Optimization of the Inner Boom of a Telescopic Forklift (Telehandler) Using Finite Element Analysis: A Review

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Abstract — The telescopic handler (telehandler) relies on a multi-section boom whose inner sections carry the rated load at long reach. These sections must be strong and stiff yet light, because boom mass reduces the useful load chart and affects machine stability. Finite element analysis (FEA), increasingly combined with parametric modelling, surrogate models and evolutionary algorithms, is now the standard route to checking and improving such booms. This paper reviews eighteen studies and one standard published between 2003 and 2026. They cover telehandler and forklift-truck booms together with closely related work on truck cranes, tower cranes and aerial-lift booms. The work is organised by load cases, failure modes and stability context, model idealisation and contact treatment, static, buckling, modal and fatigue analysis, and optimization strategy (cross-section, plate thickness, surrogate-assisted multi-objective, reliability-based and data-driven). Reported mass reductions are about 8.4% and 12.5% for two recent forklift-type booms and 11.6% for a tower-crane boom. Few studies address the telehandler inner boom directly. Few model sliding-pad contact explicitly, buckling is rarely a constraint in the optimization loop, fatigue is assessed but never optimized in the sources consulted, strain-gauge validation is uncommon, and the dynamic load factors applied range from 1.15 to 1.5. From these findings the paper identifies eight research gaps and proposes a nine-step workflow for inner-boom optimization.

Keywords: Telehandler; Telescopic Forklift; Inner Boom; Finite Element Analysis; Structural Optimization; Buckling; Lightweight Design

I. Introduction

Telescopic handlers (telehandlers, rough-terrain variable-reach trucks) combine the load-handling role of a forklift with a crane-like boom. The boom is a nest of thin-walled rectangular tubes. A hydraulic cylinder, often with chains or ropes, extends the inner sections out of the outer section. Each section rides on the one outside it through sliding pads, so loads pass between sections through small contact areas (Fig. 1).

The inner boom is critical for three reasons:

- It is the most highly stressed part at full extension, because the bending moment from the load grows with reach.
- It must resist local buckling of its thin walls, global buckling, torsion from side loads, and local contact stress at the pads.
- Its mass adds to the overturning moment and to the dead weight that the carrier must support.

Optimization therefore means reducing mass or raising stiffness while still meeting strength and stability limits. Analytical methods give early sizing, but they struggle with contact zones, plate buckling and irregular loading. FEA handles these effects, and when it is linked to an optimizer it can search many designs automatically.

Literature on telescopic-boom optimization is spread across machine types, and the telehandler inner boom is rarely the stated subject. This review therefore gathers the transferable practice: how booms are loaded and modelled, which analyses are run, and how optimization is formulated and solved. It makes three contributions: (i) it consolidates load cases, modelling choices and optimization formulations reported for telescopic booms and comparable thin-walled box structures; (ii) it compares the reported outcomes and shows, in Table 2, where the studies differ in assumptions; and (iii) it identifies research gaps specific to the telehandler inner boom and proposes a workflow for closing them.

Section 2 describes how we found and selected sources. Section 3 describes boom structure, loading and the stability context. Section 4 reviews FEA practice, Section 5 reviews optimization approaches, and Section 6 summarises the studies. Sections 7 and 8 discuss gaps and propose a workflow, and Section 9 concludes.

II. Review Scope and Method

This is a narrative review, not a systematic one. Sources were located in October 2026 by web searches of publisher and repository platforms (Atlantis Press, MDPI, the Egyptian Knowledge Bank, SAE Mobilus, SPIE Digital Library, the IASJ repository, and university repositories), using combinations of the terms *telehandler*, *telescopic boom*, *forklift boom*, *finite element*, *lightweight*, *optimization*, *buckling*, *fatigue*, *sliding pad* and *digital twin*. A source was kept if it met all of these conditions:

- It deals with FEA, optimization or structural assessment of a telescopic boom or a comparable thin-walled boom structure, or with the stability context of telehandlers.
- It was published between 2003 and 2026.
- Enough was accessible to state its method and outcome, from the full text or the abstract and bibliographic record. Three paywalled sources [1–3] were assessed from the abstract only.

Manufacturer brochures, patents and non-public presentations were excluded. Two telehandler-specific structural items are grey literature: a technical case study [4] and a master's dissertation [5]. They are included because they are the only telehandler-specific structural analyses found, and they are flagged in Table 1. Studies of telescopic booms on other machines are included because the inner-boom physics are the same: a box beam with sliding contacts under bending and torsion.

The review has limits. No Scopus or Web of Science query was run, only English-language sources were screened, and journals not indexed by open platforms may be under-

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represented. The number of sources is modest, so the findings should be read as a representative map of current practice, not an exhaustive census.

III. Structure, Loading and Stability Context of the Telehandler Inner Boom

A. Geometry and load path

A typical boom has two to four nested sections. The load acts at the fork carriage or boom head. The weight of the boom and attachment, the lifting-cylinder reaction and the support at the rear pivot pin all act together. Between sections, load is transferred through upper and lower sliding pads that define the overlap zone. In a recent analysis of a forklift-truck telescopic boom 9.89 m long when fully extended, a 1000 kg rated load at the head produced the highest stress at the hydraulic-cylinder support and the largest deflection at the tip [6], which matches the pattern reported for a 35-tonne crane boom [7].

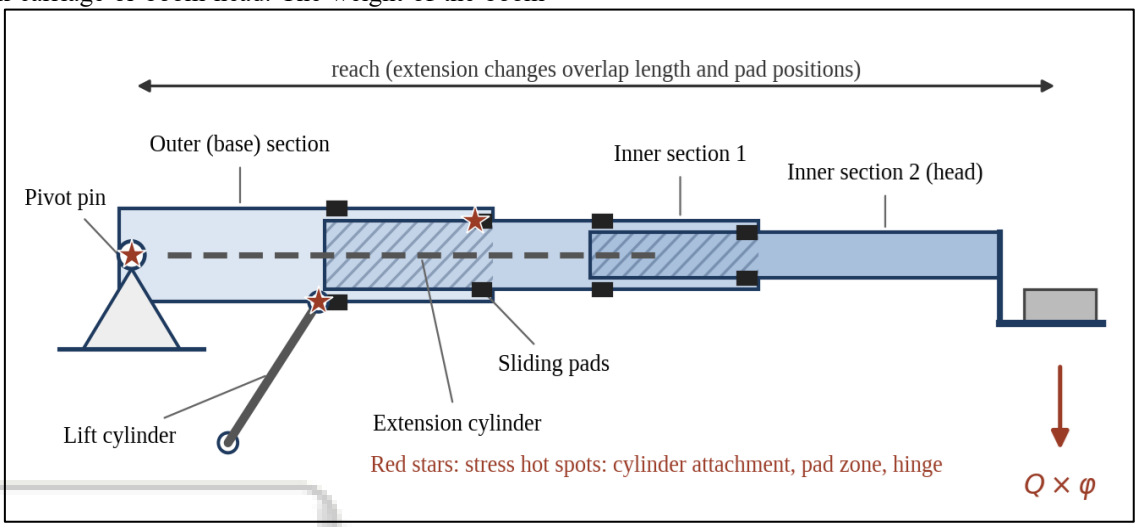


Fig. 1: Schematic of a telehandler boom: nested sections, sliding pads in the overlap zones, extension and lift cylinders, and the rated load Q multiplied by a dynamic factor φ. Red stars mark the usual stress hot spots (cylinder attachment, pad zone, hinge).

B. Typical load cases

Studies commonly examine the following cases:

- Maximum moment case: full extension, rated load, boom at a specified angle. In [6] the horizontal (0°) posture proved the most hazardous of those compared.
- Dynamic factor case: rated load multiplied by a factor for shock. The telehandler analysis in [4] used a dynamic "bounce" factor of 1.5, while the forklift boom in [6] used a lifting dynamic coefficient of 1.15.
- Side or lateral load case: producing torsion and bending about the weak axis.
- Retracted or intermediate positions: where overlap length and pad positions change the stress pattern.

The gap between 1.15 and 1.5 is large. A design optimized to one factor may fail or be over-built under the other, so the choice of factor, and the standard it comes from, must be stated in any inner-boom study.

C. Failure Modes

The failure modes most often reported are yielding at stress concentrations, local plate buckling of the compression wall, overall column-type instability, and excessive deflection or vibration at the boom head. Stress concentrations arise near hydraulic-cylinder attachments, pad contact zones and hinge points [2, 5, 7]. A dedicated study of the contact zone between inner and outer truck-crane boom segments analysed the local stress increase there and verified it by finite elements and by test [3]. Resonance is a further concern: in [6] the second natural frequency of the extended boom, about 5.9 Hz, coincided with the peak of the harmonic response at the boom tip.

D. Stability Context and Standards

Structural strength is one of two limits on a telehandler's load chart; the other is tipping stability. Stability of rough-terrain variable-reach trucks is verified by tilt-table testing under ISO 22915-14 [8], and CEN/TS 1459-8:2018 and EN 15000:2008 are among the other machine-safety standards listed for these machines [9]. Puras et al. built a bond-graph dynamic model of a telehandler to estimate stability limits and predict dynamic behaviour, and they validated it experimentally [9]. Their model treats the machine, not the boom, in detail, but it shows how the load moment grows and the rear axle unloads as the boom extends. Inner-boom mass therefore affects the structural margin and the stability margin together, which is why mass is the usual objective in boom optimization.

IV. FEA Methods Applied to Telescopic Booms

A. Model idealisation

Two modelling choices dominate. Plate or shell elements are efficient for thin-walled box sections, and solid elements are better for contact and pin regions. The telehandler analysis in [4] modelled the boom sections with plate elements, idealised the hydraulic cylinders and axles as beams, and represented the boom pin and its contact with the truck using gap and rigid-link elements. The forklift-boom study in [6] instead used solid elements (SOLID186) and confirmed mesh independence by comparing the peak equivalent stress across

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several mesh sizes. A telescopic-crane model in [10] combined shell elements (SHELL281) for the plate boom with solid elements for the sliding blocks, which keeps the pads as separate bodies at modest cost.

B. Treatment of sliding-pad contact

Treatment of the pads varies widely. At one end, explicit gap or contact elements capture where load is actually carried [4, 11]. Pavlovic et al. combined a theoretical treatment of the sliding pads with ANSYS buckling analysis [11], and Savković et al. derived and tested analytical expressions for the stress in the contact zone between boom segments [3]. At the other end, pads are tied rigidly to the boom walls. Sun et al. did this on the grounds that their analysis represents a single posture with no relative motion between sections [6]. A rigid tie is convenient for optimization but it distributes pad load over the whole contact patch and can hide local stress and edge effects. A static analysis of a truck-crane boom with a single cylinder and pinning system, optimized in MATLAB, used its results to locate stress concentrations and to strengthen the slide pads [2], which shows that pad regions are design drivers in their own right.

C. Static strength and stiffness

Static analysis is the baseline. Work on a 35-tonne Tadano telescopic crane boom compared an ANSYS model with manual beam-column calculations and strength-of-materials formulas. It found the highest stress where the hydraulic rod meets the first tube and the largest deflection at the head sheave, and it reported that the ANSYS results were somewhat less accurate than the manual calculations [7]. That result is a reminder that idealisation and meshing choices, not the solver, set the accuracy. A master's dissertation on telehandler components used HyperMesh and Nastran-format models to run five load cases on the chassis, found failure at the hinge points, and redesigned the chassis to meet stress limits [5].

D. Buckling and stability

Because the walls are thin, linear buckling analysis is a standard check. Pavlovic, Fragassa and Minak found critical loads and mode shapes above the most extreme working conditions, which confirmed stability [11]. Buckling can also be handled inside an optimization loop: Liu and Qu computed the maximum carrying capacity by buckling analysis and then used it as the applied load in the static analysis for thin-walled structure optimization [12].

E. Modal and vibration analysis

Telescopic structures with long overhang are sensitive to vibration. Yucel and Arpaci analysed telescopic platforms using solid modelling and FEA, checked the results with experimental modal analysis, and then optimized the structure to meet vibration standards [13]. Sun et al. extracted the first six constrained modes of the extended forklift boom, used harmonic response analysis to find which mode governed tip motion, and then made the second natural frequency an optimization objective [6]. Modal and flexural analysis of the optimized model was also used to suggest further improvements to a 300-tonne truck-crane boom [1].

F. Fatigue

Fatigue is analysed far less often than static strength, yet booms see repeated cycles and welded joints. Brahami, Hamri and Sfarni analysed a whole crane boom, identified two critical welded zones, and assessed them in sub-models using three structural-stress approaches with S–N curves, finding that Dong's method gave the most conservative results [14]. Cao et al. used ANSYS nCode DesignLife on a telescopic crane with a defined work cycle. The weakest point was the connection between the side ribs and the first section, with a life of 4.078×10^5 stress cycles, which they converted to about 25 years at 16,000 cycles per year [10]. Luo and Bu included fatigue in a Workbench assessment of a rescue-machine arm [15]. In none of these is fatigue life fed back as an optimization constraint.

G. Validation

Some studies validate their FE models against tests. Yucel and Arpaci used experimental modal analysis [13], Beňo, Kozak and Konjatić compare calculated stresses and deformations with measurements to judge how effective optimization has been [16], Savković et al. verified their contact-zone stress model by test on a truck crane [3], and Wang and Xing report that static experiments verified their truck-crane boom model before it was optimized [1]. Cao et al. instrumented eight points on a telescopic crane with triaxial strain gauges. The largest difference between the FE model and the measurement was 6.18 MPa, and the largest difference between their surrogate model and the measurement was 8.98 MPa, under the heaviest test load [10]. Such figures give a useful benchmark for what accuracy to expect from a shell-element boom model.

V. Optimization Approaches in the Literature

A. Cross-section shape and size optimization

Beňo, Kozak and Konjatić argue that telescopic jib arms are enclosed thin-walled bars loaded by combined bending and torsion, and that FEM in ANSYS can be used to design an optimum cross-section shape. They also note that no generally valid, scientifically proven design principles exist for such jibs [16]. This bears directly on the inner boom, whose cross-section is the main design variable. Wen optimized the cross-section size of the first telescopic boom of a 25-tonne truck crane with MATLAB's optimization toolbox and reported a lower total weight [2].

B. Parametric (APDL) optimization with evolutionary algorithms

Jia and Wan built a parametric ANSYS (APDL) model of a QTZ5010 tower-crane boom. Cross-sectional dimensions were the design variables, and stress and strain under extreme conditions were the constraints. Fuzzy and genetic algorithms searched for minimum mass, which cut boom mass by 309 kg, about 11.6%, while keeping strength within the design specification [17]. The machine differs, but the formulation, with sections as variables, mass as the objective and stress and displacement as constraints, carries over to a telehandler boom.

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C. Plate-thickness optimization with automated FE loops

Liu and Qu minimised the mass of a thin-walled structure by changing plate thickness. They wrote software that reads ABAQUS output, takes the maximum stress in each plate as a critical stress, computes a new thickness and writes a new input file. Six optimization iterations gave a minimum-mass structure with high carrying capacity [12]. The approach suits boom walls, where top, bottom and side plates can have different thicknesses.

D. Surrogate-assisted and multi-objective optimization

Direct coupling of an FE solver to an optimizer is costly when many variables are involved, so recent work replaces the solver with surrogate models. Panganiban, Ahn and Chung split the design of a nine-segment aerial-lift boom into two phases. In the first, a metamodel built from space-filling sampling and Kriging gave optimal section heights, overlaps and wall thicknesses. In the second, topology optimization located reinforcement in the supporting frame and plate thickness was then optimized [18]. Wang and Xing used the response-surface module of ANSYS Workbench to lighten a 300-tonne truck-crane boom under three dangerous working conditions while meeting strength and stiffness [1].

Two recent studies address forklift-truck telescopic booms directly. Fang et al. proposed a multi-level scheme: topology optimization of the boom cross-section, reconstruction of a parametric model from the result, and then mass minimization under stress and deformation constraints using optimal space-filling sampling, a Kriging surrogate and a multi-objective genetic algorithm. The optimized boom was 8.44% lighter and had higher torsional stiffness [19]. Sun et al. used nine cross-section parameters of a 9.89 m forklift boom, screened them with Spearman correlation, and built a hybrid surrogate in which response surfaces predicted mass and deformation and a radial-basis-function network predicted stress and natural frequency. Their NSGA-II search produced 147 Pareto solutions, and a game-theoretic combined weighting with TOPSIS selected the compromise design [6].

E. Reliability-based optimization

Deterministic optima tend to sit on constraint boundaries, so small variations in plate thickness or section size can break a constraint. Sun et al. tested this with Monte Carlo simulation (10,000 samples, coefficient of variation 0.02 on each variable). The deterministic optimum reached a sigma level of only 1.508 for maximum deformation, below the 3σ target. Re-optimizing with 3σ constraints gave a final design with 12.5% less mass and a 9.4% higher second natural frequency, with all responses at the 3σ level [6]. For an inner boom, where wall thickness tolerances and weld distortion are real, this shows that a deterministic mass figure is not a safe design figure on its own.

F. Workbench parameter optimization

Luo and Bu analysed the telescopic arm of a rescue machine in ANSYS Workbench 15.0 with static and fatigue assessments, compared designs by material and cross-section, and then ran parameter optimization [15]. It is a typical Workbench workflow that could be applied to a telehandler inner boom.

G. Data-driven surrogates and digital twins

Surrogate models are now also used to monitor booms in service. Cao et al. built a gradient-boosting surrogate (BO-LightGBM) trained on 100 FE load cases that predicts nodal stress in about 0.55 s, against 2580 s for one FE run, and linked it to a digital twin that scores load, strength, stiffness and fatigue life during operation [10]. The aim there is assessment rather than design, but the same trained surrogate could serve an optimization loop, and in-service stress data could update the load spectrum used for inner-boom fatigue design.

H. Redesign by analysis feedback

Not all optimization is algorithmic. The telehandler dissertation [5] and the case study [4] use FEA results to find weak points and redesign. In [4], analysis of a market-leading lift truck revealed design problems, and the new design was then analysed under the full design load including the dynamic factor.

VI. Summary of Reviewed Studies

Table 1 lists the reviewed studies in order of publication. Table 2 compares their modelling and optimization characteristics. Mass reductions are not directly comparable between studies, because the baselines, load cases, constraints and boom types differ.

Ref.	Structure	Analysis type	Optimization / outcome
[4] (2003)	Telehandler boom, cylinders, truck frame	Static FEA; plate and beam elements; gap and rigid links	Design strengthened under 1.5 dynamic factor
[5] (n.d.)	Telehandler chassis and dipper arm	Static, five load cases; HyperMesh / Nastran	Chassis redesigned at failing hinge points
[13] (2012)	Telescopic platform	Modal FEA with experimental modal analysis	Optimized to meet vibration standards
[18] (2012)	Nine-segment aerial-lift boom	Metamodel (Kriging); topology; thickness	Two-phase lightweight design
[7] (2013)	Telescopic crane boom (35 t)	ANSYS vs manual and strength-of-materials	Locates peak stress and deflection
[16] (2014)	Telescopic jib arm	FEM in ANSYS; comparison with experiment	Optimum cross-section shape
[3] (2014)	Truck-crane telescopic boom, contact zone	Analytical model; FEM; test on a truck crane	Guidelines for optimum box-segment design

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[12] (2015)	Thin-walled structure	ABAQUS static plus buckling	Plate-thickness optimization in six iterations
[17] (2016)	Tower-crane boom	APDL static FEA	Fuzzy + genetic algorithm; −309 kg (≈11.6%)
[15] (2016)	Telescopic rescue arm	ANSYS Workbench static, fatigue	Parameter optimization
[11] (2017)	Telescopic boom with sliding pads	ANSYS static and linear buckling	Verified resistance and stability
[14] (2019)	Crane boom, welded joints	FEA with sub-models; structural-stress fatigue	Fatigue assessment (no optimization)
[1] (2023)	300 t truck-crane telescopic boom	ANSYS; static experiments; modal, flexural	Response-surface lightweight design
[2] (2023)‡	Truck-crane boom, single cylinder and pinning	Static FEA; slide-pad stress	Cross-section size optimization (MATLAB)
[9] (2024)	Compact telehandler (machine level)	Bond-graph multibody dynamics; tests	Stability limits and dynamic behaviour
[10] (2025)	Telescopic crane boom and turntable	ANSYS static, fatigue; strain-gauge tests	BO-LightGBM surrogate; digital-twin assessment
[6] (2025)	Forklift telescopic boom, 9.89 m	ANSYS static, modal, harmonic	RSM-RBFNN + NSGA-II + 3σ; −12.5% mass, +9.4% f_2
[19] (2026)	Forklift telescopic boom cross-section	Topology and parametric FEA	Kriging + MOGA; −8.44% mass, higher torsional stiffness

Table 1. Summary of reviewed studies, in order of publication.
† Grey literature (technical case study or dissertation). ‡ Paywalled; abstract-level information only.

Ref.	Elements	Pad contact	Dyn. factor	Buckling	Fatigue	Test validation	Optimization method
[4]	Plate, beam	Explicit (gap, rigid link)	1.5	n.r.	n.r.	n.r.	Redesign by analysis feedback
[13]	Solid	n.r.	n.r.	n.r.	n.r.	Modal test	Optimization to vibration limits
[16]	n.r.	n.r.	n.r.	n.r.	n.r.	Yes	ANSYS shape optimization
[3]	Analytical + FE	Contact zone analysed	n.r.	n.r.	n.r.	Yes	Analytical design guidelines
[12]	n.r.	n.r.	n.r.	In loop	n.r.	n.r.	Plate-thickness loop (ABAQUS)
[17]	Beam	n.r.	n.r.	n.r.	n.r.	n.r.	Fuzzy + genetic algorithm
[15]	n.r.	n.r.	n.r.	n.r.	Assessed	n.r.	Workbench parameter optimization
[11]	n.r.	Theory + FE	n.r.	Linear	n.r.	n.r.	None (verification)
[14]	FE + sub-models	n.r.	n.r.	n.r.	Assessed	n.r.	None (fatigue assessment)
[1]	n.r.	n.r.	n.r.	n.r.	n.r.	Static test	Response surface (Workbench)
[10]	Shell + solid	Solid sliders	Per GB/T 3811	n.r.	Assessed	Strain gauges	None (BO-LightGBM surrogate for monitoring)
[6]	Solid	Rigid tie	1.15	n.r.	n.r.	n.r.	RSM-RBFNN; NSGA-II; TOPSIS; 3σ reliability

Table 2. Modelling and optimization characteristics of selected studies.
*n.r. = not reported in the material consulted for this review. "Assessed" means included in the analysis but not used as an optimization constraint. Studies are listed only where at least one characteristic could be established.

VII. Discussion and Research Gaps

A. What the Studies Agree On

Across machine types the formulation is consistent: cross-section dimensions and plate thicknesses are the design variables, mass is the main objective, and stress and deflection are the main constraints [6, 12, 17, 19]. The solution strategy has moved from direct APDL or ABAQUS loops [12, 17] towards surrogate-assisted multi-objective search [6, 18, 19], and one study has added reliability constraints [6]. Reported mass reductions fall in a narrow band of about 8% to 12.5% [6, 17, 19], which suggests, with caution because the baselines differ, that conventionally designed booms carry a margin of that order.

B. Materials, Welds and Manufacturing

The reviewed forklift and crane booms use structural steels in the Q235 to Q460 range, with the higher grade used for the heavily loaded parts [6, 10]. Higher-strength steel allows

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thinner walls, but thinner walls increase buckling and stiffness risk, and welded joints set fatigue life [14]. The optimization formulations described in the sources consulted do not include plate-thickness steps, weld length or forming limits as constraints, so a mathematically optimal boom may not be the cheapest to build.

C. Gaps

- G1. Little work isolates the telehandler inner boom. Only [4] and [5] concern telehandler structures, and neither is a peer-reviewed optimization of the inner boom. The two recent forklift-boom studies [6, 19] treat the boom assembly or its cross-section rather than the inner section as a separate part with its own overlap and pad loads. Telehandler booms are smaller than crane booms and have a shorter overlap, they work with forks and varied attachments, and they take frequent dynamic loads from travelling over rough ground.
- G2. Contact and pad modelling. Boom sections interact through sliding pads, and models that merge sections or tie the pads rigidly can miss local stress at the overlap [3, 6]. Studies that treat contact and pads explicitly [4, 10, 11] suggest that this should be standard in inner-boom work.
- G3. Buckling inside the loop. Strength and deflection are often the only constraints in the optimization loop [6, 17, 19]. Since thin walls often fail by buckling, buckling load factors should be constraints too, as in [12]. Reducing plate thickness to save mass without a buckling constraint can move the failure mode from yielding to instability.
- G4. Fatigue is assessed but not optimized. Fatigue appears in [10, 14, 15], but in none of the optimization studies consulted is fatigue life a constraint or objective. Telehandler booms see repeated load cycles from rough-ground travel, so this gap matters more for them than for many cranes.
- G5. Experimental validation. Few optimization papers validate with strain gauges on a real boom. Validation is shown in [13], [16], [1], [3] and [10], but it is not routine, and none of the validated studies concerns a telehandler inner boom.
- G6. Inconsistent load factors. The dynamic factor ranges from 1.15 [6] to 1.5 [4] across the studies that state one. Without a standard-referenced factor, results cannot be compared between studies or between machines.
- G7. Joint treatment of mass, stiffness, dynamics, reliability and manufacturability. Mass is the dominant objective. Head deflection, natural frequency and reliability have been combined for a boom in [6], but manufacturability is rarely handled together with them.
- G8. Structure and stability are treated separately. Telehandler stability models [9] and boom structural models are developed independently. A boom-mass change alters both, yet no reviewed study links the two.

VIII. Proposed FEA-Based Optimization Workflow for the Inner Boom

Combining the practices in the reviewed work, a suitable workflow is shown in Fig. 2 and described below.

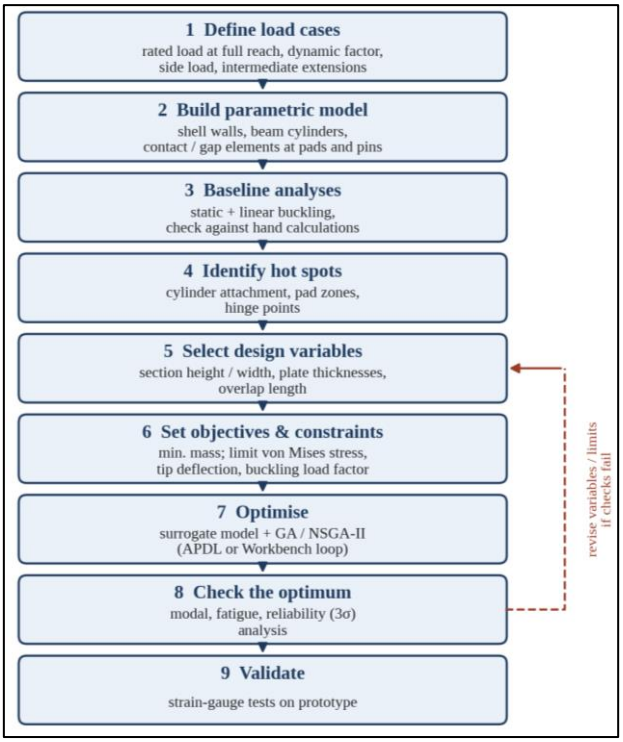


Fig. 2: Proposed FEA-based optimization workflow for the telehandler inner boom.

- Define load cases: rated load at maximum reach, a dynamic factor taken from the applicable standard, side load and intermediate extensions.
- Build a parametric model: shell elements for the walls, beam elements for cylinders, and contact or gap elements for the pads and pin [4, 10].
- Run baseline static and linear buckling analyses, and verify against hand calculations [7].
- Identify stress hot spots near the cylinder attachment, pad zones and hinge points [3, 5, 7].
- Select design variables: cross-section height, width, plate thicknesses and overlap length [12, 16], screened by sensitivity analysis [6].
- Set objectives and constraints: minimise mass; limit von Mises stress, tip deflection and buckling load factor, and treat the natural frequency as an objective or constraint where resonance is a risk [6].
- Optimize with a surrogate model and a genetic or NSGA-II algorithm in an APDL or Workbench loop [6, 12, 15, 17, 19].
- Check the optimum with modal analysis, fatigue [10, 14, 15], and a reliability (3σ) analysis of the deterministic result [6].
- Validate with strain-gauge tests on a prototype [10, 16].

If the checks in step 8 fail, the loop returns to step 5 with revised variables or limits.

IX. Conclusion

Literature from 2003 to 2026 shows that FEA, combined with parametric modelling and optimization algorithms, can reduce mass in telescopic booms by roughly 8% to 12.5% while keeping strength and stability [6, 11, 17, 19]. Recent forklift-boom studies add surrogate models, multi-objective search and reliability analysis [6, 19], and crane studies add

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fatigue assessment and strain-gauge validation [10, 14]. Telehandler-specific structural studies remain few [4, 5], so much of what is known about inner-boom optimization comes from cranes and aerial-lift booms. Future work should concentrate on contact modelling at the pads, buckling and fatigue constraints in the optimization loop, standard-referenced load factors, the link between boom mass and machine stability, and experimental validation on real telehandler booms.

ABBREVIATIONS

APDL, ANSYS Parametric Design Language; FEA / FEM, finite element analysis / method; GA / MOGA, genetic algorithm / multi-objective genetic algorithm; NSGA-II, non-dominated sorting genetic algorithm II; RBFNN, radial-basis-function neural network; RSM, response surface method; TOPSIS, technique for order preference by similarity to ideal solution; 3σ, three-sigma reliability level.

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HIGHLIGHTS

[1] Reviews FEA-based analysis and optimization of telescopic booms, 2003–2026.

[2] Telehandler inner-boom studies are scarce; crane and forklift work transfers.

[3] Rigid pad ties, missing buckling and fatigue constraints are the main gaps.

[4] A nine-step workflow for inner-boom optimization is proposed.

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