fails is expected. The displacement is set to 35 mm downwards from the unloaded position over a duration of 22 seconds. A force sensor, in line with the hydraulic piston, records the force in a resolution of 0.01 s . Cameras are set up to record the deformation of the component or delamination of patches.

7.8.3 Static Test Results

Three static tests were performed: The 3 mm recreation of the Toyota serial part (specimen A7), a decreased thickness of 2.2 mm base thickness with 1.2 mm patches (specimen B1) and a 1.8 mm base thickness with 2.2 mm patches (specimen C1). The force-displacement curves are shown in diagram 7.37a with respective gradients, describing the component stiffness, in 7.37b. Deformation and failure of the specimens after testing can be seen in figure 7.38. The results show the initial linear-elastic force-displacement response, followed by yielding and the subsequent load-bearing behavior up to final failure.



(a) Force F over displacement s for static testing of three different specimens. The curves were slightly smoothed for improved visibility.



(b) Gradients dF/ds of the above force-displacement curves. The curves were smoothed for improved visibility. In the range upwards of 25 mm, where stiffness of the samples decreasing abruptly, the values occasionally exceed the diagram scale.

Figure 7.37: Force-displacement curves and respective gradients for three tested specimens - unpatched 3mm version A7 and patched versions B1 and C1 in 2.2 mm and 1.8 mm, respectively. An onset of local yielding can be observed at $F_{y1,A7} = 7 \text{ kN}$, $F_{y1,B1} = 4.8 \text{ kN}$ and $F_{y1,C1} = 3.3 \text{ kN}$, beginning of major plastic deformation at $F_{y2,A7} = 16.5 \text{ kN}$, $F_{y2,B1} = 15 \text{ kN}$ and $F_{y2,C1} = 17 \text{ kN}$ and maximum force at $F_{max,A7} = 30.4 \text{ kN}$, $F_{max,B1} = 29.2 \text{ kN}$ and $F_{max,C1} = 23.5 \text{ kN}$. Variants A7 and B1 behave comparably, while C1 shows a lower stiffness, initially high resistance to major plastic deformation but ultimately fails at much lower force than the others.



(a) Specimen A7, 3 mm base thickness without patch, after static testing with 25 mm deflection.



(b) Specimen B1, 2.2 mm base thickness with 1.2 mm patch, after static testing with 35 mm deflection.



(c) Specimen C1, 1.8 mm base thickness with 2.2 mm patch, after static testing with 35 mm deflection.

Figure 7.38: Deformation/ failure of the specimens after static test in the region of ball joint attachment. From left to right: Overview, frontal view on the attachment region, view from the right side to the attachment region. Specimen A7 shows mayor plastic deformation and localized thinning. Specimen B1 additionally shows a failure of the material around one mounting hole, a crack in the highly-stressed right-side shoulder of the base material, and slight patch delamination in the vicinity of high base layer deformation. Specimen C1 shows similar failures as B1, with a larger area of delamination.

Specimen A7, 3 mm base variant with no patch, begins to initially show high approximately constant stiffness, which is the slope of the force-displacement curve, indicated by a linear curve and constant Df/dx . Then, a slight decrease in stiffness occurs, seen from a decrease of the curve slope, from displacement $x_{y1,A7} = 2 \text{ mm}$ at force $F_{y1,A7} = 7 \text{ kN}$. The deformation continues at a still approximately linear curve, so lower but constant stiffness, for a short period. A drop in the curve then occurs after $x_{y2,A7} = 6.1 \text{ mm}$ at $F_{y2,A7} = 16.5 \text{ kN}$. After a brief increase in slope, the gradient gradually decreases again. Finally, after $x_{max,A7} = 21.7 \text{ mm}$ at $F_{max,A7} = 30.4 \text{ kN}$ the maximum force recorded is reached. The curve begins to slope down, until the recording stops after 25 mm . After testing, the specimen shows high plastic deformation close to the ball joint attachment at B. The whole head at B twisted under load, with high plastic deformation on the shoulders. However, no cracks but only local necking formed, as the test was aborted early.

Specimen B1, 2.2 mm base + 1.2 mm patch, initially exhibits a comparable stiffness to B1, but a first kink in the curve can be observed earlier after $x_{y1,B1} = 1.5 \text{ mm}$ at $F_{y1,B1} = 4.8 \text{ kN}$. In the following, stiffness is decreased, indicated by a shallower curve, but remains about constant until it drops after $x_{y2,B1} = 6.5 \text{ mm}$ at $F_{y2,B1} = 15 \text{ kN}$. The point is less distinct compared to B1, and the drop in stiffness is smaller. Remarkably, stiffness increases again gradually, until after $x_{max,B1} = 22.2 \text{ mm}$ at $F_{max,B1} = 29.2 \text{ kN}$ the maximum load is reached. Stiffness decreases again, before the part finally fails after 27 mm , indicated by a sudden drop in force. Visually, the specimen shows high plastic deformation of the whole B region and a crack opening on the shoulder. The material around one of the ball joint mounting holes on the front also partially failed.

Specimen C1, 1.8 mm base + 2.2 mm patch, starts with a lower stiffness than A7 and B1 and earlier kinks in the force-displacement curve after $x_{y1,C1} = 1.8 \text{ mm}$ at $F_{y1,C1} = 3.3 \text{ kN}$. After a transition phase of approximately 1.5 mm , stiffness increases again and stays relatively constant until after $x_{y2,C1} = 10.3 \text{ mm}$, $F_{y2,C1} = 17 \text{ kN}$ is reached. A plateau without load increase over approximately 1.2 mm can be observed, before the stiffness increases again and stays relatively constant until the maximum load is achieved after $x_{max,C1} = 21.7 \text{ mm}$ at $F_{max,C1} = 23.5 \text{ kN}$. Shortly after the peak, at 22.3 mm , a sudden small drop in force occurs. Mayor part failure can be observed shortly after at 25 mm indicated by a sudden drop in force as with specimen B1. Visually, also specimen C1 shows high plastic deformation. A crack occurred in the highly stressed shoulder of B. Delamination of the upper patch occurred in the vicinity of highest local deformation. Like specimen B1, it also shows failure around a mounting hole.

Another specimen was tested exploratively. It was a 1.8 mm base variant with no patch. It's resulting force-displacement curve can be found in appendix C, figure C.3.

7.8.4 Static Test Discussion

In the following, the test results for specimens A7, B1 and C1 will be first analyzed individually, with explanations for the behavior given, and then compared to each other with a conclusion of what can be derived from the observations.

Specimen A7, 3 mm base variant with no patch, showed the predicted behavior for the serial geometry. initially, the part behaves linear elastic, with some small settling and supporting effects engaging early on. The stiffness is consistent with the expected behavior from the linear-elastic simulation, see section 7.6 and figure 7.24. The recording of a stiffness decrease at $F_{y1,A7} = 7 \text{ kN}$ can be interpreted as local plastic deformation occurring, which is quickly supported by the surrounding material. At $F_{y2,A7} = 16.5 \text{ kN}$ a "yield plateau" can be observed, which is a sudden displacement without any force increase. It can be observed in tensile testing for many materials such as low-strength steels with high carbon content, but is not expected for the low-carbon high strength steel used here, see [DWYG25] for yield plateau criteria and [STGF14] for S500 stress-strain curves. The plateauing effect here will thus likely originate from local plastic failure or instability.

Even after plastic deformation, the specimen held up well until maximum load is reached. The geometry seems to be designed to support plastic deformation without immediate collapse - as it is desired for wishbone links that should deform highly before rupture. Unfortunately, the test was aborted after $x = 25 \text{ mm}$ before the part was able to break. However, localized thinning on the one shoulder and thickening on the other shoulder at B is visible, consistent with the stress hotspots predicted in the FEA.

Overall, it can be concluded that this design behaved as expected, failure occurred in the expected locations and the results validate the assumptions from the FEA model.

Specimen B1, 2.2 mm + 1.2 mm, starts to behave linear elastic on the first millimeters of deflection. It is slightly decreased compared to specimen A7, but within the expected range from the simulation. Shortly, some dips in the force-displacement curve indicate local yielding, which however is immediately supported by the surrounding material. The second dip in the curve at $F_{y2,B1} = 15 \text{ kN}$ indicates higher local yielding with more surrounding material engaging in support. It could be, that the first dip relates to only very local support effect within the base material, subsequent load shift and the patch fully engaging on the second dip. Considering that this load shift has to be transferred through the adhesive, which has a lower shear modulus than steel, the slope of the curve is not as steep as expected for a monolithic steel sample. Remarkably, the deformation continues to behave highly linear, indicating a successful load shift into the patch. After a non-distinct transition to the range of plastic deformation, the maximum resistance is reached at $F_{max,B1} = 29.2 \text{ kN}$. It is followed by a sudden drop in resistance. This may

indicate local failure already occurring. From the recordings, one mounting hole started rupturing at this point, which could be the root cause of this. Finally, the part collapses completely with a sudden stiffness drop and a crack visible. Still, the force required for deformation stays on a high constant level indicating a favorable behavior for an overload case, in which the wishbone link should deform highly before complete failure.

In summary, this design shows some room for improvement at the geometry surrounding the mounting holes, but overall exhibits a desirable behavior for a wishbone link.

Specimen C1, 1.8 mm + 2.2 mm, exhibits a decreased initial stiffness, which, while also shown in the simulation, was not expected as drastically. The first dip in the force-displacement curve occurs very early and distinctive at $x_{y1,C1} = 1.8 \text{ mm}$. This is earlier than would be expected for local yielding. Potentially this results from low stiffness bonding between base and patch layer, allowing settling effects to occur. After the initial drop in stiffness, it increases again and stays on a comparable level to before, indicating a successful load-shift to the patches. Mayor plastic deformation occurs at a load level of $F_{y2,C1} = 17 \text{ kN}$, which, considering the first early indication of yielding is very high and shows that this patch design is capable of supporting high-stress regions. However, from that point onset, the overall stiffness is drastically decreased, as the maximum strength is only 6.5 kN higher than the yielding point. The design does not seem to be able to support the structure well in case of mayor deformation. The minor drop in stiffness at the maximum can be observed from the recordings to relate to the rupture around the attachment hole, and the subsequent larger drop to failure of the base layer and delamination of the adjacent region.

To summarize, this specimens initial settling, small difference between onset of yielding and maximum load, as well as the sudden rupture do not indicate a favorable behavior. However, the high yield load shows the capability of high load transfer to the patches. Likely, inconsistencies in the manufacturing such as high bond thickness from shape deviations or insufficient bonding led to a worse performance than the design may be theoretically capable of. This impression is reinforced by the explorative comparison to the unpatched 1.8 mm variant, which shows almost no difference in initial behavior. Finally, the material around the mounting holes prematurely giving in shows that there is potential for improvement also in that region, which was not yet considered.

Comparison Based on Static Test

Based on the assumption that the force-displacement behavior for design A (specimen A7) represents the desired behavior of a serial wishbone link, the lightweight alternatives B and C (specimens B1 and C1) can be compared to it - to address the suitability of this thesis' approach for mass reduction of wishbone links, and generally, the ability of load-shift into the patches via the adhesive.

Regarding the suitability for a wishbone link, from specimen B1, the design with a 2.2 *mm* base layer and 1.2 *mm* patch exhibits a behavior comparable to the serial design. Although showing a slightly reduced initial stiffness as expected from the simulation, the behavior remains largely similar. It even exceeds it in stiffness after mayor plastification onset, indicating it being more capable of keeping stiffness after some moderately severe overload event. The reduced maximum strength is within the expected deviation based on rounding down of sheet thickness during prototype material selection.

From specimen C1, limitations of the prototype manufacturing became apparent. Settling effects and early failure of the mounting holes reduced result conclusiveness. Nevertheless, compared to the serial design, a partially equally stiff behavior and even higher load at onset of mayor yielding indicates the possibility of reaching a suitable behavior - optimization to the gluing process and optimization of the patch geometry at the mounting holes may be required. Thus, with more optimization of the patched designs, it can be assumed that the serial designs performance can be reached or even exceeded with reduced mass designs by this thesis' approach.

Regarding the proof of the proposal, whether patching of load paths can shift load from high-stress regions, the results were conclusive. Both patched designs showed a high perseverance in upkeep of stiffness also after local plastic deformation.

The mass optimized geometries were computed based on a linear elastic model. They were tuned to have the same maximum internal stresses within the elastic range. Component stiffness and plastic deformation were not calculated to match the reference. Also, a monolithic model of the patched areas was used, disregarding a lower shear modulus or local shear failure of the adhesive, or any incomplete bonding of the patches. The optimization approach merely assumes that a load path optimization will also benefit component stiffness, plastic deformation behavior and part failure. These things considered, the results verify the underlying assumptions as they show a stiff behavior under load for an extended range of deformation, no preliminary failure of the adhesive and the ability to support yielding regions.

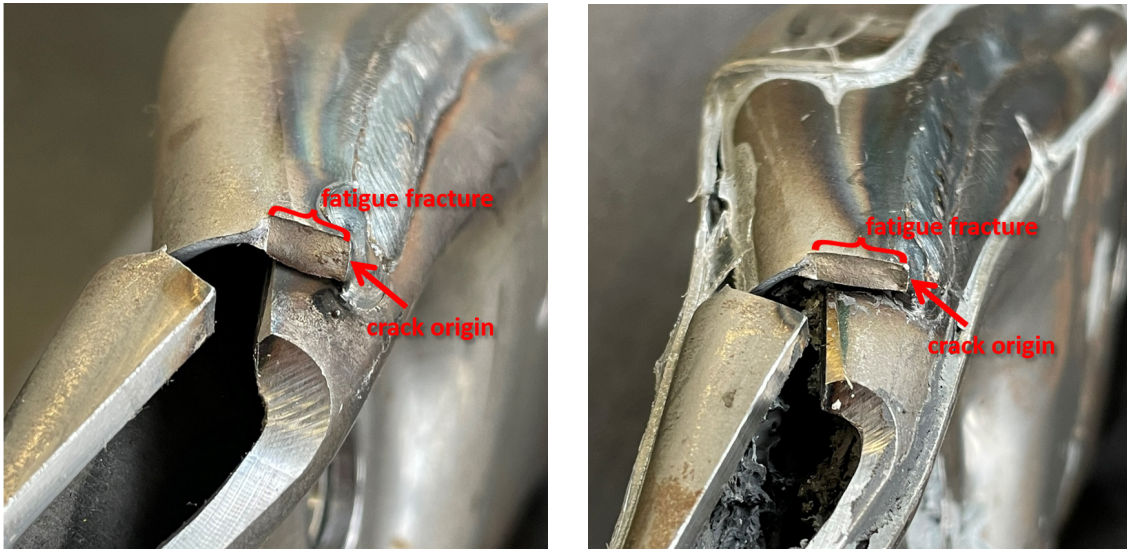
7.8.5 Fatigue Test Setup

After determination of the static strength, fatigue testing with two load levels followed. The test aimed to test low cycle fatigue at around $n = 50\,000$ cycles. Initially planned to be performed with same magnitude as simulated, which is a pulsating load of $F_U = 4860\text{ N}$ in negative Z direction, it was slightly increased after static testing to $F_U = 6000\text{ N}$, accounting for higher static strength of the specimens than originally expected. After the first fatigue tests, it was further increased to $F_U = 8000\text{ N}$ to decrease run time, as specimens survived $n = 1\,000\,000$ cycles and testing had to be aborted before failure could be observed. Loading was pulsating with slight pretension at $R = 0.1$. Testing

was continued until failure occurred. Indication of failure was either a deflection above the expected limit, which was set to 6 mm, or cracks detected during recurring visual inspections. The test setup was kept consistent with the static test with respect to load direction and fixture configuration.

7.8.6 Fatigue Test Results

First fatigue testing was performed for the specimens type A (3 mm without patch) and type B (2.2 mm with 1.2 mm patch). for Type A, 4 specimens were tested at $F_U = 6000\text{ N}$. They survived between 180 000 cycles with observed fracture and 1 000 000 cycles where testing was aborted without any fracture indication. To produce more meaningful results from the limited amount of specimens available, after hitting 1 000 000 cycles for two consecutive test, the load was increased to $F_U = 8000\text{ N}$. There, the specimen failed after 177 000 cycles. For the first available specimen type B, testing was continued at that same level. The specimen survived 110 000 cycles before a beginning fracture was observed. Figures 7.39a and 7.39b show the fractured specimens after cycling.

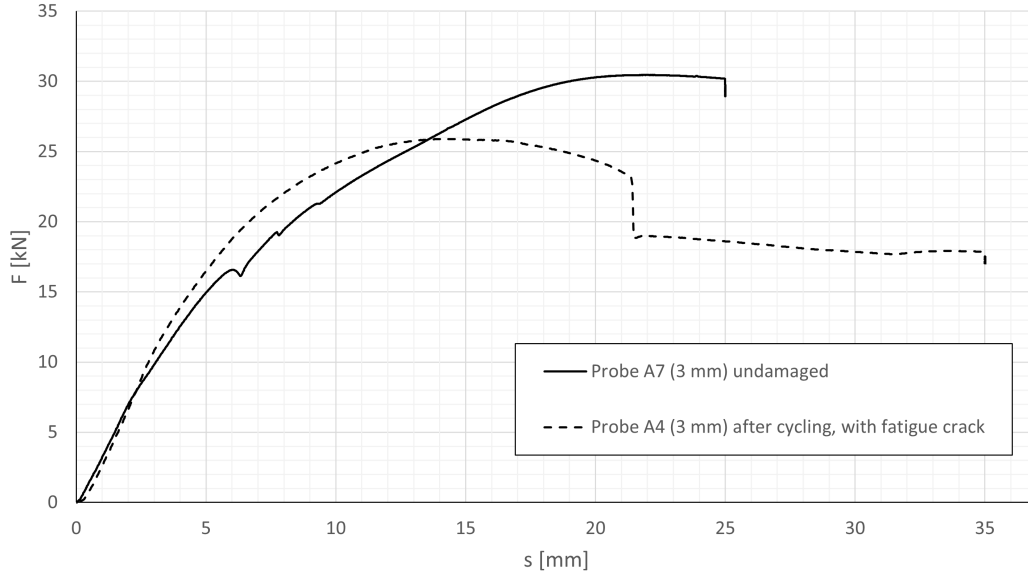


(a) Specimen A4 (3 mm without patch) fatigue fracture after 177 000 load cycles at $F_U = 8000\text{ N}$ and subsequent static overload. The fatigue region is highlighted.

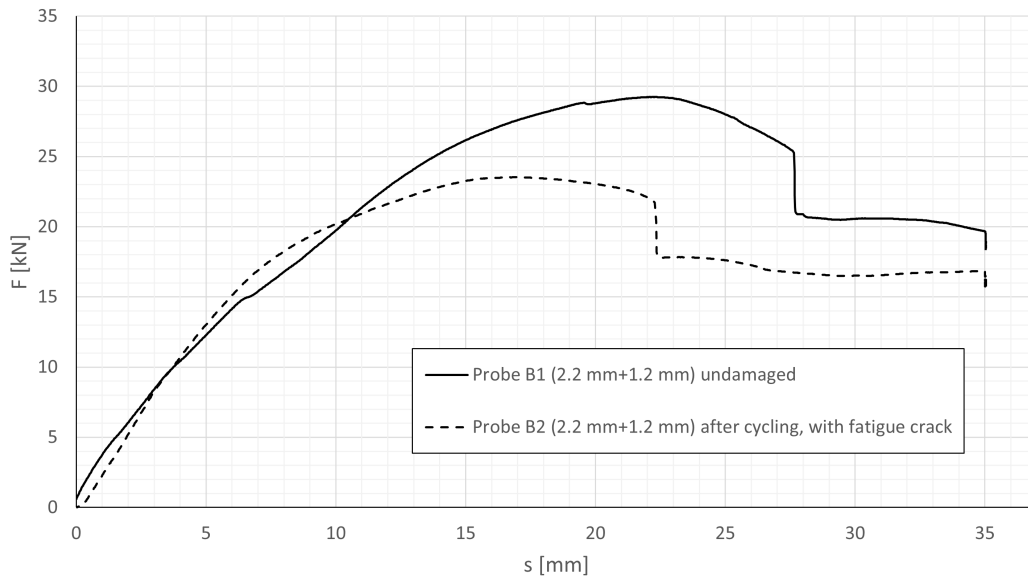
(b) Specimen B2 (2.2 mm with 1.2 mm patch) fatigue fracture after 110 000 load cycles at $F_U = 8000\text{ N}$ and subsequent static overload. The fatigue region is highlighted.

Figure 7.39: Wishbone link fatigue fractures after cyclic testing and subsequent overload. Both specimens show a similar crack origin and fatigue fracture length.

In addition, the specimens were examined for their behavior after initial fracture, to assess whether they would still respond favorably under an overload. This evaluates whether a worn component could still provide a certain level of safety, as required for load-bearing chassis components, as discussed, for example, in [SBKW23]. The force-displacement curves, compared to undamaged components, can be found in figures 7.40a and 7.40b. For both specimens, a decrease in ultimate strength can be observed.



(a) Specimens type A (3 mm without patch) force-displacement curves for an undamaged state (A7) vs after beginning fatigue (A4). The pre-damaged specimen shows an initially stiffer behavior, but lower ultimate strength before failure.



(b) Specimens type B (2.2 mm with 1.2 mm patch) force-displacement curves for an undamaged state (B1) vs after beginning fatigue (B2). The pre-damaged specimen shows an initially stiffer behavior, but lower ultimate strength before failure.

Figure 7.40: Force-displacement curves comparing an undamaged component with one featuring a fatigue crack of approximately 5 mm length present before static overload test.

7.8.7 Fatigue Test Discussion

The preliminary fatigue testing performed on the first specimens shows no clear indication. The type A specimens exhibit a much higher fatigue strength than anticipated, surviving one million cycles at the load level originally intended to cause failure at 50 000 cycles, and still exceeding this life at the increased load level. This behavior is also observed for the first tested type B specimen, which survived nearly as long.

From the static tests, it was observed that first yielding for all specimen types occurs close to or below the expected load level. At the same time, the tests revealed a strong support effect from the adjacent structure on local yielding. Although this effect was considered in the application of the FKM Guideline, as introduced in section 7.3, it may have been underestimated. Alternatively, other effects may not have been properly accounted for, such as stress concentrations at the weld toe or heat influence in the base material.

The crack origin and crack growth direction are consistent and as expected for both specimen types. In both cases, cracks initiate within the weld heat-affected zone near the weld toe and propagate perpendicular to the assumed maximum principal stress, which is parallel to the part edge.

The static overload test performed after fatigue failure was conducted to compare the behavior of a pre-damaged component with that of a new, undamaged specimen. The results show that the presence of an initial crack reduces the ultimate strength by 15 % for type A and 20 % for type B specimens. Remarkably, the initial component stiffness is not reduced and in some cases even exceeds that of the undamaged specimens. This may be attributed to settling effects or local plastification that already occurred during cyclic loading, causing the load to be carried immediately by a stiffer structure.

In summary, the first fatigue tests show comparable fatigue behavior between unpatched specimens of type A and mass-reduced, patched specimens of type B. Further tests are required to confirm these initial observations.

Chapter 8

Summary

Summarizing the previous chapters, a novel method for determining appropriate patch geometries was introduced, including a description of its core mathematical operations. The method is based on the idea that patches are most effective when used to create efficient main load paths by locally stiffening the structure along a line from load application to support. Its main functions are stiffness computation, pathfinding with Dijkstra's algorithm, patch discretization, and variation analysis of patch width and thickness combinations. Without the use of any local concept such as stress up until the final evaluation, the method enables geometrical optimization that promotes a natural flow of forces and thus provides potential for mass reduction.

A major advantage of this method is that the resulting geometries are close to manufacturable solutions without requiring extensive rework, as the output directly provides uniform thickness distributions and coherent patch geometries. By extending load path analysis with a pathfinding algorithm, the method enables the explicit definition of geometrical restrictions, such as areas that are less suitable or unsuitable for patching. This allows the direct incorporation of manufacturing and design constraints while still maintaining significant mass-reduction potential.

Based on the method, a functional software tool was developed to find optimal patches for a given meshed structure and defined single load case. Written in MATLAB with an ANSYS interface created, the software performs stiffness computation, result interpolation, pathfinding, patch growth and variation analysis automatically based on finite element analysis and exports the result data. The software proved capable of reliably computing patch optimizations with relatively little manual input.

Bonding of patches using an epoxy-based structural adhesive demonstrated suitable mechanical properties. Static and fatigue strength were sufficient to enable reliable load transfer into the patch, and the application process is compatible with established manufacturing procedures for structural automotive sheet metal components. From a comparison of liquid adhesives and an adhesive tape regarding application, static strength and fatigue strength, it could be concluded that adhesives can exhibit suitable properties

for bonding patches under high static and fatigue load applications. It was possible to select a liquid adhesive for a prototype build process. However, further investigation is required to establish a stable and reproducible application and curing process, as significant deviations in mechanical properties were observed during initial testing.

On a serial wishbone suspension link, the patch optimizer software demonstrated considerable mass-reduction potential through patching for a realistically constrained model under representative loading. It was shown that the software can handle restrictions on the component's weld seam efficiently, as it achieved a mass reduction comparable to that of an unrestricted case. A refined finite element simulation confirmed the validity of the optimizer's abstraction level and supported the expected structural behavior of the optimized designs. Two mass-reduced prototype designs and a serial component reference were manufactured and an associated test procedure was developed. The optimization yielded a considerable mass reduction of 410 *g* with respect to the reference component for the moderately mass-reduced wishbone link and 560 *g* for the further mass reduced design while maintaining maximum stress at a comparable level to the reference design.

Static load testing confirmed the suitability of the mass-reduced designs for the considered load case and demonstrated the general suitability of the method for load redirection. Some discrepancies in the design and manufacturing process partially limited the conclusiveness of the tests, as some deviations between specimens could not be explained completely.

Initial fatigue tests also showed comparable behavior. An expanded investigation of the fatigue strength is required for a more robust comparison between mass-reduced and serial-part designs and to thoroughly assess the suitability of the approach for dynamically loaded components.

Overall, the method proposed in this thesis enables manufacturable lightweight designs for sheet metal components. A dedicated software tool was developed and validated through simulation, prototype manufacturing, and experimental testing, which yielded considerable mass reductions for a wishbone link. The combination of low manufacturing effort, resulting from the uniform patch thickness approach with explicit consideration of geometrical restrictions, and substantial mass reduction underlines the method's potential for serial application.

8.1 Outlook

Several steps of the method, although consistent with structural mechanics principles, were developed empirically. Whether U^* and the gradient translation to path definition are the best metric for defining the most efficient load paths is up to debate. A further in-depth investigation on load path theory may lead to a more profound model.

While the method presented shows potential and some real-world application seems

promising, there are some steps still required to transfer this into a fully working method and software for industry application. The consideration of multiple load cases must be enabled, as components typically transmit different load magnitudes and directions, often through different application points. For now, patch and base material remain the same as initially. There is potential in using harder to form higher strength steels or other materials for patches as they can be geometrically simpler than the base component. Also, at lower thickness, some higher strength materials for the base part may be possible to use for manufacturing. Different thicknesses for separate patches could further increase mass reduction potential.

With regard to the pathfinding method, some optimization could decrease computation time. Instead of U^* , the element stiffness matrix may be used directly to evaluate the stiffest path. Whether other methods able to create discrete patches, such as evolutionary algorithms, could result even more efficient mass utilization could be investigated. Also, a more intelligent, perhaps evolutionary variation analysis on patch width and thickness may be quicker and able to find precise results in a broader range.

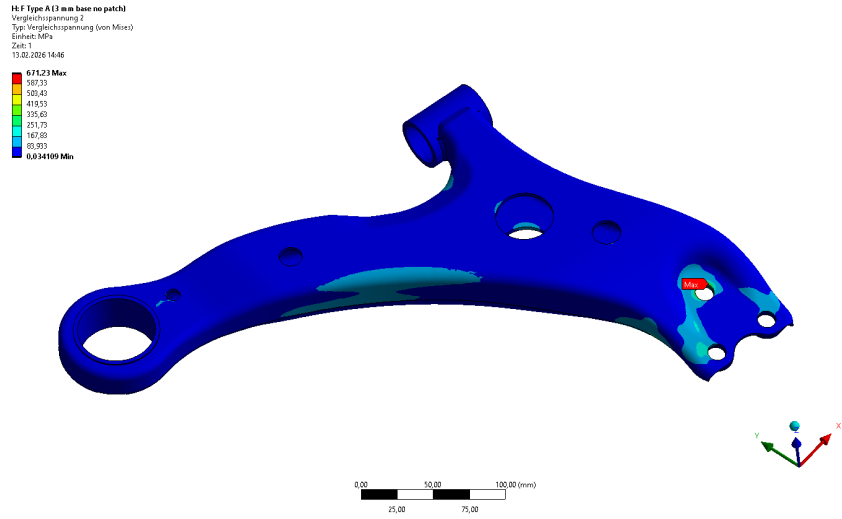
For software, a full integration of the FEA solver may increase performance further. Integrated meshing, constraint definition and consistent GUI could enhance user experience. A better path density function would increase precision and counteract mesh-dependency. The modeling of multi-layer shell elements could improve the representation of the adhesive layer.

At the time of submission, experimental validation of the manufactured prototypes has not yet been fully completed. Further optimization of the prototype build process may yield improved results for static testing. Continued fatigue testing will provide a more comprehensive assessment of long-term fatigue performance and clarify whether the current configuration is suitable for highly dynamically loaded components, particularly with regard to the adhesive and fatigue mitigation in the base material due to load shift into the patches.

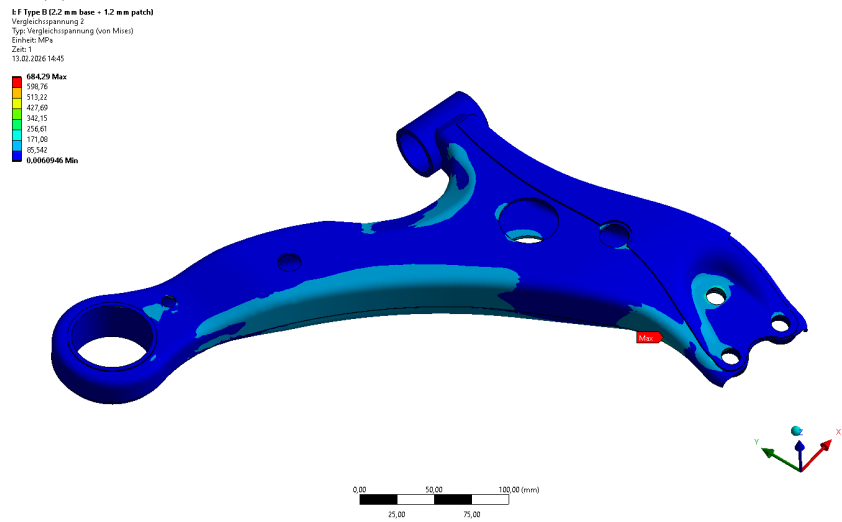
The method and associated manufacturing approach were developed with cost-effectiveness in mind. Whether competitive lightweighting costs can be achieved and whether the approach will gain industrial acceptance must be evaluated in real serial production scenarios.

Appendix A

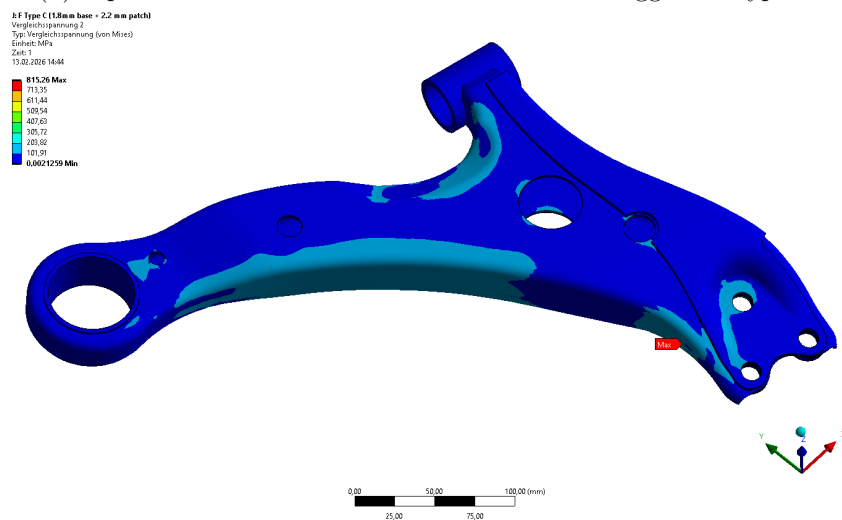
Virtual Validation Figures



(a) Equivalent stress distribution and maximum flagged for type A.

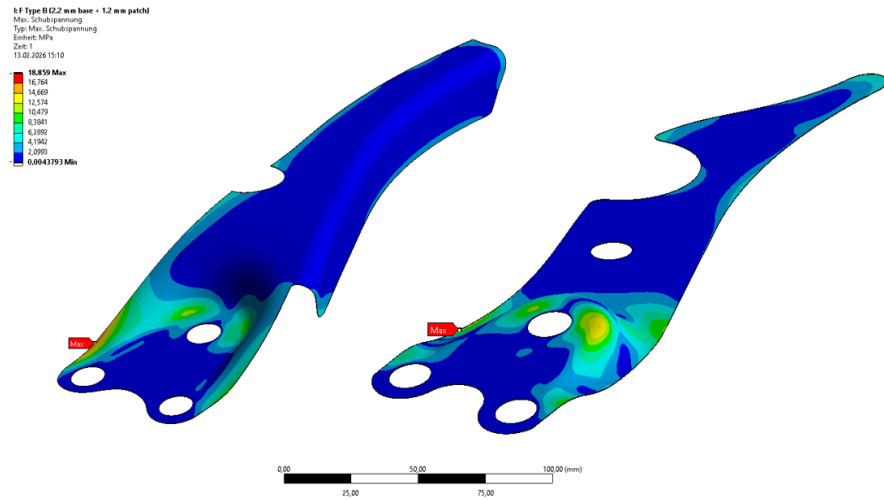


(b) Equivalent stress distribution and maximum flagged for type B.

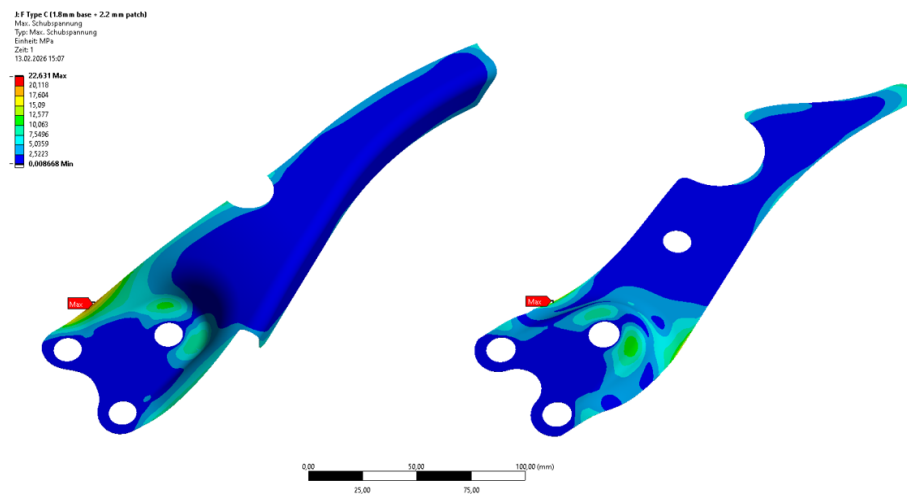


(c) Equivalent stress distribution and maximum flagged for type C.

Figure A.1: Maximum Von Mises equivalent stress distribution for the three design types. Maximum location is flagged.

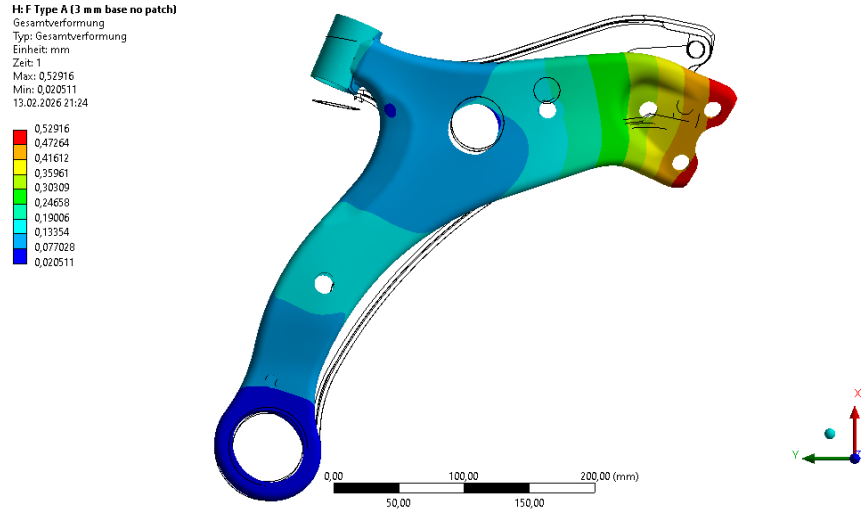


(a) Shear stress distribution in the adhesive layers and maximum flagged for specimen type B. Left: top adhesive layer. Right: Bottom adhesive layer.

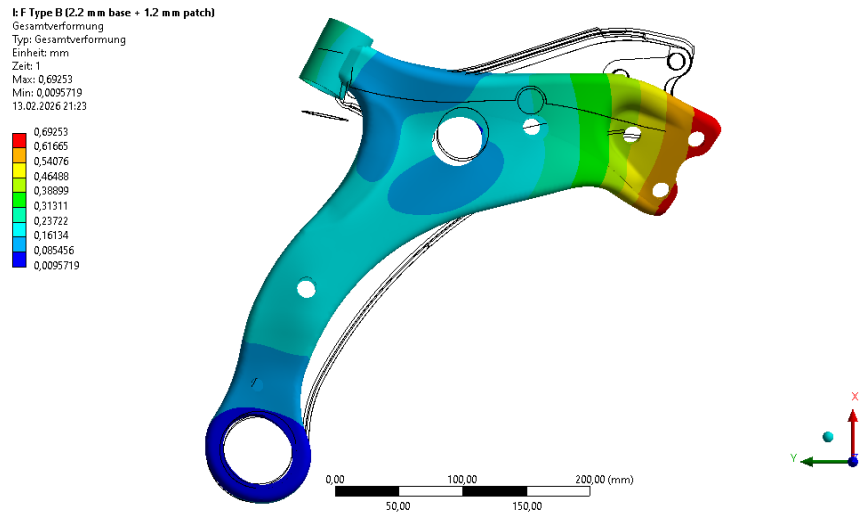


(b) Shear stress distribution in the adhesive layers and maximum flagged for specimen type C. Left: top adhesive layer. Right: Bottom adhesive layer.

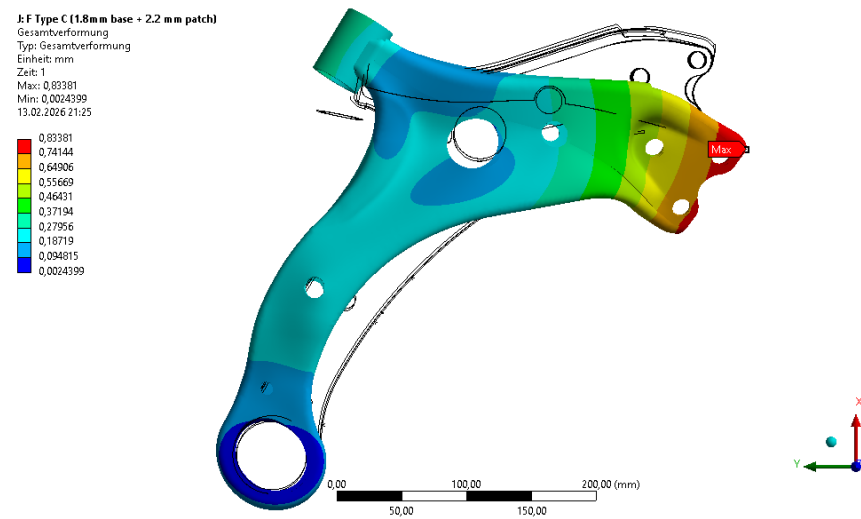
Figure A.2: Shear stress distribution for the two patched design types. Maximum location is flagged.



(a) Total deformation of design variant A.



(b) Total deformation of design variant B.



(c) Total deformation of design variant C.

Figure A.3: Total deflection of the variants under design load. Linear-elastic analysis performed in ANSYS. Deformation scaled by a factor of 100 for visualization.

Appendix B

Prototype Figures

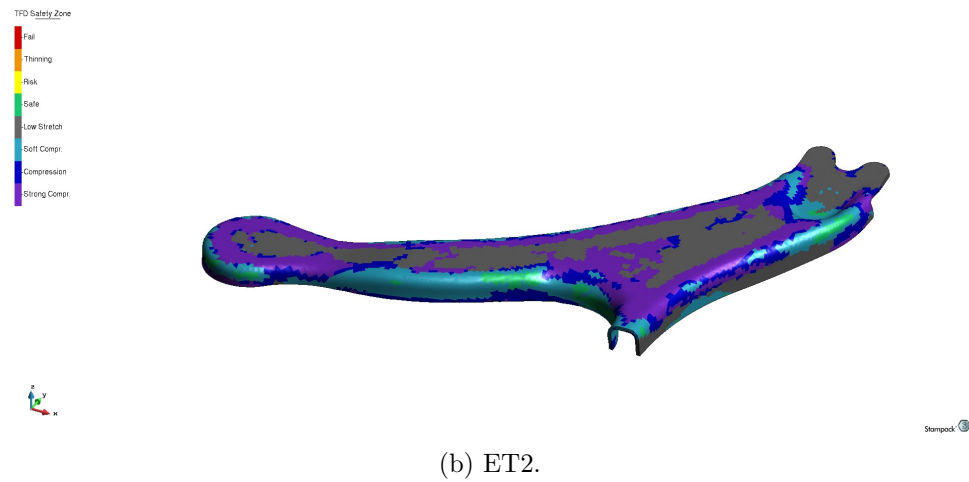
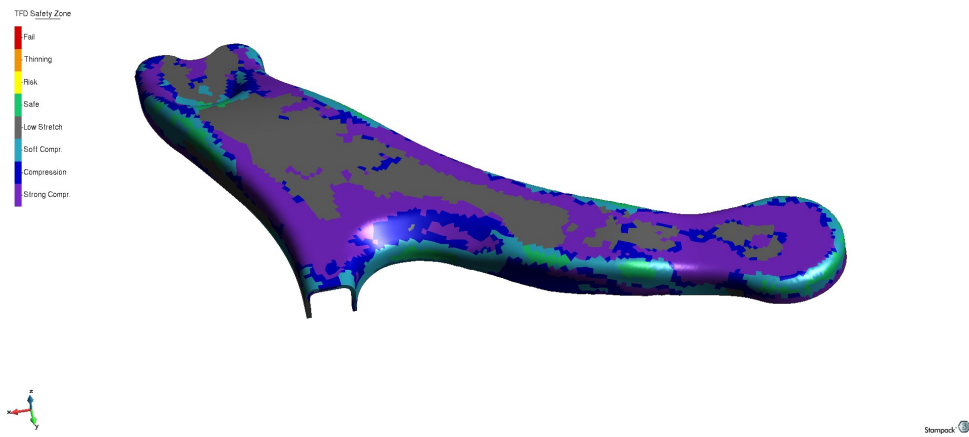


Figure B.1: Wishbone link triaxial failure diagrams (TFD). The tight bending radii along the edges induce thickening of the regions. This leads to visible wrinkles. Some regions are stretched, but still in a safe manner due to the two-step forming process.

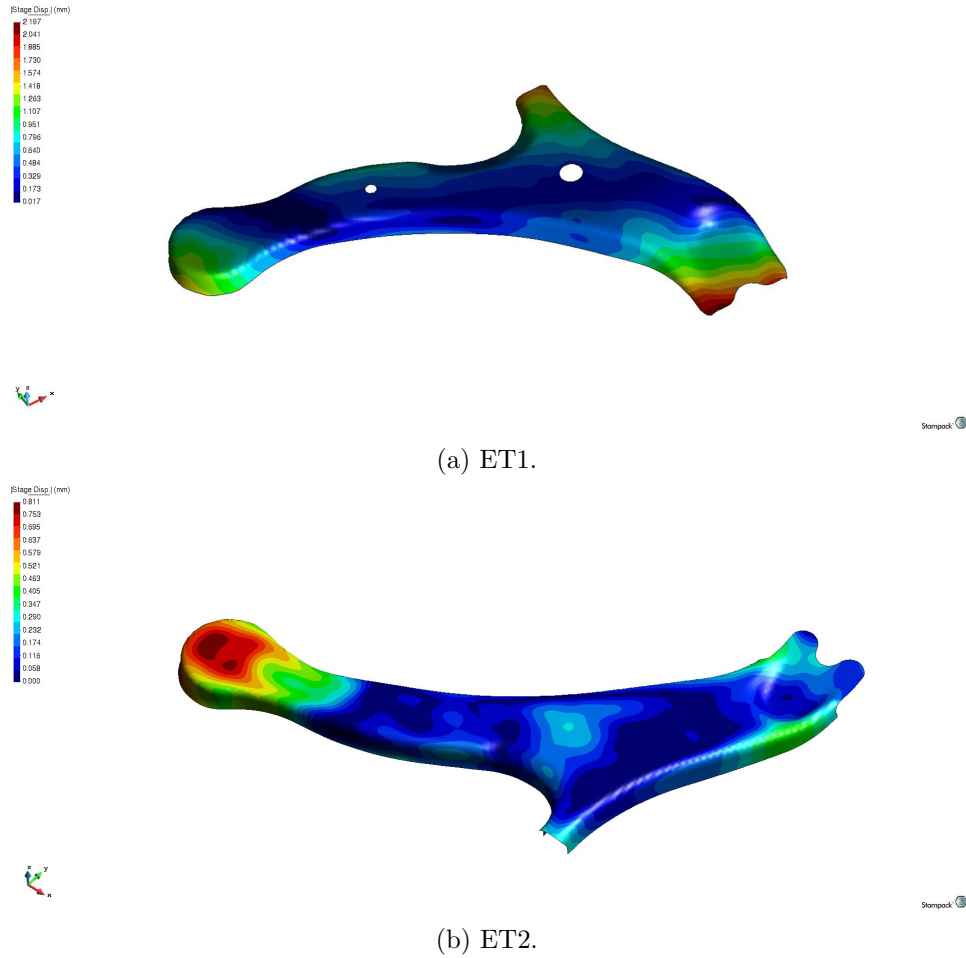


Figure B.2: Wishbone link springback after forming OP1.3. Considerable springback can be observed in region C.

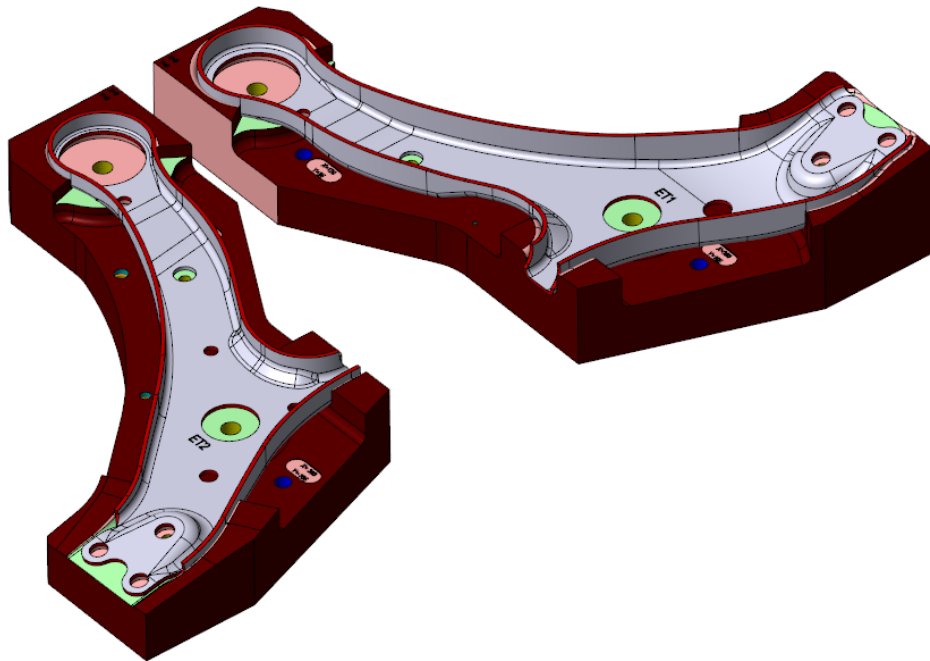


Figure B.3: Wishbone link trimming step OP1.4 milling fixture for both ET1 and ET2. The parts get clamped down to their target shape, so any minor deviations are compensated. Then, the whole edge is milled down to final dimensions.

Appendix C

Experimental Validation Figures

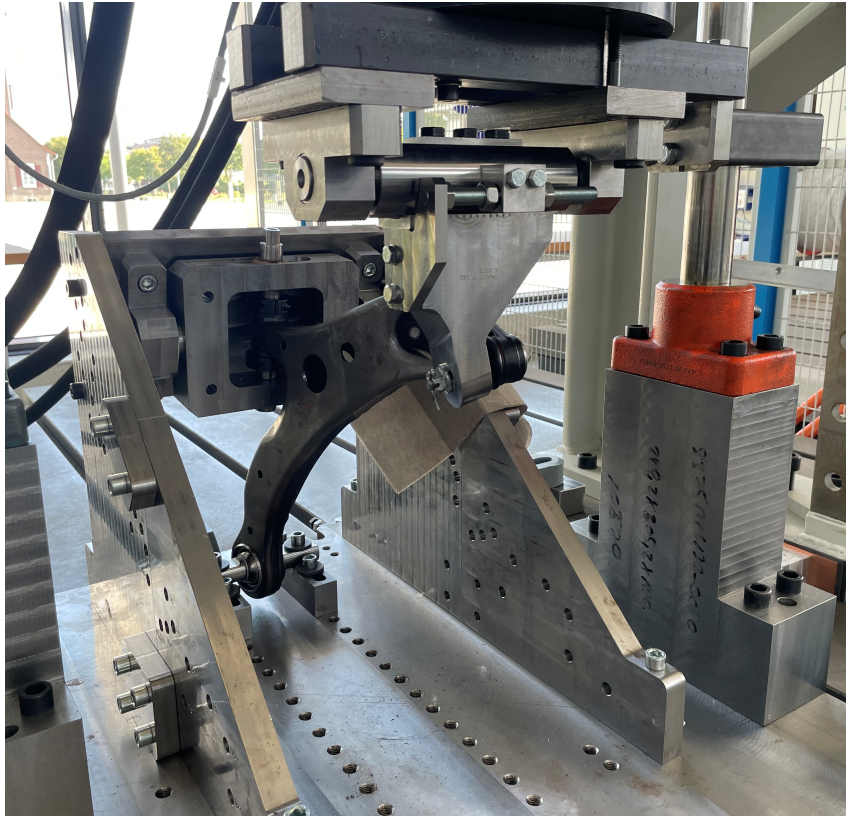


Figure C.1: Final test rig design with a 3 mm specimen mounted. In the front-right, the attachment of the load piston to point B can be seen, while in the rear-left, the mountings to the fixture's frame at A and C are visible.

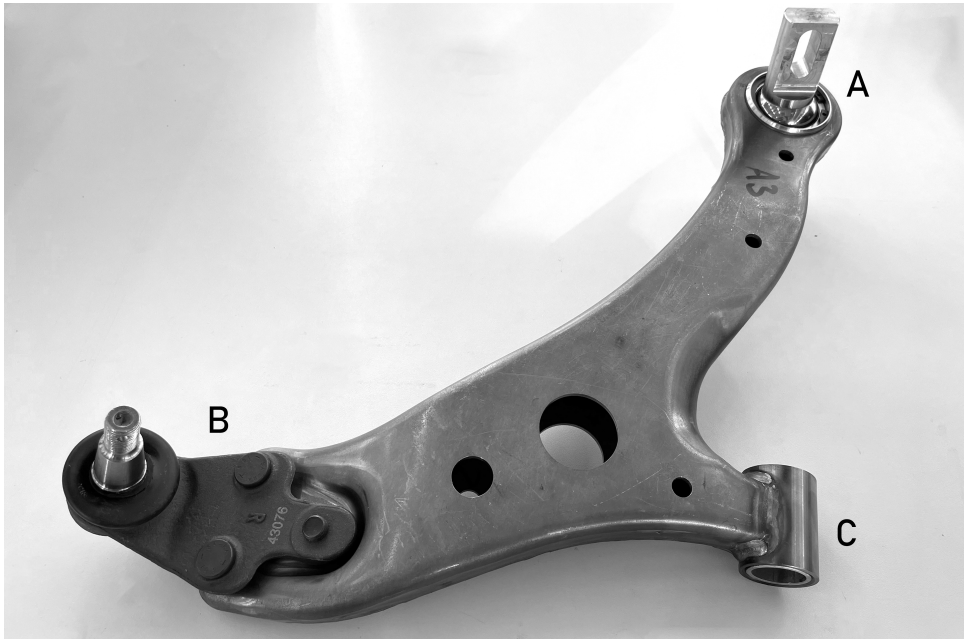


Figure C.2: 3 mm specimen with bearings at A and C and ball joint at B attached. Bearings at A and C substitute rubber bushings from the serial parts. The ball joint at B is an aftermarket component, geometrically equivalent to the respective OEM part.

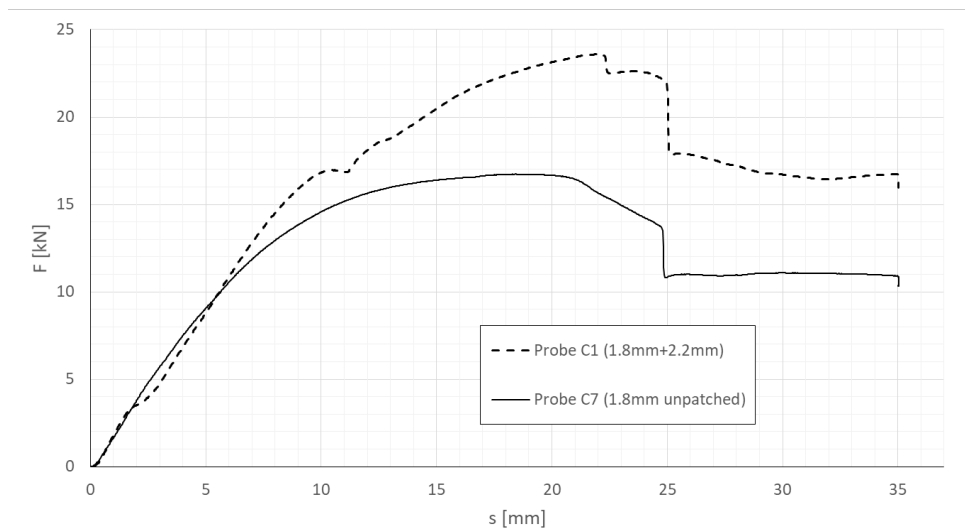


Figure C.3: Force-displacement curve of unpatched specimen C7 with 1.8 mm base thickness compared to the patched probe C1. It can be seen that initial stiffness is comparable, while C1 shows some irregularities early on. Later after onset plastic deformation, C7 stiffness decreases rapidly and it reaches a much lower maximum load before failure.

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Methodological Remarks

Declaration on the Use of Generative AI Tools

During the preparation of this work, AI-based language tools (ChatGPT) were used exclusively for linguistic refinement, providing suggestions regarding grammar, wording, and sentence structure for selected paragraphs. All AI-generated suggestions were reviewed individually and implemented only where deemed appropriate.

The outline, structure, and draft of this thesis, as well as all intellectual content, analyses, and conclusions, were developed independently by the author. Full responsibility for the content of this thesis lies solely with the author.

Erklärung zur Zitation von Inhalten aus studentischen Arbeiten

In Ergänzung zu meinem Antrag auf Zulassung zur Promotion in der Fakultät für Maschinenbau der Universität Paderborn erkläre ich gemäß §11 der Promotionsordnung und unter Beachtung der Regelung zur Zitation studentischer Arbeiten:

Die von mir vorgelegte Dissertation habe ich selbstständig verfasst, **und ich habe keine anderen** als die dort angegebenen Quellen und Hilfsmittel benutzt. Es sind **Inhalte** studentischen Ursprungs (studentische Arbeiten) in dieser Dissertation enthalten.

Ich habe die verwendeten Arbeiten entsprechend der Regelung *Zitation aus studentischen Arbeiten in Dissertationen* zitiert.

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