RESEARCH REPORT

SIMPRO - Task-2-1
Guiderail Optimization by HEEDS

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Confidentiality: Public

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Summary

SIMPRO Task 2.1 Parameter optimization included a case application. This case was provided by KONE and was related to an elevator guiderail mass optimization. Elevator guiderail FEM model was optimized using Abaqus FEM software together with HEEDS optimization software.
Results of the work helped in understanding the structural optimization work and learning the HEEDS software usage.

Confidentiality	Public

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1. Introduction

This report describes the work done for SIMPRO project, Task 2.1.2 parameter optimization with FEM. The work is related to the case KONE guiderail mass optimization using Abaqus and HEEDS. Work was done in February-May, 2013, at VTT Espoo.

Other work within the SIMPRO project and case KONE guiderail optimization is done and reported separately. One report is written on the guiderail optimization with Abaqus and Isight at VTT Espoo. Another separate guiderail optimization subtask is done and reported for multibody simulation work with ADAMS and HEEDS at VTT Espoo. Also a thesis work is ongoing at Tampere University with guiderail optimization using ANSYS and Tampere University developed optimization code called "Simpro Optimization Tool".

In this work a simple multi-span T-profile beam is modelled in Abaqus and connected to HEEDS optimization software for finding acceptable minimum weight solutions for the guide rail geometry. The model is limited to a part of a single guiderail installation with a fixed amount of support brackets and a fixed loading. The loading represents the elevator guide forces from the roller to the guide rail. Guide forces are bending the guiderail beam due to unbalanced elevator loading and elevator dynamics.

A simplified analytical Excel model of the guiderail deflection and buckling calculation was also built. This model is fast, robust and was used with HEEDS for investigating the optimization part of the process. The model describes a single span beam with rotational springs as end supports and a point load at middle of the span. Variables are the beam length, support spring stiffness and the profile section geometry. Results are the deflection, ride comfort parameter and the buckling load.

The optimization objectives were defined as minimum guiderail line-mass and maximum total span with a fixed amount of support brackets. Also, a bending angle was defined as an angle α which is based on the maximum beam deflection and beam length. Other constraints were the absolute maximum beam deflection and rotation together with minimum beam axial buckling load value. Variables were the beam length, beam cross section as a discrete set of existing T-profile geometries and the support bracket stiffness values. Load locations were varied but load values were constant.

In this work, focus was on the optimization process setup with structural models and the usage of the modelling and optimization tools. A lot of useful experience was gained from the work.

2. Investigated Methods

The computational tools used in this work were Dassault Systems Abaqus FEM software and Red Cedar Technologies HEEDS optimization software.

The Abaqus CAE environment with a variation of models and solvers were used for modelling and analysis. The Abaqus analysis model is connected to HEEDS optimization software by tagging selected geometry and modelling variables as parameters.

The optimization case is defined within HEEDS and runs as a batch of some 200 to 2000 Abaqus (or Excel, or almost any software model) analysis solutions with different combinations of the selected variable parameters.

Methods for Abaqus models, Excel model and HEEDS optimization are described here.

2.1 Abaqus Modelling Mehods

A variety of Abaqus models with different solvers and elements were tested. It was difficult to arrive at a satisfying setup that met all demands for the case KONE definition. Both static and dynamic solutions were tested. Dynamic solver models approached multibody simulations in complexity. However, when implemented with FE analysis tools, they were ridden with modelling difficulties and solution stability problems. The computational effort is also an issue with large, detailed and long dynamic simulations. On the other hand, small size static solutions with beam element approximations may include too many simplifications and therefore be useless for practical results in product development use. However, it must be said that simple models often give valuable qualitative insight to the analysis events and basic model behavior.

Graphical views of the model geometry can be seen in the Figures below. Some tested models differ from these in terms of modelling approach and/or geometry of the model. They all still represent the guiderail and the bracket supports loaded by bending forces from the elevator guide rollers.

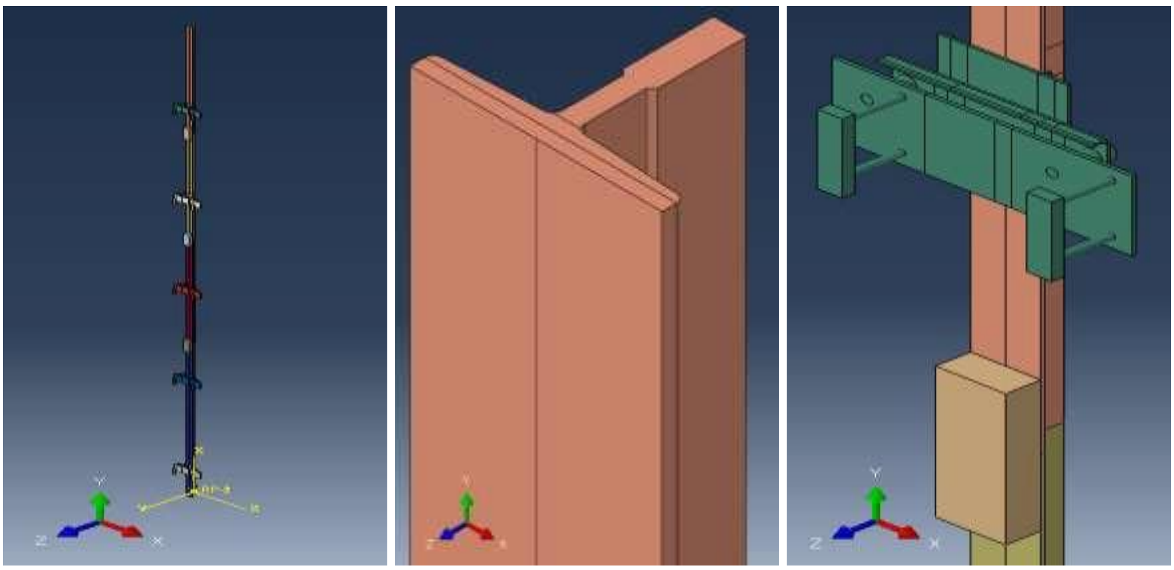


Figure 1. Solid model showing the single guiderail T-profile with four beam spans, five bracket supports and three fishplate connections.

2.1.1 Dynamic Implicit with Solid Elements

- Detailed but heavy
- Possibility of sliding surface to surface contacts
- Travelling load is still difficult to set up properly
- Brackets and fishplates included
- Dynamic solution is irrelevant without a larger model scope including the elevator car and the other guiderail

2.1.2 Dynamic Implicit with Beam Elements

- Less details and DOFs, quicker to solve
- Loss of T-profile geometry details
- Support and load location is not accurate
- Travelling loads with Fortran subroutine DLOAD
- Brackets included as springs
- Fishplates omitted as springs caused problems in beam connections
- Dynamic solution is irrelevant without a larger model scope including the elevator car and the other guiderail

2.1.3 Linear Static with Solid Elements

- No dynamics
- Simple to manage
- Computationally robust
- No travelling loads
- Fixed load positions
- Brackets and fishplates included

2.1.4 Linear Static with Beam Elements

- No dynamics
- Simple to manage
- Fast to compute
- Computationally robust
- No travelling loads
- Fixed load positions
- Loss of T-profile geometry details
- Support and load location is not accurate
- Brackets and fishplates included as springs

2.1.5 Dynamic Explicit with Solid Elements

- Too heavy to compute
- Contact features can be defined
- Contacts are just a work around for applying the travelling loads. Does not justify the use of the explicit solver.
- Brackets and fishplates included
- Dynamic solution is irrelevant without a larger model scope including the elevator car and the other guiderail

2.1.6 Dynamic Explicit with Beam Elements

- Too heavy to compute
- Contact features can be defined even between beam elements
- Beam element contacts did not work accurately enough to apply loads correctly
- Even with beam elements, contacts are just a work around for applying the travelling loads. Does not justify the use of the explicit solver.
- Dynamic solution is irrelevant without a larger model scope including the elevator car and the other guiderail

2.1.7 Dynamic Analysis with Modal Modelling Approach

The modal modelling approach was not tested, but has its advantages in dynamic models. Especially when the lowest bending modes are of interest and the high frequency domain can be omitted. Modal representation with the few lowest modes can render the dynamic model very fast and even robust. On the other hand, the method will probably introduce a set of limitations to the already troublesome travelling load application on the detailed location of the beam profile. Also, it may not be the best option to a optimization case where the geometry is changed at every calculation and the modes need to be computed separately for every design.

2.1.8 Linear Buckling Analysis

- Analysis is a simple linear axial buckling eigensolution with first mode read as lowest buckling load result
- Included in the same Abaqus analysis job, as a preliminary step
- Was found to interfere with the following dynamic solution and causing problems by reducing the computational stability
- Separate analysis could be built to the optimization job

2.1.9 Modelling Issues to Consider

- Model size and DOFs
- Time step size
- Simulation length
- Loading, contacts with solids, beams, forces
- Moving loads, Fortran subroutines
- Time dependent loads
- Moving distributed loads over nodes
- Multiple components
- Detailed load location for variable T-profiles, generation of torsion
- Simplified T-profile geometry
- Bracket representation with springs
- Fishplate representation with springs
- Dynamic model computational robustness and stability

2.2 Excel Model

A simple fast model was needed, when investigating the optimization part of the work with HEEDS. Abaqus dynamic models had difficulties and the ones that were successful were slow to solve. A simple Excel model was built to this need.

The simple Excel sheet calculation is for a single span beam loaded at mid span with point load and supported by coil springs from both ends. Deflection and buckling load is calculated for the T-profile sections and variable spring coefficients. Figures below show the beam model. An overview of the calculation sheet is shown in the appendix.

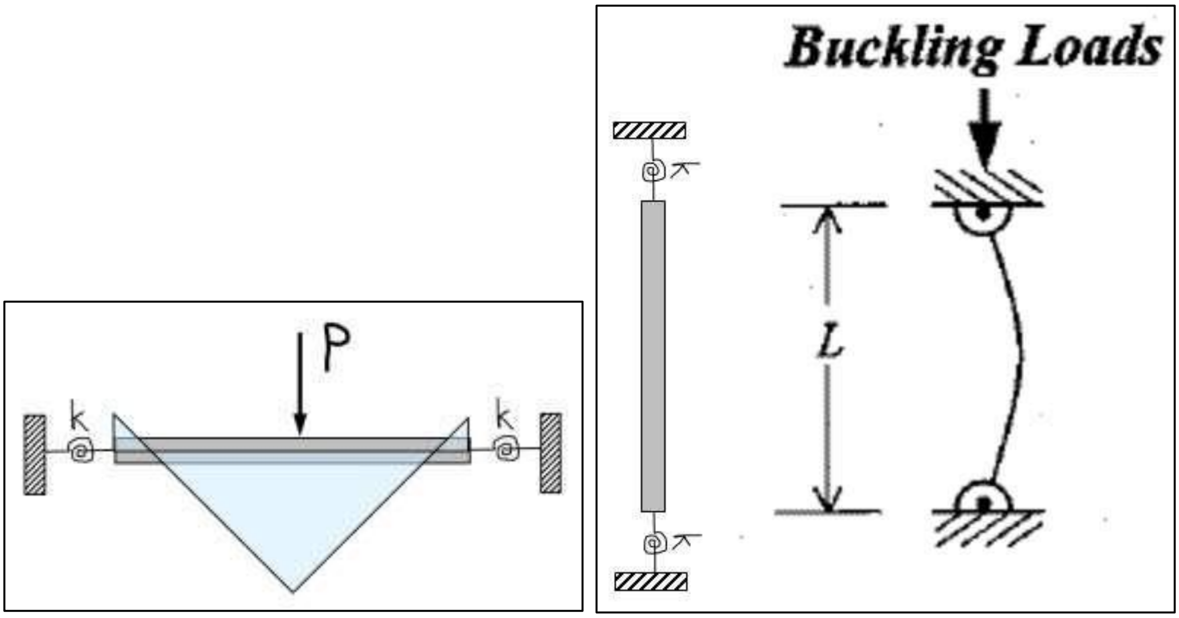


Figure 2.
LEFT: Excel bending beam model showing coil spring supports with stiffness k, central point load P and the resulting bending moment diagram.
RIGHT: Buckling beam with coil spring supports showing axial buckling force direction.

Formulation for maximum deflection of the coil spring supported beam with a point load at mid span is shown below. Maximum deflection δ is found at the midpoint. The maximum deflection is a function of the beam span length *L*, load level *F*, Young modulus *E*, bending inertia *I* and the coil spring stiffness *k* as shown below.

$$\delta = \frac{4FL^3}{192EI} - \frac{3FL^3}{384EI\left(\frac{EI}{kL} + \frac{1}{2}\right)}$$

The Euler buckling load is also a function of the coil spring stiffness k and formulated as.

$$F = \frac{\pi^2 EI}{K^2 L^2}$$

Here the buckling beam effective length factor *K* is defined as shown below with coil spring stiffness parameter *k*. Effective length of K=1 corresponds to the pin-pin supports and the effective length of K=0.5 corresponds to the clamp-clamp supports.

$$K = \sqrt{\frac{2EI + kL}{2EI + 4kL}}$$

2.3 HEEDS Optimization

The HEEDS optimization run will try out different designs by changing the model variable parameters. The parameter values are set by the optimization algorithm and the user defined variable constraints. This means that the candidates for the optimal model configuration are defined by the algorithm logic, but more importantly, influenced by the user defined optimization problem setup. The setup includes algorithm selection, variable selection, variable upper and lower limits selection and variable interconnections definition. Also an objective variable must be set to be minimized or maximized. With a single objective algorithm only one objective variable is defined. In a multi-objective algorithm multiple variables can be minimized and/or maximized in the same problem definition. The model variables need to be constrained appropriately and parameter variables need to be discretized to vary with a defined resolution. HEEDS then runs the full Abaqus analysis calculation for each design.

In a single-objective optimization problem the result is a minimum or a maximum of the objective function. In multi-objective problems, results are always Pareto-fronts and trade off views of two or more objective variables. HEEDS calculates a single performance function to rank the solutions but this function is subject to rather arbitrary weighting factors to be set by the user.

HEEDS uses its default proprietary search methods SHERPA (for single objective optimization) and MOSHERPA (for multi-objective optimization). The SHERPA algorithms combine several different search methods simultaneously, adapting and refining them as the search progresses.

Classical algorithms listed below are available in HEEDS in addition to DOE studies, robustness and reliability studies and evaluation studies.

- SHERPA (**S**ystematic **H**ybrid **E**xploration that is **R**obust, **P**rogressive and **A**daptive)
- MO-SHERPA
- Genetic Algorithm
- Quadratic Programming
- Simulated Annealing
- Response Surface Method
- Multi-Start Local Search
- Particle Swarm Optimization
- Nelder-Mead Simplex

Advantages of SHERPA search method are listed below (according to the HEEDS user manual [1]).

- Finds better solutions the first time, without iterating to identify the best method or the best tuning parameters for your problem.
- Enables non-experts to successfully apply automated optimization the first time.
- Performs direct optimization based on actual model evaluations, rather than using approximate response surface models.
- Identifies better-quality solutions for broad classes of problems, and performs global and local optimization at the same time.
- Uses multiple strategies concurrently to more effectively and efficiently search even complex design spaces.
- Adapts itself to each problem, eliminating the need for user-specified tuning parameters.
- Achieves both global and local search simultaneously.

The work flow for setting up the HEEDS optimization with an Abaqus model is described below.

- Build a beam model in Abaqus
- Edit the desired model variables and record a Abaqus.rpy python file
- Edit the absolute paths to relative paths and save rpy file with a different name
- Create a HEEDS project .hds file and define
    - Optimization processes and methods
    - Analyses, run commands and command line parameters
    - Variables types and variable limits as min, baseline and max values to be used
    - Variables and responses dependencies and formulas
    - References to input and output files for each sub process
        - previous step output is input for the next step in the HEEDS process
        - python file is for editing dimensions
        - cae file is for applying python script and writing the inp file
        - dat file and odb files are for reading the analysis output
    - Tagging of variables in files
    - Method for optimization, algorithm, amount of runs
    - Objectives and constraints for the optimization process
- Run HEEDS with 300 cases
    - a single analysis takes about 5 minutes => 25 hours total
- Monitor and post process results with HEEDS POST
- View best designs and analysis results .odb with Abaqus CAE

2.4 Computation effort

Computation effort is intentionally kept relatively light as the work is done on a single desktop computer. Secondly, the work involves a lot of investigation and testing with the light weight models. Initial plans to expand to models to be solved in high performance computation (HPC) environment were not realized in this work.

In the optimization work, the total computational effort is a product of three issues:

- The FE-model degrees of freedom
  - time to compute one equilibrium time instant
- The duration (in simulation seconds) of a single dynamic implicit simulation and the time step size used
  - initial buckling eigensolution
  - dynamic solution for transient gravity and load onset, 1 s with 5 ms time step
  - dynamic solution for moving the two loads up the rail, 5 s with 5 ms time step
- Amount of optimization cases computed
  - This is related to the amount of objectives and variable parameters included
  - Problem definition and optimization algorithm performance are also key factors
  - 300 evaluations with archive size of 20 (28 cycles)

First issue is reducing the model size to a minimum that is needed to see the desired phenomenon. This is necessary for keeping the total computational effort reasonable in the optimization work.

It could be said that going from dynamic to static solution may take out the second of the three issues.

Maybe the use of high performance computing (HPC) in a distributed multi-processing environment could handle the third issue concerning the amount of optimization cases. However, the HPC environment was not used in this work due to the modelling difficulties. Before adding fire power in computing hardware, much more important is the optimization problem definition, set up and choice of algorithms that can reduce the computational effort related to the third issue. For example, the response surface method might prove useful in this case. The response surface optimization methods are not used or tested in this work.

3. Selected Abaqus Dynamic Implicit Model with Beam Elements

The model and the two solution steps are described in this chapter. Also the optimization process is described here.

3.1 Abaqus Multispan Beam Model

The model is a multi-span beam with five equal length intervals. Support brackets are defined as linear springs. The spring coefficients for the spring supports are defined separately for the translational and rotational degrees of freedom.

The model is subjected to the buckling analysis first, then the two dynamic analysis steps. The first dynamic analysis step is only for setting up the model closer to equilibrium and no results are read from this step. The second dynamic implicit analysis step is where the loads, defined in the FORTRAN user subroutine, start to move along the guiderail with the elevator speed.

Some of the cases do not run for the full 5 s time period due to transients and solution failure resulting from the sudden jump when the loads falls off from the other end of the guide rail. This may happen if the beam is short and the elevator height and speed is high.

The two moving loads are applied with the user defined FORTRAN subroutine. The load is defined for each 0.1 m long beam element as a DLOAD, which has a constant value over the element. The constant value for all the elements in the model is defined as a function of the coordinate y and time t. Parameters are the start height y_{Start}, elevator height L_{Car}, elevator speed u. The constant (over the element) value changes in time periodically with the frequency ω. This frequency corresponds to the guide wheel rolling frequency. Wheel radius is r (about 0.15 m). It is set as $\omega = 2\pi u/(2\pi r) = u/r$. The wheel force phase is set with ϕ_{HI} and ϕ_{LO}. They are both set to zero. Beam element size is set with $L_{Element}$. The MAX function gets values from 0 to unity. It is a triangular shape function defining the load to zero everywhere except where the node y-coordinate is less than an element length away from the defined instant load y location.

$$F_{HI}(y,t) =$$

$$(3000 + 500\sin(\omega t + \phi_{HI})) * MAX\left\{\left(1 - \frac{\left(ABS\left(y - \left(y_{Start} + L_{Car} + ut\right)\right)\right)}{L_{Element}}\right), 0\right\}$$

$$F_{LO}(y,t) =$$

$$(1000 + 500\sin(\omega t + \phi_{LO})) * MAX\left\{\left(1 - \frac{\left(ABS\left(y - \left(y_{Start} + ut\right)\right)\right)}{L_{Element}}\right), 0\right\}$$

The set values of 3000±500 N and 1000±500 N applied to an element length of 0.1 m results in a point load set of F_{HI}=300±50 N and F_{LO}=100±50 N. The lower load value is smaller to demonstrate a resultant moment reaction from the elevator car side. This moment is now constant, though it should supposedly decrease when the elevator car height, L_{Car} is increased for the same elevator car weight.

The forces are applied in the x-direction (perpendicular to T-profile web) only. No force is applied in z-direction (perpendicular to T-profile flange). No separate torsion moment (about beam axis) is applied. The forces act through the T-profile neutral axis with no offset to produce torsion moment.

The buckling load is applied to the negative y direction (beam axis direction) and has a 0.3 m rigid offset in the x-direction (lateral direction, perpendicular to the T-profile web). This is to model the eccentricity of the ideal buckling load and introduce some conservatism in buckling load prediction.

The buckling load analysis is non-conservative and should be considered only as a initial check and starting point for detailed buckling analysis if the buckling load is a critical design driver.

The design value for the buckling load F_k is calculated from the deceleration and the elevator total moving mass in an situation where the elevator emergency brake is applied. Calculation formula shown below is according to the European Standard for elevator safety EN 81-1.

$$F_k = \frac{k_1 g_n (P + Q)}{n}$$

With the given values of P = KT = 4075 kg for elevator moving mass, Q = 2000 kg for rated load, impact factor for “instantaneous safety gear or clamping device” (EN 81-1, table G.2) set as k_1=5, and the number of rails set to n=2, this equation gives a design buckling load of 150 kN. This is the load that the guide rail structure needs to withstand without buckling.

The spatially three dimensional structure is modelled with 1D beam elements with a T-profile. Material is linear elastic steel with E=200 GPa and ρ=7850 kg/m^3.

The T-rails and their properties are described in the following figure.

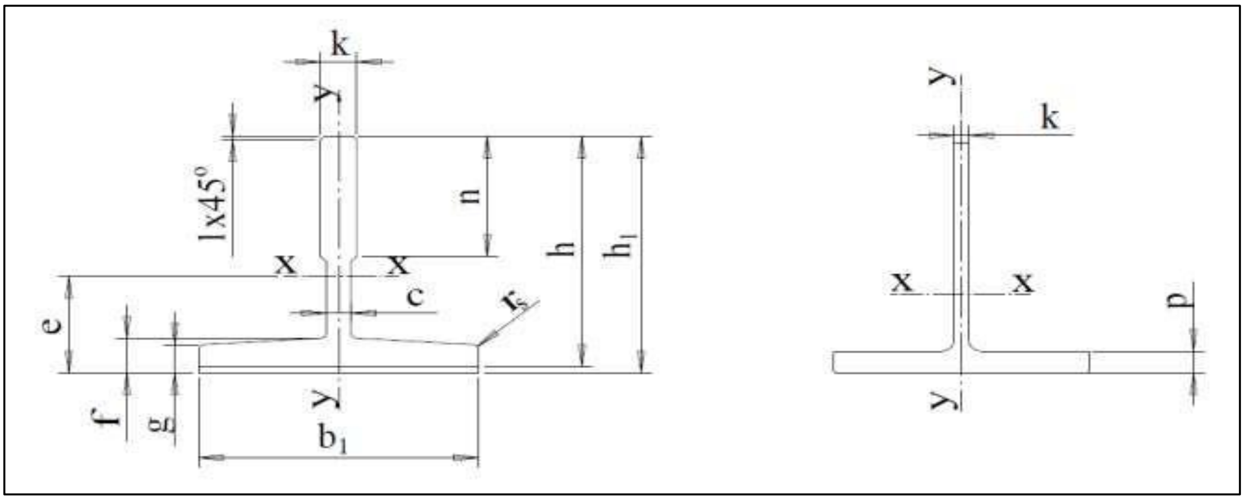


Figure 3. Guiderail profile dimensions. Beam element uses simplified I-beam geometry.

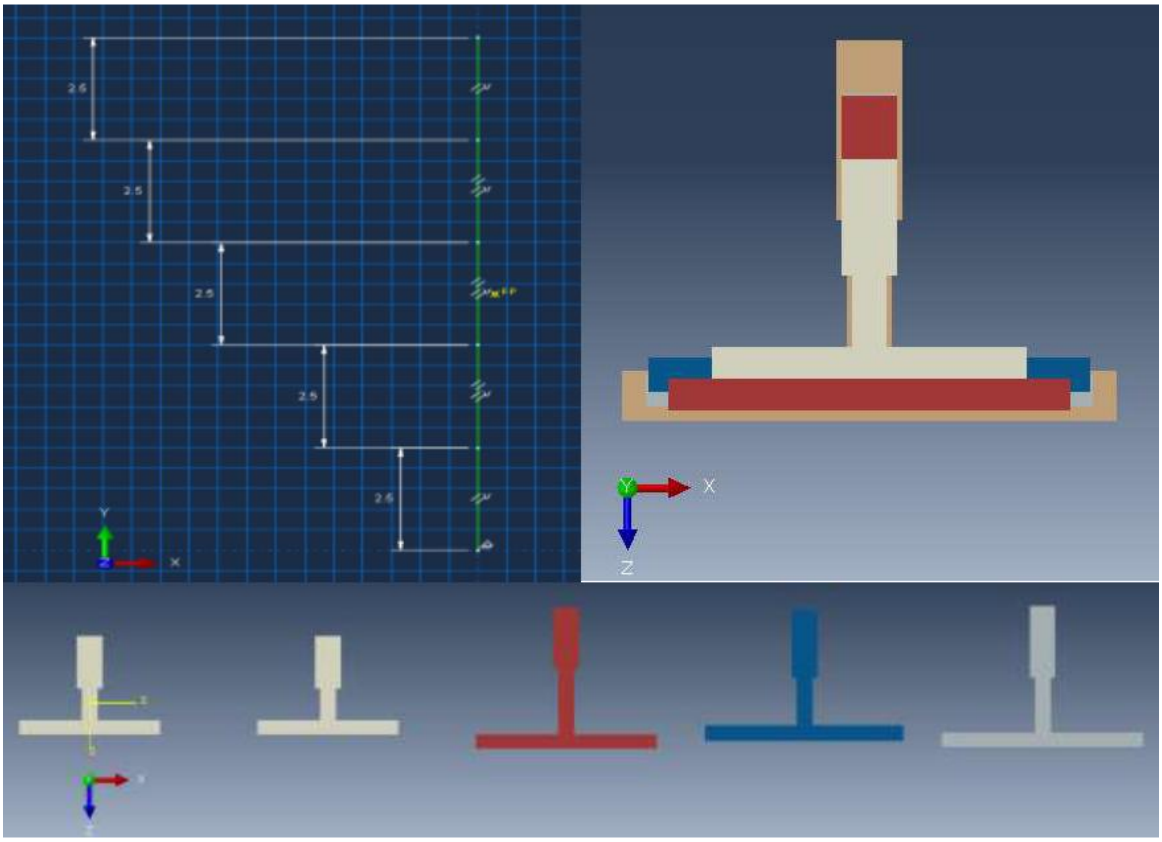


Figure 4. LEFT: Beam model. RIGHT AND BOTTOM: T-profile models.

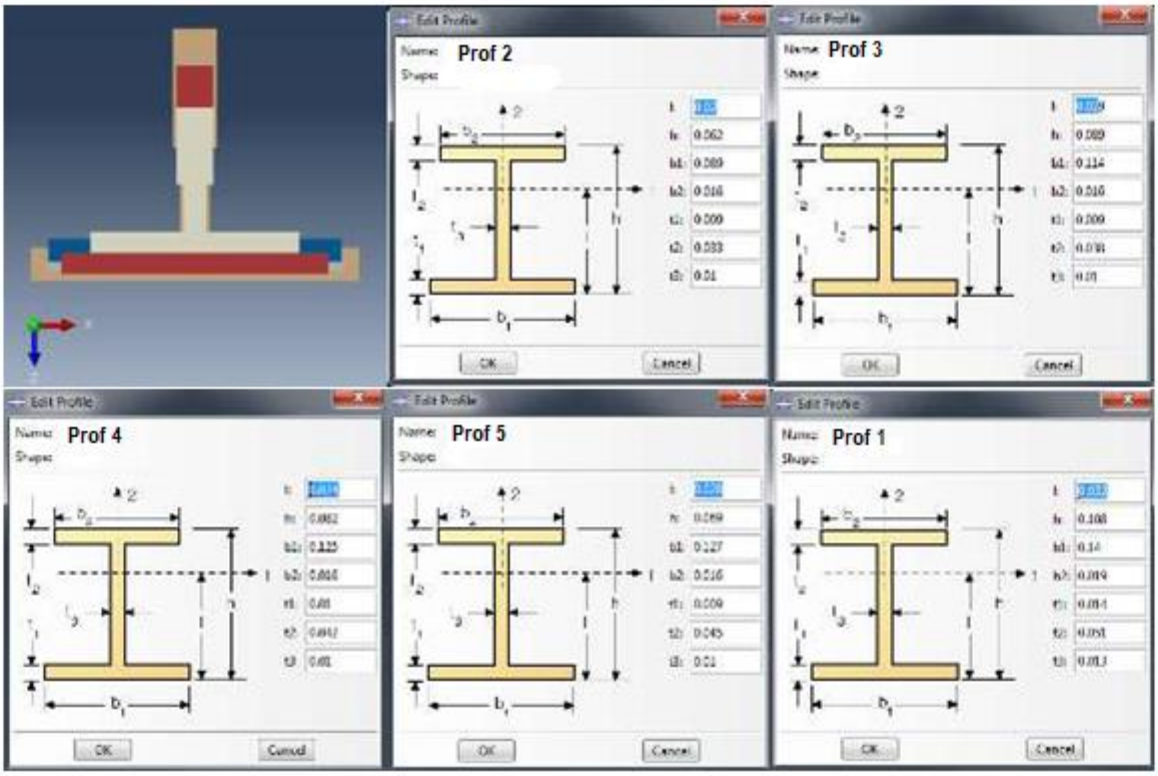


Figure 5. T-profile sections for beam element model.

3.2 Abaqus Buckling Solution Step

The buckling solution step was added afterwards to the same abaqus job as a single buckling step with a single load at the centermost midspan. Only one buckling case was considered due to the equal lengths of the spans.

The first eigenmode was read as the buckling load result. A total of ten eigenmodes was computed.

3.3 Abaqus Dynamic Implicit Solution Step

The dynamic implicit solution starts with the gravitation step. In this step the gravity and the stationary guide load pair is onset to the beam model. This step is nonphysical and intended for setting up the loads and letting the transient vibrations decay.

Actual results are read from the next dynamic implicit step, which is 5 seconds in duration with a 50 ms time step. Guide loads move with the defined constant velocity representing the elevator car speed and the defined constant separation distance representing the elevator car height.

3.4 HEEDS Optimization Setup

HEEDS optimization was used with the Abaqus dynamic implicit beam element model.

The Abaqus model is edited for the relevant geometry and analysis changes and the replay file (rpy) is recorded. This file is a Python script defining the Abaqus CAE actions to the model. This Python file together with the cae and inp files are linked to HEEDS and the rest of the definitions and tagging done within HEEDS user interface.

The optimization problem set up is multi-objective type (MO-SHERPA). The line mass and the maximum deflection are to be minimized and the total length of the beam is to be maximized.

Limiting values for constraints are applied for the maximum resultant stress, maximum deflection, maximum rotation and minimum buckling load value.

The modifications are made to the T-profile as a discrete set of five selected profiles. The five equal spans are varied in length between the limiting values.

Both translational and rotational stiffness for the bracket supports are varied by orders of magnitude using a discrete set of values. The car speed and car height is varied. The speed is set from a discrete set and the height varies between the limiting values.

The problem set up is summarized in the table below. The discrete sets that were varied are shown in the tables. They are the five T-profile sections, three car speeds, five translational and five rotational spring coefficients for the bracket supports. The other variables are shown in the optimization summary table. A total of 300 designs are analysed with 214 feasible designs and 31 errors as results.

Study Details	
Study name:	ImplicitBeam
Location:	D:/HEEDS_OptimizationSoftware_Study/HEEDS_Testing/ImplicitBeamWithBuckling
Status of Study:	Completed
Type:	Parametric Optimization
Method:	Multi-Objective SHERPA (MO-SHERPA)
Started on:	huhtikuu 18, 2013 @ 16:29:24
Last updated on:	huhtikuu 19, 2013 @ 9:17:18
Design Information	
# Designs completed:	300 designs
# Error designs:	20 designs
Optimization Statement	
Objective:	Minimize LineMass
	Minimize MaxDeflection
	Maximize L_total
Subject to:	MaxStress <= 1e+08
(4 constraints)	MaxDeflection <= 0.005
	MaxRotation <= 0.05
	F_Buckling >= 150000
By modifying:	Profile (discrete)
(6 variables)	1.5 <= L_1 <= 3.5
	k_trans (discrete) [1e7, 5e7, 1e8, 5e8, 1e9]
	k_rot (discrete) [5e7, 1e8, 5e8, 1e9, 5e9]
	U_car (discrete) [4, 6, 10]
	4 <= L_car <= 7.5

Figure 6. HEEDS optimization set up. Summary table for Abaqus dynamic implicit model.

Set	Ordered
Profile_Set	Yes
U_car_Set	Yes
k_trans_Set	Yes
k_rot_Set	Yes

	Items
1	Prof 2
2	Prof 3
3	Prof 4
4	Prof 5
5	Prof 1

Set	Ordered
Profile_Set	Yes
U_car_Set	Yes
k_trans_Set	Yes
k_rot_Set	Yes

	Items
1	4
2	6
3	10

Set	Ordered
Profile_Set	Yes
U_car_Set	Yes
k_trans_Set	Yes
k_rot_Set	Yes

	Items
1	1e7
2	5e7
3	1e8
4	5e8
5	1e9

Set	Ordered
Profile_Set	Yes
U_car_Set	Yes
k_trans_Set	Yes
k_rot_Set	Yes

	Items
1	5e7
2	1e8
3	5e8
4	1e9
5	5e9

Figure 7. HEEDS parameters. Discrete sets used with Abaqus dynamic implicit model.

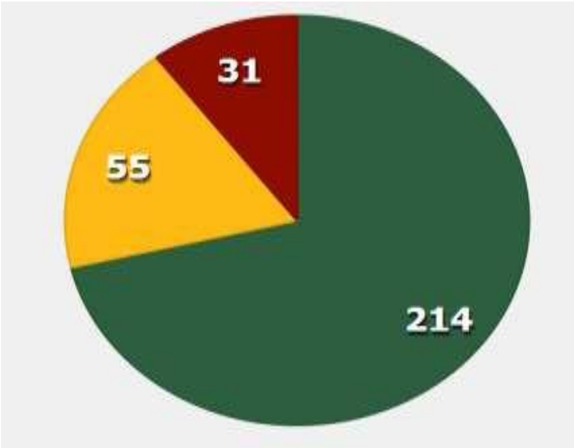


Figure 8. HEEDS optimization run. Feasible, infeasible (constraints violated) and error (no result obtained from model) cases. From Abaqus dynamic implicit model optimization.

4. Discussion - Limitations

Optimization with a dynamic model needs simplifications in model size to keep the solution time reasonably short. The following limitations of the model are described here.

The model is simple and represents only a single guiderail with spring supports and a travelling guide load pair.

This model set up could be seen as only a part of the relevant dynamic system of two guiderails, four roller wheel sets and the elevator car. The whole dynamic system modelling with the elevator car included is closer to the multi body simulation (MBS) work.

The T-profile is simplified from the shapes and dimensions of the real profiles. Loading is simplified to a guide load pair in one plane and the load values are fixed. Location of the loads relative to the torsional center is approximate.

The translational and rotational stiffness of the support brackets are varied separately as parameters without any connection to the actual bracket design geometry.

The rail-to-rail fishplate connections are not included. Therefore, the beam has a continuous stiffness defined by the T-profile section geometry.

The selected multi-span beam with six supports and five spans is a local representation of the entire guiderail. It is an arbitrary choice between the 300 m global model of say, 50 spans and a single span beam model.

The optimization problem is defined only by maximizing the total span and minimizing the specific weight, i.e. beam mass per unit length. The individual variables are constrained at rather arbitrary lower and upper limits. The variable constraints could be more detailed in later stages.

The set up with five predefined T-profiles as a discrete set variable fixes the beam section stiffness to five values. This leads to result sets that are in five branches corresponding to the T-profiles' bending stiffness, since the material properties are fixed.

5. Results

The results section is divided to Abaqus results, Excel results, Abaqus with HEEDS results and Excel with HEEDS results.

5.1 Results from Abaqus Model

Some results from the design number 179 are shown here. Result views are from the buckling solution and the dynamic implicit Abaqus solution. The results give an overview of the computation of one single optimization design. Design number 179 is selected as an example as it is the highest ranking design for the heaviest T-profile (profile 1) from the five profile discrete set.

5.1.1 Abaqus Buckling Results

The unit load is applied at the central midspan with a 0.3 m eccentricity offset from the beam neutral axis in the positive x-direction. The unit load is in the negative y-direction. The first and lowest eigenvalue is +1.00777e6, indicating a buckling load of 1000 kN in the negative y-direction. The analysis model lowest buckling load result needs to be more than the design buckling load of 150 kN calculated according to the elevator standard [2].

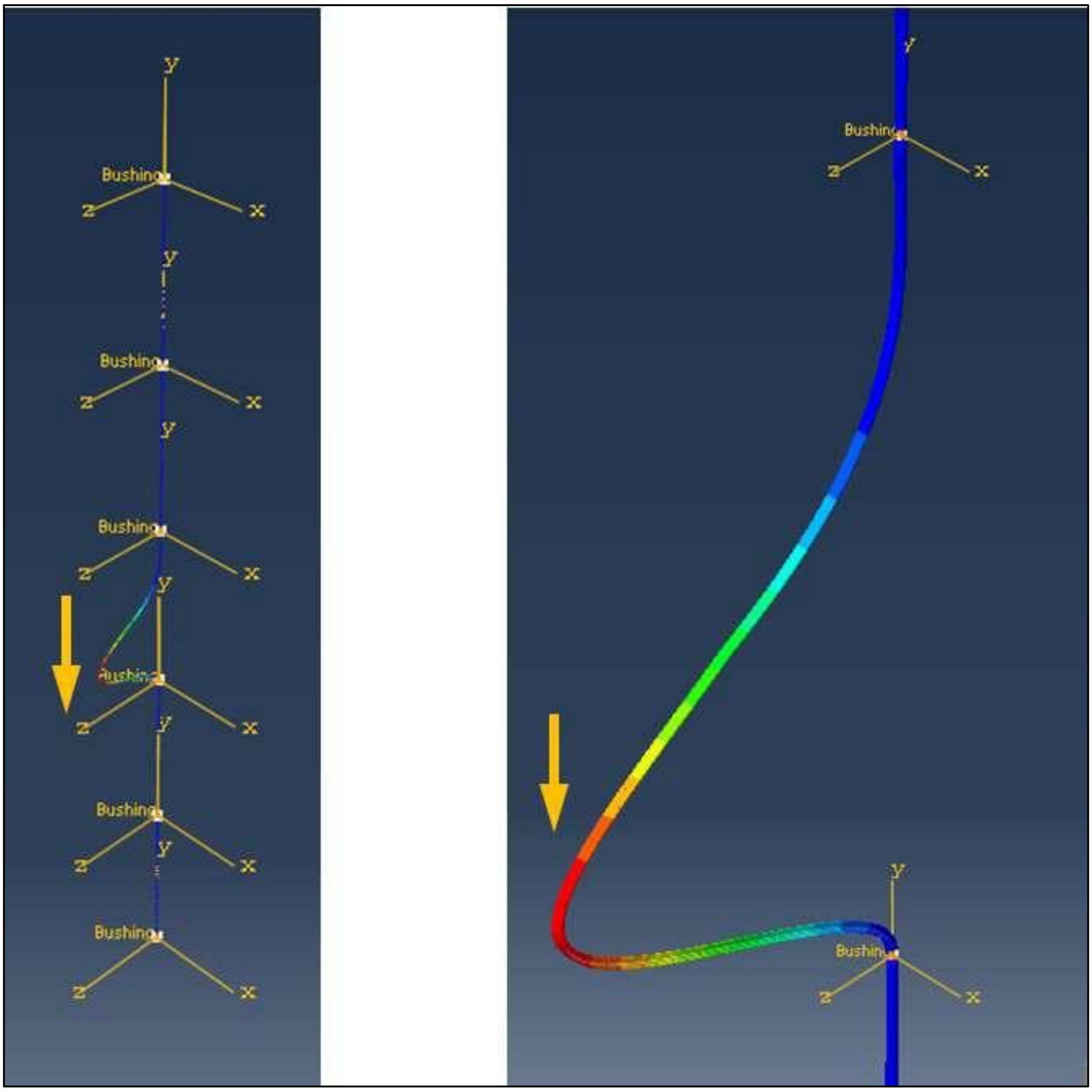


Figure 9. Abaqus linear buckling load result view. Result example is from HEEDS design number 179.

5.1.2 Abaqus Dynamic Implicit Results

The Abaqus dynamic implicit solution is a 5 s long simulation with a time step size of 50 ms. Images from the simulation from HEEDS design number 179 is shown below. Images are snapshots taken with 0.5 s intervals to give an overview of the simulation. The first frame shows the base state of the model. The second frame is at t=0.0 s, the third frame is at t=0.5 s, the last frame is at t=5.0 s. The gravity step with load introduction transients is not included in the actual results and is not shown in the figure below.

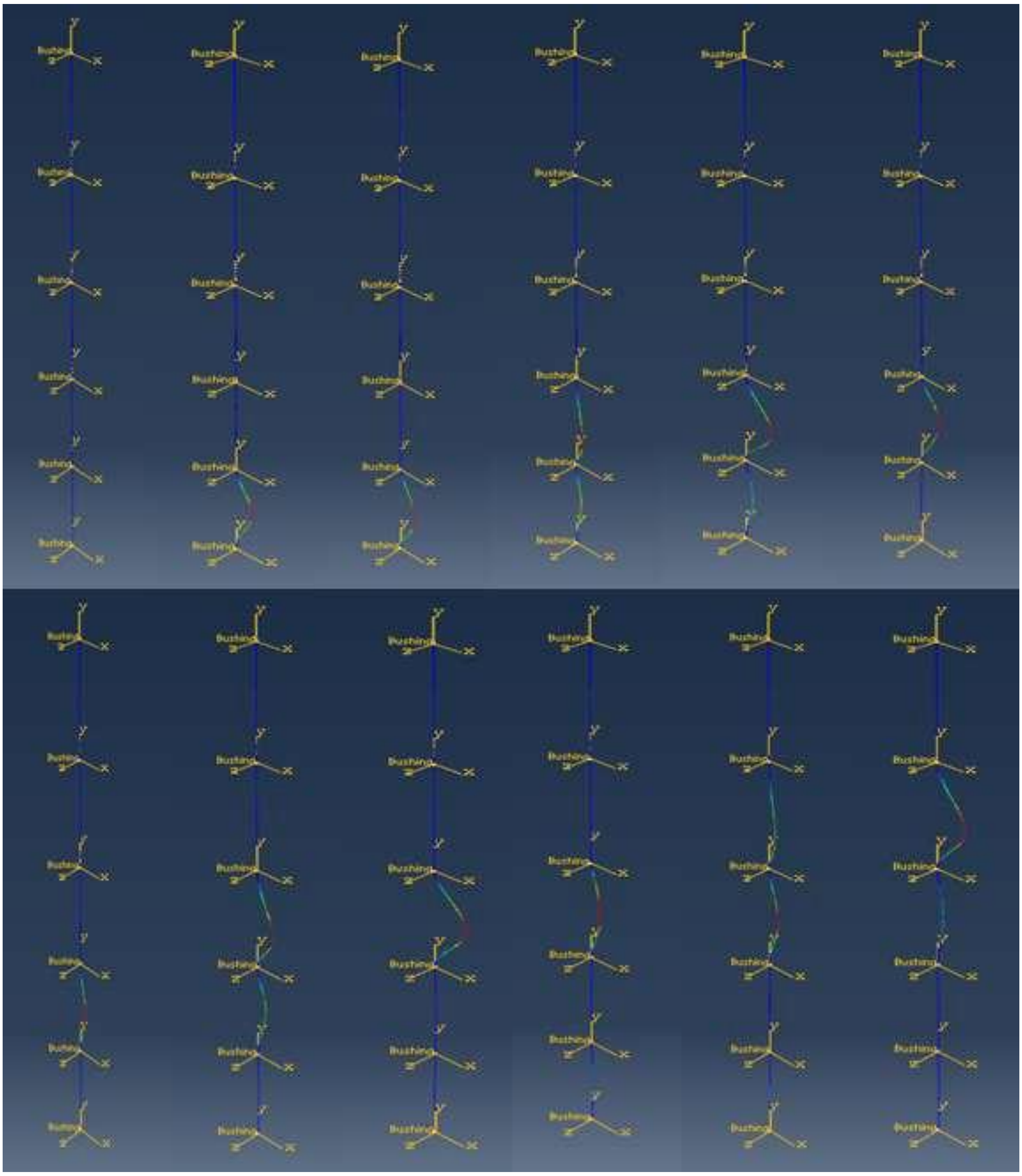


Figure 10. Snapshots with a 0.5 s interval from 0…5 s from a dynamic implicit simulation. The two travelling loads are located at the maximum deflections. First frame shows the reference model without loading. Result is from HEEDS design number 179.

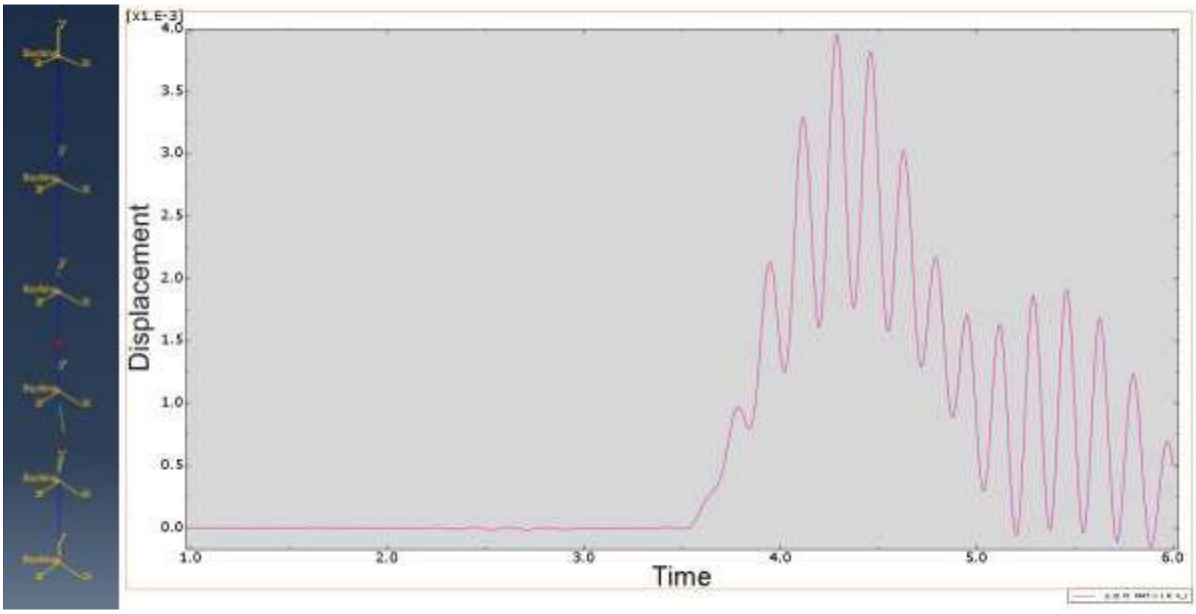


Figure 11. Displacement as a function of time from a mid-span node shown on the left. Static deflection dominates and the undamped dynamic oscillation is superimposed to the static deflection during and after the loads pass by the node location. This result view is a qualitative example from HEEDS design number 179.

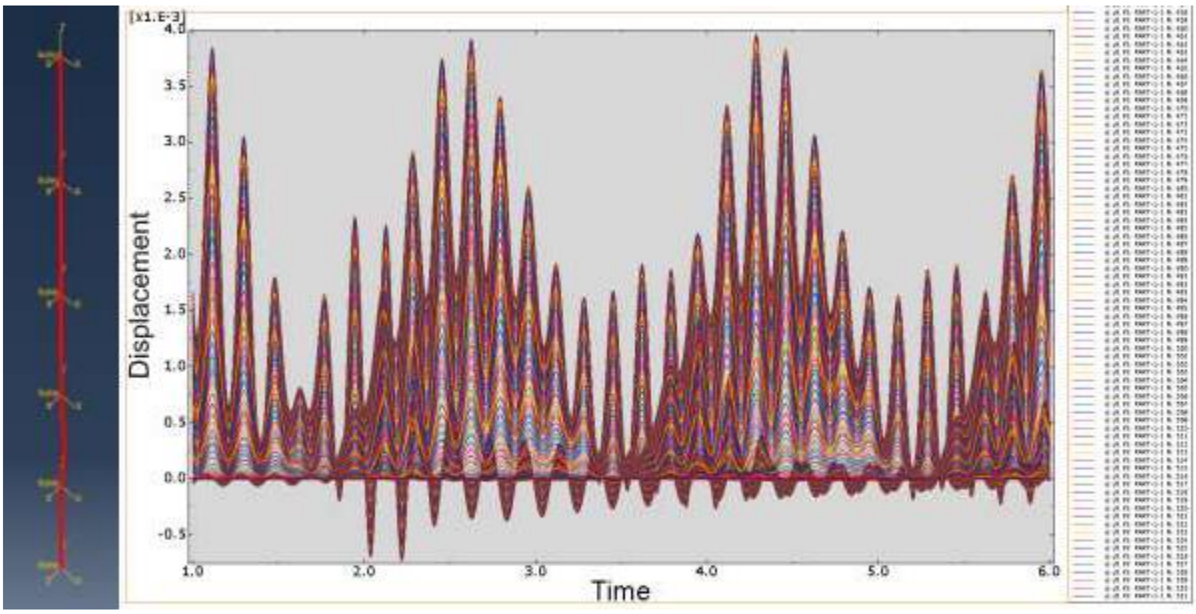


Figure 12. Displacement as a function of time from all nodes shown on the left. Over all minimum and maximum deflection values can be seen to be between -0.7 mm to +3.9 mm. This result view is a qualitative example from HEEDS design number 179.

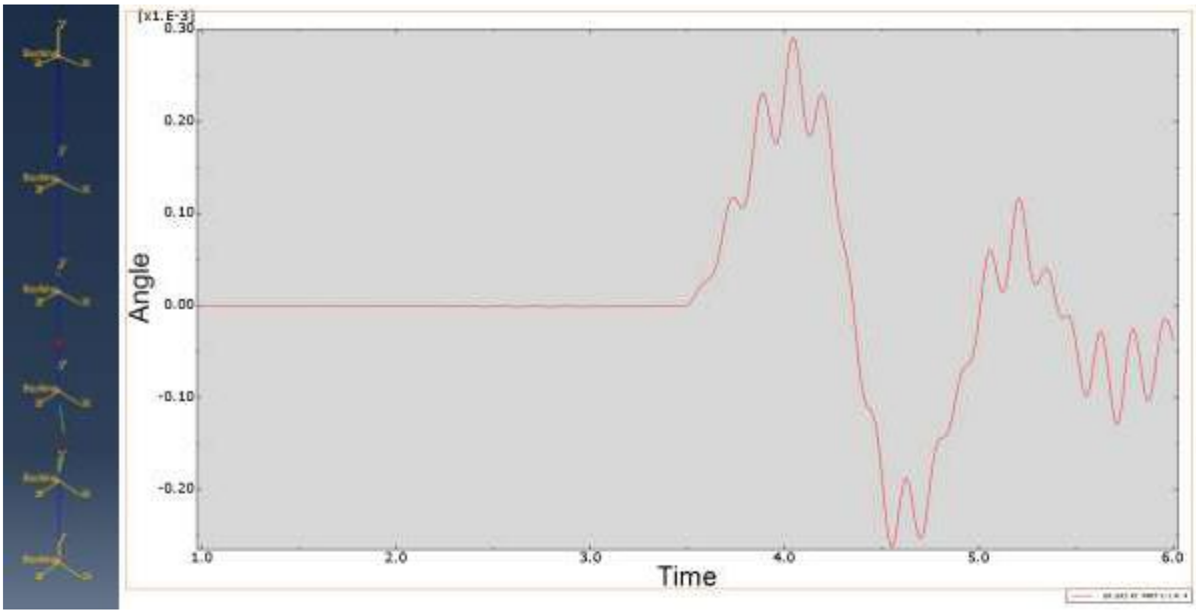


Figure 13. Beam rotation (bending gradient) as a function of time from a mid-span node shown on the left. Static deflection dominates in rotation angle too. The undamped dynamic oscillation is superimposed to the static rotation angle during and after the loads pass by the node location. This result view is a qualitative example from HEEDS design number 179.

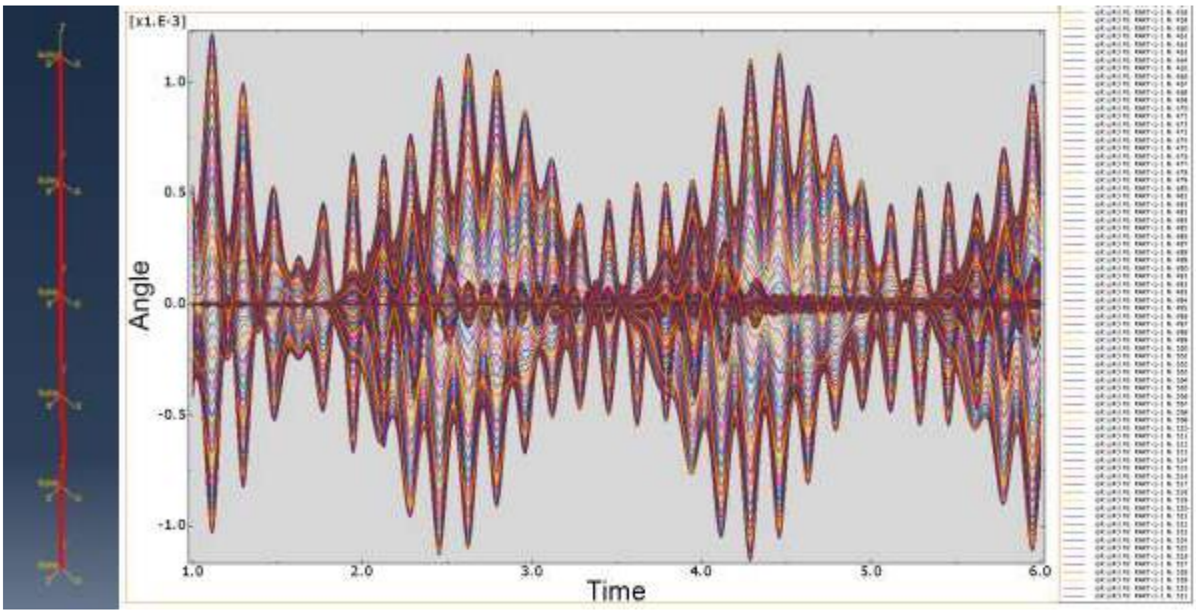


Figure 14. Beam rotation (bending gradient) as a function of time from all nodes shown on the left. Over all minimum and maximum deflection values can be seen to be from -1.1 mrad to +1.2 mrad. This result view is a qualitative example from HEEDS design number 179.

5.2 Results from Excel Beam Calculation

Resulting values are measured by a beam length of 2.97 m, beam profile 1 and the spring stiffness adjusted to match the calculated deflection. Loads are maximum values selected individually from the two available ride comfort load cases (F.x.6.1 and F.y.6.2)[4].

Reference results are matched to measured deflection calculation with profile **1**. Reference result values are: deflection δ_x=0.138 mm support stiffness **k=0.6 MNm/rad**, buckling force 2.0 MN. The bending angle is 62 mrad (milliradians) as per definition, but due to units mismatch compared to the Mathcad calculation it is divided by 1000. Thus the value is 62 µrad (microradians), which corresponds to a relative value of 33%.

When this relative ride comfort parameter is increased to the upper limit of 100 % by decreasing the spring stiffness, we find the lowest acceptable value for the support stiffness. However, with the guiderail profile T1, and a zero stiffness value (hinge supports) we get relative ride comfort of only 58 %, implying that the T-profile could be chosen to a smaller size.

By selecting the profile 2 we get a relative ride comfort value of 115 %. Then increasing the support stiffness value to k=1.30 MNm/rad we arrive to relative ride comfort value of 100 %.

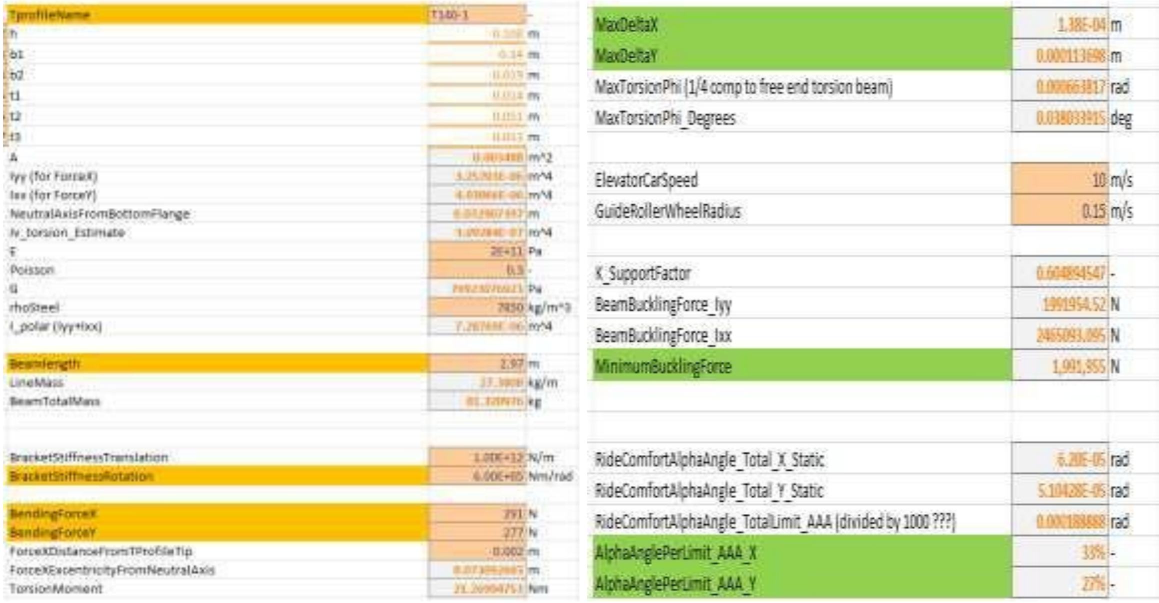
Result values for all seven T-profiles with the original reference support stiffness value of **k=0.6 MNm/rad** are shown in the table below.

The values corresponding to relative ride comfort of 100 % for profile **2** are: deflection δ_x=0.421 mm, stiffness **k=1.30 MNm/rad**, buckling force 0.47 MN, bending angle 189 µrad, relative value 100 % as seen in the Excel view figures below.

The values corresponding to relative ride comfort of 100 % for profile **3** are: deflection δ_x=0.420 mm, stiffness **k=0.175 MNm/rad**, buckling force 0.66 MN, bending angle 188 µrad, relative value 100 % as seen in the Excel view figures below.

Table 1. Results from Excel model with measurement numbers and support stiffness k = 0.6 MNm/rad.

TprofileName	UNIT	1	2	3
h	m	0.108	0.062	0.089
b1	m	0.14	0.089	0.114
b2	m	0.019	0.016	0.016
t1	m	0.014	0.009	0.009
t2	m	0.051	0.033	0.038
t3	m	0.013	0.01	0.01
Beamlength	m	2.97	2.97	2.97
LineMass	kg/m	27.4	12.0	16.1
BracketStiffnessRotation	Nm/rad	6.00E+05	6.00E+05	6.00E+05
BendingForceX	N	291	291	291
ElevatorCarSpeed	m/s	10	10	10
StaticMaxDeltaX_Total	m	0.000138	0.000482	0.000282
K_SupportFactor (1.0 for pure hinge supports, 0.5 for pure clamp supports)	-	0.60	0.52	0.54
MinimumBucklingForce	N	2.E+06	4.E+05	9.E+05
BendingAngle_Total_X_Static	rad	0.000062	0.000216	0.000126
BendingAngle_TotalLimit_AAA (divided by 1000 ???)	rad	0.000189	0.000189	0.000189
AnglePerLimit_AAA_X	-	33%	115%	67%



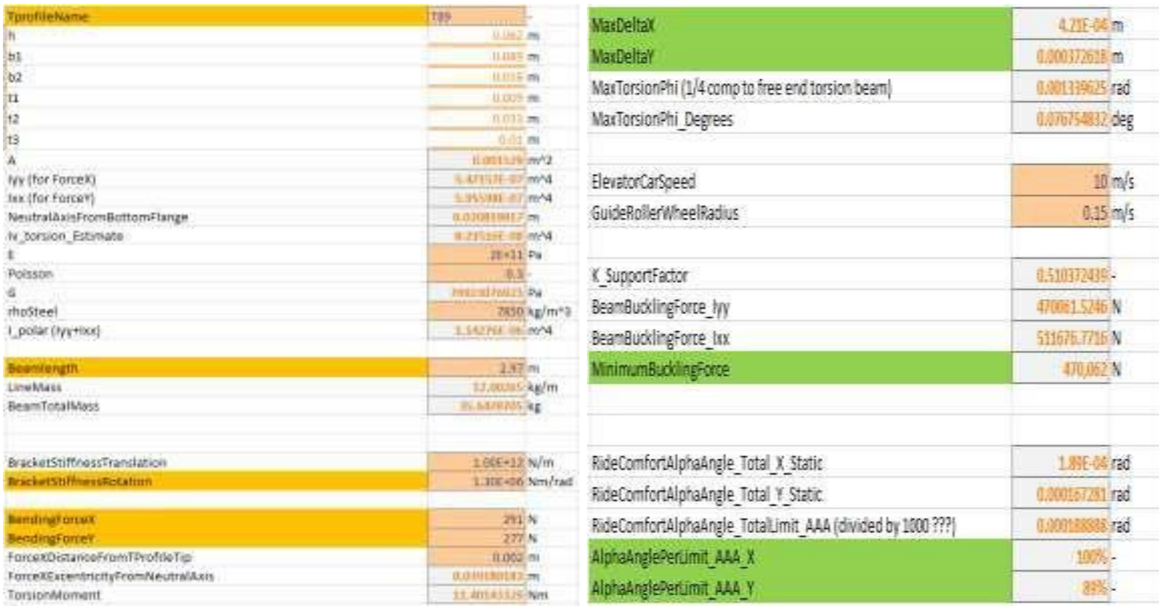
TprofileName	T140-1	-
h	[illegible]	m
b1	0.14	m
b2	[illegible]	m
t1	[illegible]	m
t2	[illegible]	m
t3	[illegible]	m
A	[illegible]	m^2
Iyy (for ForceX)	[illegible]	m^4
Ixx (for ForceY)	[illegible]	m^4
NeutralAxisFromBottomFlange	[illegible]	m
Iv_torsion_Estimate	[illegible]	m^4
E	2E+11	Pa
Poisson	0.3	-
G	[illegible]	Pa
rhoSteel	7850	kg/m^3
I_polar (Iyy+Ixx)	[illegible]	m^4
Beamlength	2.97	m
LineMass	[illegible]	kg/m
BeamTotalMass	[illegible]	kg
BracketStiffnessTranslation	1.00E+12	N/m
BracketStiffnessRotation	6.00E+05	Nm/rad
BendingForceX	291	N
BendingForceY	277	N
ForceXDistanceFromTProfileTip	0.002	m
ForceXExcentricityFromNeutralAxis	[illegible]	m
TorsionMoment	[illegible]	Nm

MaxDeltaX	1.38E-04	m
MaxDeltaY	0.000113698	m
MaxTorsionPhi (1/4 comp to free end torsion beam)	0.000663817	rad
MaxTorsionPhi_Degrees	0.038033915	deg
ElevatorCarSpeed	10	m/s
GuideRollerWheelRadius	0.15	m/s
K_SupportFactor	0.604894547	-
BeamBucklingForce_Iyy	1991954.52	N
BeamBucklingForce_Ixx	2465093.095	N
MinimumBucklingForce	1,991,955	N
RideComfortAlphaAngle_Total_X_Static	6.28E-05	rad
RideComfortAlphaAngle_Total_Y_Static	5.10428E-05	rad
RideComfortAlphaAngle_TotalLimit_AAA (divided by 1000 ???)	0.000188888	rad
AlphaAnglePerLimit_AAA_X	33%	-
AlphaAnglePerLimit_AAA_Y	27%	-

*Figure 15. Example view of the Input and output values from the Excel calculation model. With measurement values, profile 1 and **reference** rotation stiffness parameter k=6e5.*

5.2.1 Excel Model Optimized To Full AAA Ride Comfort Limit – Profile 2 and Increased Bracket Stiffness

The full utilization of the AAA ride comfort limit with the smaller profile 2 needs and increased support stiffness. Spring coefficient k needs to go from 6.0e5 Nm/rad to 1.3e6 Nm/rad. If the stiffness is set to the initial value, the relative ride comfort value would be



TprofileName	T89	-
h	[illegible]	m
b1	[illegible]	m
b2	[illegible]	m
t1	[illegible]	m
t2	[illegible]	m
t3	0.01	m
A	[illegible]	m^2
Iyy (for ForceX)	[illegible]	m^4
Ixx (for ForceY)	[illegible]	m^4
NeutralAxisFromBottomFlange	[illegible]	m
Iv_torsion_Estimate	[illegible]	m^4
E	2E+11	Pa
Poisson	0.3	-
G	[illegible]	Pa
rhoSteel	7850	kg/m^3
I_polar (Iyy+Ixx)	[illegible]	m^4
Beamlength	2.97	m
LineMass	[illegible]	kg/m
BeamTotalMass	[illegible]	kg
BracketStiffnessTranslation	1.00E+12	N/m
BracketStiffnessRotation	1.30E+06	Nm/rad
BendingForceX	291	N
BendingForceY	277	N
ForceXDistanceFromTProfileTip	0.002	m
ForceXExcentricityFromNeutralAxis	[illegible]	m
TorsionMoment	[illegible]	Nm

MaxDeltaX	4.21E-04	m
MaxDeltaY	0.000372618	m
MaxTorsionPhi (1/4 comp to free end torsion beam)	0.001339625	rad
MaxTorsionPhi_Degrees	0.076754832	deg
ElevatorCarSpeed	10	m/s
GuideRollerWheelRadius	0.15	m/s
K_SupportFactor	0.510372439	-
BeamBucklingForce_Iyy	470061.5246	N
BeamBucklingForce_Ixx	511676.7716	N
MinimumBucklingForce	470,062	N
RideComfortAlphaAngle_Total_X_Static	1.89E-04	rad
RideComfortAlphaAngle_Total_Y_Static	0.000167281	rad
RideComfortAlphaAngle_TotalLimit_AAA (divided by 1000 ???)	0.000188888	rad
AlphaAnglePerLimit_AAA_X	100%	-
AlphaAnglePerLimit_AAA_Y	89%	-

*Figure 16. Example view of the Input and output values from the Excel calculation model. Measurement values adjusted to AAA limit by selecting profile 2 and **increasing** rotation stiffness parameter k.*

5.2.2 Excel Model Optimized To Full AAA Ride Comfort Limit – Profile 3 and Decreased Bracket Stiffness

TprofileName	T114	-
h	0.085	m
b1	0.114	m
b2	0.016	m
t1	0.009	m
t2	0.038	m
t3	0.01	m
A	0.002054	m^2
Iyy (for ForceX)	1.11396E-06	m^4
Ixx (for ForceY)	1.78821E-06	m^4
NeutralAxisFromBottomFlange	0.029299903	m
Iv_torsion_Estimate	1.04835E-07	m^4
E	2E+11	Pa
Poisson	0.3	-
G	76923076923	Pa
rhoSteel	7850	kg/m^3
I_polar (Iyy+Ixx)	2.93457E-06	m^4
Beamlength	2.97	m
LineMass	16.1239	kg/m
BeamTotalMass	47.887983	kg
BracketStiffnessTranslation	1.00E+09	N/m
BracketStiffnessRotation	1.75E+05	Nm/rad
BendingForceX	291	N
BendingForceY	277	N
ForceXDistanceFromTProfileTip	0.002	m
ForceXExcentricityFromNeutralAxis	0.057700097	m
TorsionMoment	16.79072833	Nm

MaxDeltaX	4.20E-04	m
MaxDeltaY	0.000290254	m
MaxTorsionPhi (1/4 comp to free end torsion beam)	0.001546275	rad
MaxTorsionPhi_Degrees	0.088595009	deg
ElevatorCarSpeed	10	m/s
GuideRollerWheelRadius	0.15	m/s
K_SupportFactor	0.619999241	-
BeamBucklingForce_Iyy	660263.1941	N
BeamBucklingForce_Ixx	1036549.462	N
MinimumBucklingForce	660,263	N
RideComfortAlphaAngle_Total_X_Static	1.88E-04	rad
RideComfortAlphaAngle_Total_Y_Static	0.000130305	rad
RideComfortAlphaAngle_TotalLimit_AAA (divided by 1000 ???)	0.000188888	rad
AlphaAnglePerLimit_AAA_X	100%	-
AlphaAnglePerLimit_AAA_Y	69%	-

*Figure 17. Example view of the Input and output values from the Excel calculation model. Measurement values adjusted to AAA limit by selecting profile 3 and **decreasing** rotation stiffness parameter k.*

5.3 HEEDS Results with Abaqus Dynamic Model

The dynamic implicit Abaqus model was used with HEEDS for one optimization run with 300 cycles.

HEEDS tries to find designs which have a linemass as low as possible and a total span as long as possible. The maximum deflection value is limiting. The discrete T-profile sections were used. Results are shown in the table below.

It can be seen from rank 1 and 2 results that a length of 8.53 m can be covered with a single beam span using the profile 3 (16.1 kg/m) and with a deflection limit of 5 mm. From rank 3 to 8 results a longer span of 10.565 m is covered but with a heavier profile 1 (27.4 kg/m). Result is partial and difficult to grasp, since other variables (translation support stiffness, rotation support stiffness, car speed, car height/load spacing) are changed simultaneously.

Design Id	46	61	179	269	237	257	157	203	299	206
Flag	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE
Rank	1	2	3	4	5	6	7	8	9	10
Mass	687.6844	687.6844	1446.391	1446.391	1446.391	1446.391	1446.391	1446.391	598.1967	598.1967
LineMass	16.1239	16.1239	27.3808	27.3808	27.3808	27.3808	27.3808	27.3808	16.1239	16.1239
MaxStress	22921844	22908736	15541878	15660038	16097966	15885000	16408263	20276048	20000640	20020222
MaxDeflection	0.004682	0.004682	0.003964	0.00397	0.004472	0.004573	0.004614	0.004732	0.00348	0.003489
MaxRotation	0.000606	0.000606	0.000327	0.000329	0.000369	0.000379	0.00038	0.000397	0.000606	0.000607
MaxAcceleration	2.713494	2.714597	1.895969	1.958848	2.134178	2.071949	2.174394	2.475411	1.431506	1.476088
L_total	42.65	42.65	52.825	52.825	52.825	52.825	52.825	52.825	37.1	37.1
Eigenvalue	426208	426163	1007770	1009450	1007770	1007770	1007770	764356	-682305	-682305
F_Buckling	426208	426163	1007770	1009450	1007770	1007770	1007770	764356	682305	682305
Performance	26.52142	26.52142	25.44024	25.44023	25.43973	25.43963	25.43959	25.43947	20.97262	20.97261
Profile	3	3	1	1	1	1	1	1	3	3
L_1	8.53	8.53	10.565	10.565	10.565	10.565	10.565	10.565	7.42	7.42
k_trans	1	1	5	5	5	4	5	1	5	5
k_rot	4	3	2	4	2	2	2	1	3	3
U_car	3	3	2	2	2	2	2	1	1	1
L_car	5.4	5.4	6.31	6.31	5.505	4.14	5.155	6.31	5.54	5.505

Figure 18. Results from the Abaqus dynamic implicit model. Ranking is arbitrary and the actual ranking of results should be evaluated by the user. See figure 7 in the end of chapter 3 for actual values for discrete sets.

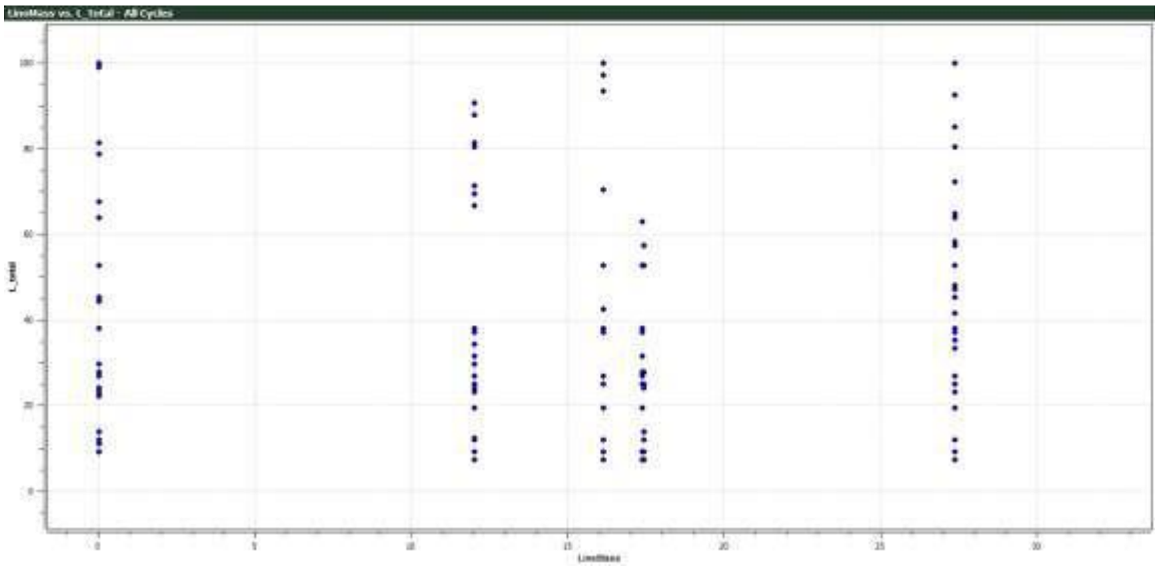


Figure 19. With five preselected profiles the linemass has five discrete values. Mass optimization is not continuous.

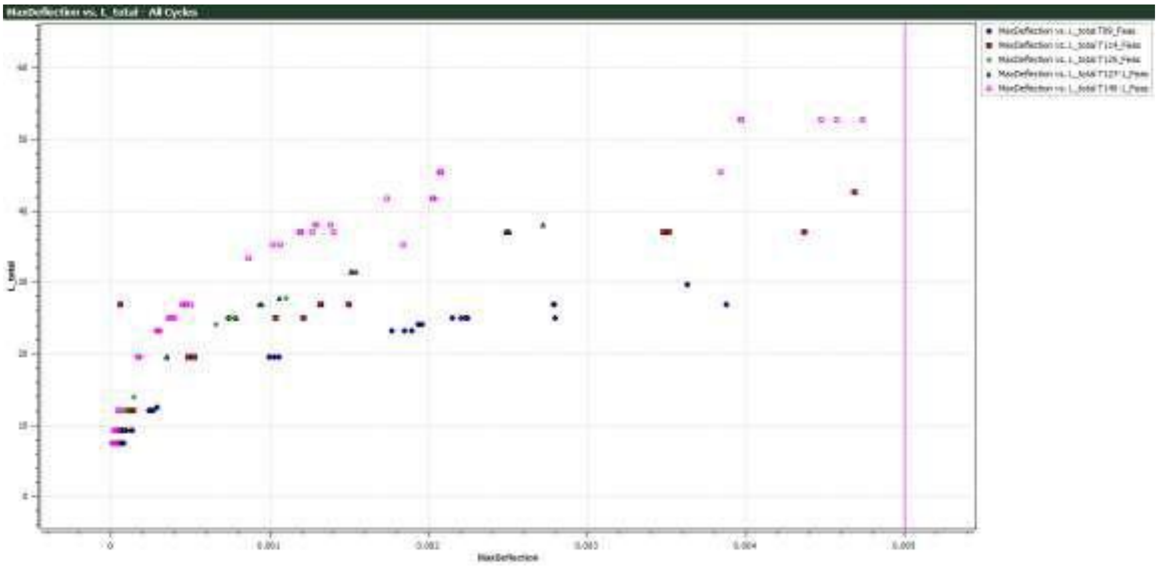


Figure 20. With five preselected profiles the results are in five branches for each profile.

5.4 HEEDS Results with Excel Beam Calculation

The Excel model calculates the maximum deflection, the bending angle and the buckling load for a single span beam with coil spring supports.

The model is connected to HEEDS with different kinds of multi-objective optimization set ups with different kinds of objective variables to be minimized or maximized together with several constraints. Both the discerete profile set and free formed profile geometries are used.

5.4.1 Results With Discrete Profile Set

With the preselected discrete profile set, Excel model results are qualitatively similar to the Abaqus dynamic implicit model. Five branches of results corresponding to five profiles.

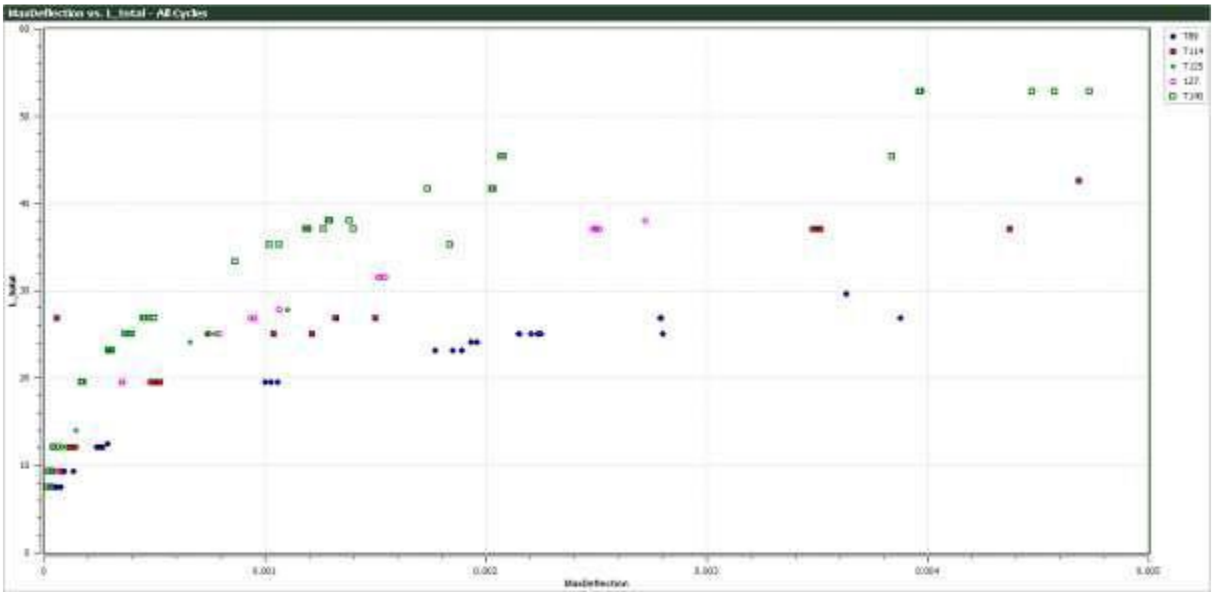


Figure 21. Excel results using the five preselected profiles. X: Max deflection, Y: Total beam length.

5.4.2 Results With Free Profile Dimensions

Results are more continuous when optimizing with free T-profile dimensions. Free dimensions allow for all kinds of combinations of bending stiffness, torsion stiffness and linemass. Multi-objective optimization results are Pareto fronts as shown in the figure below.

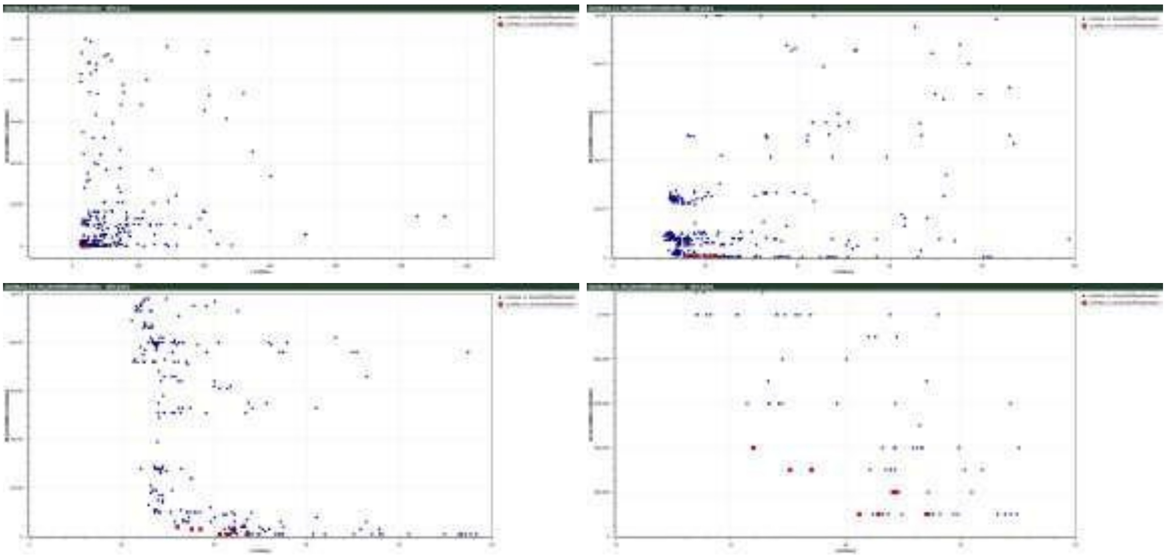


Figure 22. Visual view of the Pareto front when beam length is maximized, linemass minimized and bracket rotation stiffness minimized.
***X-axis** is linemass, **Y-axis** is support spring rotation stiffness. **Blue dots** are all 683 feasible designs from 1000 cycles. **Red dots** are the ranked top ten designs. View is zoomed to top ten Pareto front starting from the top left view to the bottom right view. Decreasing spring stiffness correlates with higher linemass profiles.*

Table 2. Top five ranking results with the Excel beam model and free profile geometry. Torsion angle constraint is 0.5 mrad.

Rank	1	2	3	4	5
Time	15:16:31	15:16:23	15:18:36	15:17:23	15:16:26
Cycle #	39	38	48	43	38
Evaluation #	780	766	976	863	771
Design Source	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy
Design Flag	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE
performance	0.210641231	0.202624031	0.19983837	0.190856388	0.183523074
h	0.075	0.072	0.067	0.065	0.074
b1	0.107	0.107	0.099	0.099	0.108
b2	0.0535	0.05243	0.04851	0.04851	0.05616
t1	0.0195	0.018	0.01675	0.013	0.01924
t2	0.01221	0.00972	0.009045	0.01092	0.0120472
t3	0.005885	0.0041944	0.0033957	0.0053361	0.0067392
A	0.002994497	0.002621348	0.002236943	0.002035936	0.003042341
Iyy_for_ForceX	2.14766E-06	1.9547E-06	1.44061E-06	1.15583E-06	2.19926E-06
Ixx_for_ForceY	1.92441E-06	1.51088E-06	1.11815E-06	1.13624E-06	1.92685E-06
NeutralAxisFromBottomFlange	0.025696824	0.022768623	0.021012602	0.023433288	0.025938403
Iv_torsion_Estimate	9.97523E-07	7.5719E-07	5.59771E-07	5.48917E-07	1.08869E-06
I_polar_IyyIxx	4.07206E-06	3.46558E-06	2.55877E-06	2.29208E-06	4.12612E-06
Beamlength	3.46	3.12	3.38	3.48	3.42
LineMass	23.5067987	20.57757891	17.56000073	15.98209908	23.8823757
BeamTotalMass	81.33352351	64.2020462	59.35280248	55.61770478	81.6777249
BracketStiffnessRotation	100000	100000	300000	400000	100000
TorsionMoment	13.76522415	13.74433071	12.80033273	11.51391314	13.40392476
MaxDeltaX	0.00045874	0.000370232	0.000423941	0.000482992	0.000435588
MaxDeltaY	0.000476624	0.000431906	0.000477859	0.000465125	0.000460936
MaxTorsionPhi_14_comp_to_free_end_torsion_beam	0.000155174	0.000184059	0.000251195	0.000237235	0.000136847
MaxTorsionPhi_Degrees	0.008890833	0.010545796	0.014392422	0.013592561	0.007840774
ElevatorCarSpeed	10	10	10	10	10
K_SupportFactor	0.732966387	0.734090448	0.585917253	0.554527754	0.737247969
BeamBucklingForce_Iyy	659134.3669	735531.03	725055.3798	612661.2438	682853.948
BeamBucklingForce_Ixx	590616.9469	568526.4011	562762.0068	602276.0358	598272.3327
MinimumBucklingForce	590616.9469	568526.4011	562762.0068	602276.0358	598272.3327
BendingAngle_Total_X_Static	0.000176778	0.000158219	0.000167235	0.000185054	0.00016982
BendingAngle_Total_Y_Static	0.00018367	0.000184575	0.000188505	0.000178209	0.000179702
AnglePerLimit_AAA_X	94%	84%	89%	98%	90%
AnglePerLimit_AAA_Y	97%	98%	100%	94%	95%

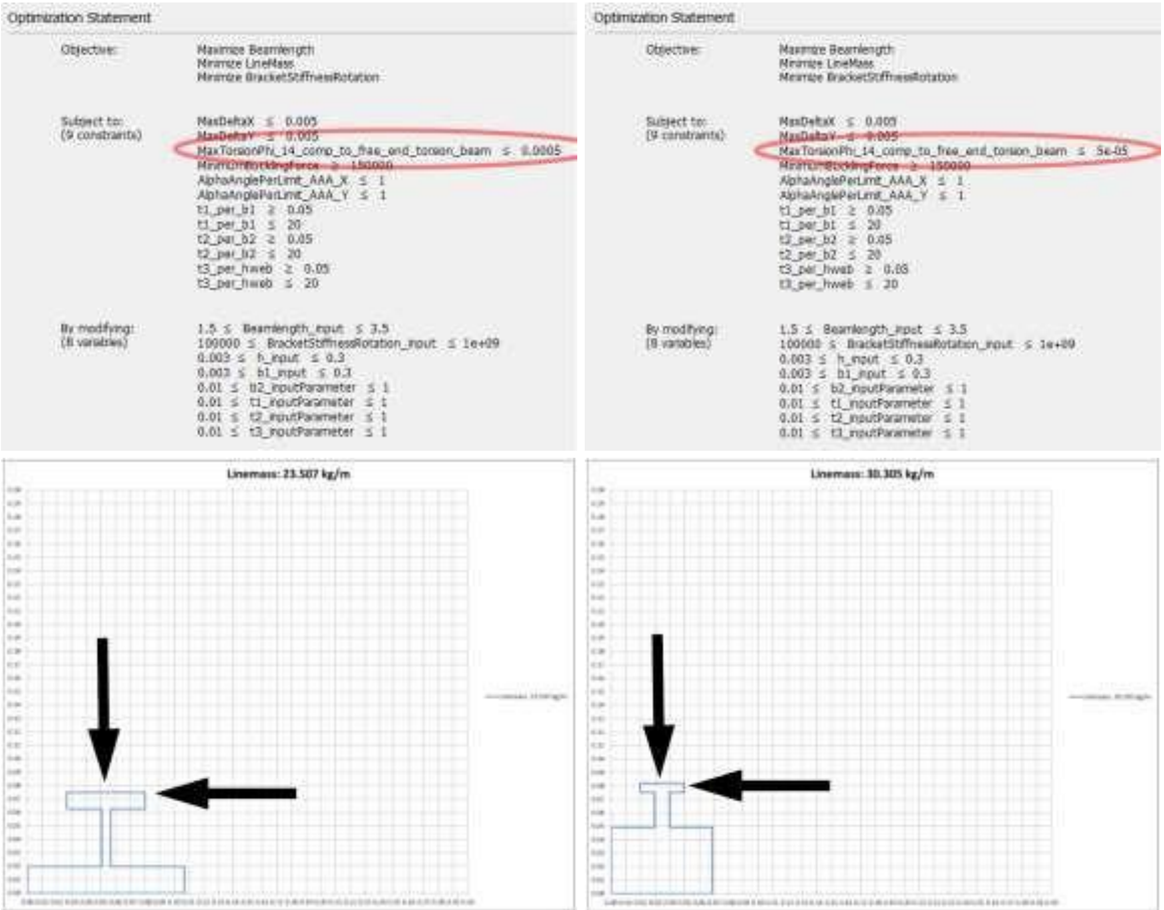


Figure 23. Two results from the Excel optimization. Only the torsion constraint is different for the optimization set up.
LEFT: Torsion angle constraint is 0.5 mrad. Bending is limiting the design and less torsion resistance is needed in result geometry.
RIGHT: Torsion angle constraint is 0.05 mrad. Torsion is dominating the design and more torsion resistance is needed in result geometry.

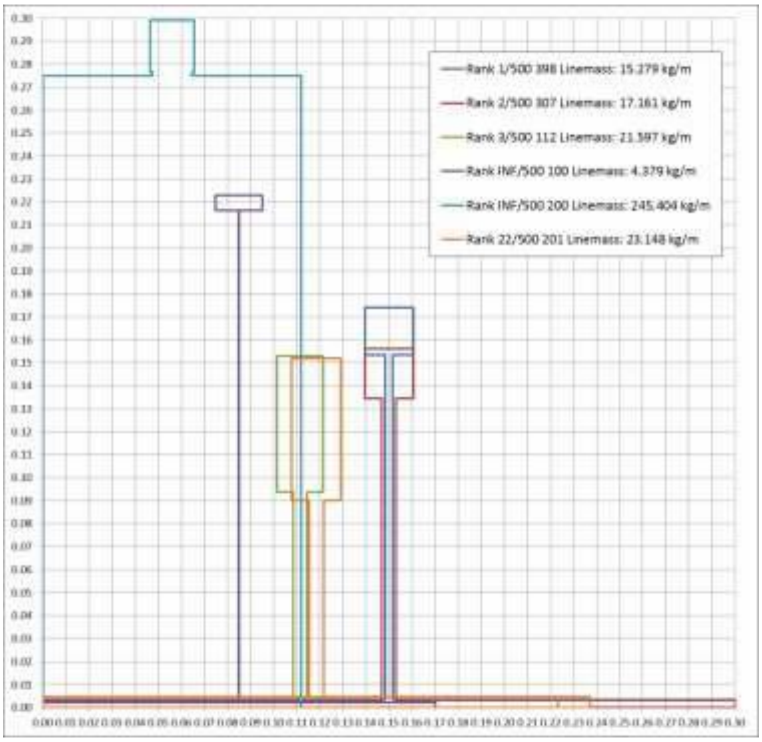


Figure 24. Samples of various feasible and non-feasable T-profile results when the cross section geometry was optimized. Additional constraint were used for minimum thickness, shape control and aspect ratio control. Slender results are seen when bending resistance is limiting. Bulky sections are seen when torsion resistance is limiting.

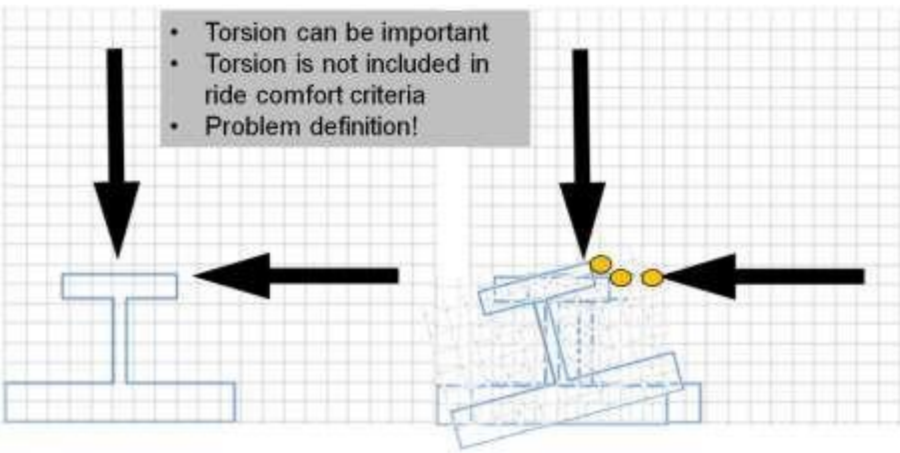


Figure 25. In some designs the torsional stiffness may dominate the apparent rigidity of the guiderail support. Torsion load is also a function of the T-profile section height at each desgn. Higher sections induce more torque when the guide force is defined to act at the top flange of the section and the support is at the bottom flange.

Table 3. Results for two optimization runs differing in torsion constraint value. Rank 1 result shown from both runs. Beam span lengths are different but the relative ride comfort values are both close to 100%.

Torsion Constraint	**mrad**	**0.5**	**0.05**
TprofileName	-	Free	Free
h	m	0.075	0.082
b1	m	0.107	0.069
b2	m	0.0535	0.02967
t1	m	0.0195	0.0492
t2	m	0.01221	0.006888
t3	m	0.005885	0.0100878
A	m^2	0.002994497	0.003860562
Iyy (for ForceX)	m^4	2.14766E-06	1.36527E-06
Ixx (for ForceY)	m^4	1.92441E-06	1.58996E-06
NeutralAxisFromBottomFlange	m	0.025696824	0.03316177
Iv_torsion_Estimate	m^4	9.97523E-07	3.145E-06
E	Pa	2E+11	2E+11
Poisson	-	0.3	0.3
G	Pa	76923076923	76923076923
rhoSteel	kg/m^3	7850	7850
I_polar (Iyy+Ixx)	m^4	4.07206E-06	2.95523E-06
Beamlength	m	3.46	2.86
LineMass	kg/m	23.5067987	30.30541196
BeamTotalMass	kg	81.33352351	86.67347822
BracketStiffnessTranslation	N/m	1.00E+09	1.00E+09
BracketStiffnessRotation	Nm/rad	1.00E+05	1.00E+05
BendingForceX	N	291	291
BendingForceY	N	277	277
ForceXDistanceFromTProfileTip	m	0.002	0.002
ForceXExcentricityFromNeutralAxis	m	0.047303176	0.04683823
TorsionMoment	Nm	13.76522415	13.62992501
MaxDeltaX	m	4.59E-04	3.86E-04
MaxDeltaY	m	0.000476624	0.000325772
MaxTorsionPhi (1/4 comp to free end torsion beam)	rad	0.000155174	4.02831E-05
MaxTorsionPhi_Degrees	deg	0.008890833	0.002308051
ElevatorCarSpeed	m/s	10	10
K_SupportFactor	-	0.732966387	0.701669901
BeamBucklingForce_Iyy	N	659134.3669	669193.7735
BeamBucklingForce_Ixx	N	590616.9469	779322.5771
MinimumBucklingForce	N	590,617	669,194
BendingAngle_Total_X_Static	rad	1.77E-04	1.80E-04
BendingAngle_Total_Y_Static	rad	0.00018367	0.000151875
BendingAngle_TotalLimit_AAA (divided by 1000 ???)	rad	0.000188888	0.000188888
AnglePerLimit_AAA_X	-	94%	95%
AnglePerLimit_AAA_Y	-	97%	80%

5.5 Additional Results

Useful experience in combining Abaqus and HEEDS software was gained. The optimization problem set up and practical optimization work was learned. The learning process itself can be considered as results.

User defined Fortran subroutines were investigated and used in Abaqus for the moving load definitions. The Fortran script is shown in the appendix.

HEEDS was taken into use as a new software package for optimization. HEEDS usage was learned.

6. Conclusions

Elevator guiderail mass optimization with Abaqus and HEEDS as well as Excel and HEEDS was performed as a case application within a SIMPRO optimization subtask.

An Abaqus dynamic implicit model was supposed to be built for simulating the elevator car as a rigid mass model moving along the two flexible guide rails with geometry imperfections, flexible bracket springs and fishplate springs. The modelling was not successful and therefore several other modelling approaches were investigated and also a simple analytical Excel model was built.

The Abaqus dynamic implicit model consisted of a continuous beam with six bracket supports. Constant loads travelled along the beam and maximum deflections were read as results. This was connected to HEEDS and a multi-objective optimization run was performed. Pareto results were obtained as a balance between maximum beam span achieved and maximum deflection allowed with the five T-profile sections available.

As the model did not include the elevator mass in any way, there was no point in continuing with a dynamic model. A simple static model was built in Excel for calculating buckling load and deflection for a spring supported beam with a load at the mid-span. This enabled running HEEDS with a faster and more robust model. More testing of HEEDS and different optimization algorithms was possible. Also the T-profile dimensions could be freely varied. However, more design constraints were needed to control the T-profile design in terms of size, shape, minimum thickness and slenderness.

Even with the Excel model, the multi-objective optimization with HEEDS provides only Pareto plots as results. This is a curve (with two objectives) or a design space (with multiple objectives) where all designs are optimal and an additional task remains to find or evaluate the best design(s) form this design space. HEEDS does ranking of individual designs, but this is subject to user defined weight factors.

Future work would benefit from the experience gained from this case. More meaningful numbers as results could be achieved with a more tightly defined optimization problem.

7. References

[1] Red Cedar Technologies, Customer Portal, HEEDS MDO 7.1 Documentation, User's Manual. Available at: http://www.redcedartech.com/docs/HEEDSMDO/Welcome.htm. Accessed Jan 9, 2014.

[2] Safety rules for the construction and installation of lifts - Part 1: Electric lifts, European Standard, EN 81-1:1998

[3] KONE internal document.

[4] KONE internal document.

Appendix A

User defined Fortran subroutine used for defining the two moving guide shoe loads

Pasted code text

Code editor screenshots with exact FORTRAN77 syntax and highlighting

```
      SUBROUTINE DLOAD(F,KSTEP,KINC,TIME,NOEL,NPT,LAYER,KSPT,COORDS,
     &                 JLTYP,SNAME)
      INCLUDE 'ABA_PARAM.INC'

      DIMENSION TIME(2), COORDS(3)
      CHARACTER*80 SNAME


C     user coding to define F
      REAL Y_ST,U_CAR,T_DLY,L_ELEM,L_CAR,
     2  F_HI,F_LO,PHI_HI,PHI_LO,W,
     3  T,Y,U


      PI      = 3.141592654
      Y_ST    = 0.0
      U_CAR   = 4.0
      T_DLY   = 1.0
      L_ELEM  = 0.1
      L_CAR   = 4.0

      F_HI    = 0.0
      F_LO    = 0.0
      PHI_HI  = 0.0  *2*PI
      PHI_LO  = 0.0  *2*PI
      W       = 1.0 *U_CAR*2*PI

      F       = 0.0
      T       = 0.0
      Y       = 0.0
      U       = 0.0


C     F = 1.0 * TIME(2) * COORDS(2)

C     F = 5.0 * TIME(2) * MIN( (TIME(2)/0.1)**4, 1.0)

C     F = MAX(2.1, 1.1)

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
CCC
CCCCCCCCCC One force in x-dir, moving in y-dir    CCCCCCCCCC
```

```
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
CCC
C      T = MAX((TIME(2)-1.0), 0.0)
C      Y = COORDS(2)
C
C      F = 1000. * MAX(
C     2 (1.0-(ABS(Y-(Y_ST+U*T)) / L_ELEM))
C     4 , 0.0)


CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
CCC
CCCCCCCCCC Two forces separated by distance L_CAR CCCCCCCCCC
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
CCC
C      T = MAX((TIME(2)-1.0), 0.0)
      T = TIME(2)
      Y = COORDS(2)

      IF (T.LT.T_DLY) THEN
        T=0.0
        U=0.0

        F_HI = (3000.+0.*SIN(W*T+PHI_HI)) * MAX(
     2 (1.0-(ABS(Y-(Y_ST+L_CAR+U*T)) / L_ELEM))
     4 , 0.0)

        F_LO = (1000.+0.*SIN(W*T+PHI_LO)) * MAX(
     2 (1.0-(ABS(Y-(Y_ST+U*T)) / L_ELEM))
     4 , 0.0)

      ELSE
        T=T-T_DLY
        U=U_CAR

        F_HI = (3000.+0.*SIN(W*T+PHI_HI)) * MAX(
     2 (1.0-(ABS(Y-(Y_ST+L_CAR+U*T)) / L_ELEM))
     4 , 0.0)

        F_LO = (1000.+0.*SIN(W*T+PHI_LO)) * MAX(
     2 (1.0-(ABS(Y-(Y_ST+U*T)) / L_ELEM))
     4 , 0.0)
      END IF


C      F_HI = 1000. * MAX(
C     2 (1.0-(ABS(Y-(Y_ST+L_CAR+U*T)) / L_ELEM))
C     4 , 0.0)
C
C      F_LO = -1000. * MAX(
C     2 (1.0-(ABS(Y-(Y_ST+U*T)) / L_ELEM))
C     4 , 0.0)

C      F = MIN(T,1.0)*(F_HI+F_LO)
      F = F_HI + F_LO
```

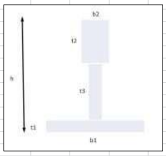
```
C                  IF(NPT.EQ.1) WRITE(6,*) ‘User subroutine message: LOAD APPLIED’,F,’AT
TIME=’,TIME(2)

      IF(NPT.EQ.1) THEN
        WRITE(6,999) 'T=',T,' Y=',Y,' F=',F,' U=',U
999     FORMAT(4(A,E12.4))
      END IF


C      IF(NPT.EQ.1) THEN
C        WRITE(6,999) 'T=',TIME(2),' Y=',COORDS(2),' F=',F
C999     FORMAT(A,E12.4,A,E12.4,A,E12.4)
C      END IF

C      IF(NPT.EQ.1) THEN
C        WRITE(6,'(3(1X,E12.4E3))')
C     & 'F=',F,'TIME=',TIME(2),'COORDS=',COORDS(2)
C     END IF

      RETURN
      END
```

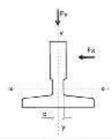
```
      SUBROUTINE DLOAD (F,!<STEP,!<INC,TIHE,NOEL,NPT,LAYER,!<SPT,COORDS,
     &                  JLTYP,SNAHE)
      INCLUDE 'ABA PAruU1.INC'

      DIMENSION TI!1E(2), COORDS(3)
      CBARACTER*SO SNAJ1E




C      user coding to define F
      REAL Y_ST,U_CAR,T_DLY,L_ELEH,L_CAR,
     2  F_HI ,F_LO ,PHI_HI ,PHI_LO ,W ,
     3  T ,Y ,U


      PI        3.141592654
      Y ST      0.0
      UCAR      4 .0
      T DLY     1 .0
      L ELEH     0 .1
      L CAR     4 .0

      F HI      0.0
      FLO       0.0
      PHI HI    0.0   *2*PI
      PHI LO    0.0   *2*PI
      W         1.0  *U CAR*2*PI

      F         0.0
      T         0.0
      y         0.0
      u         0.0
```

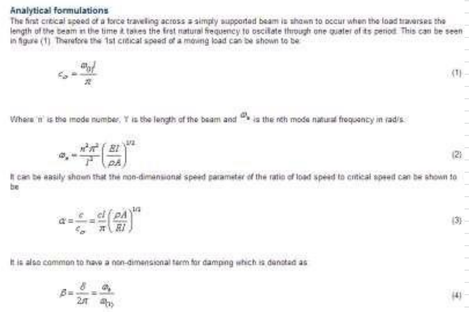
```
C       F   1.0  •TIME (2) •COORDS (2)

C       F   5.0  •TIME (2) •MIN ( (TIME (2)10.1).4,  1.0)

C       F = MAX (2.1, 1.1)

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
CCCCCCCCCC One force in x-dir, moving in y-dir    CCCCCCCCCC
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C       I = MAX ((TIME (2)-1.0), 0.0)
C       Y   COORDS (2)
C
C       F   1000. •MAX (
C      2 (1.0- ABS (Y- (Y_ST+U•T)) / L_ELEM))
C      4 ' 0.0)


CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
CCCCCCCCCC Two forces separated by distance L_CAR CCCCCCCCCC
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C       I = MAX ((TIME (2)-1.0), 0.0)
      I    TII1E(2)
      Y = COORDS(2)

      IF  (T.LT.T_DLY) THEN
        T=0.0
        U=0.0

        F_HI = (3000.+0.*SIN (W*T+PHI_HI)) * MAX (
     2 (1.0- ABS (Y- (Y_ST+L_CAR+U*T)) / L_ELEH))
     4 ' 0.0)

        F_LO = (1000.+0.*SIN (W*T+PHI_LO)) * MAX (
     2 (1.0- ABS (Y- (Y_ST+U*T)) / L_ELEH))
     4 ' 0.0)

      ELSE
        T=T-T DLY
        U=U CAR

        F HI = (3000.+0.*SIN (W*T+PHI_HI)) * MAX (
     2 (1.0- ABS (Y- (Y_ST+L_CAR+U*T)) / L_ELEH))
     4 ' 0.0)

        F_LO = (1000.+0.*SIN (W*T+PHI_LO)) * MAX (
     2 (1.0- ABS (Y- (Y_ST+U*T)) / L_ELEH))
     4 ' 0.0)
      END IF
```

```
C       F_HI = 1000.*MAX(
C      2 (1.0-ABS(Y-(Y_ST+L_CAR+U*T))/L_ELEM))
C      4 * 0.0)
C
C       F_LO = -1000.*MAX(
C      2 (1.0-ABS(Y-(Y_ST+U*T))/L_ELEM))
C      4 * 0.0)

C       F = MIN(T,1.0)*(F_HI+F_LO)
       F = F_HI + F_LO


C   IF(NPT.EQ.1) WRITE(6,*) 'User subroutine message: LOAD APPLIED',F,' AT TIME=',TIME(2)

       IF (NPT.EQ.1) THEN
         WRITE(6,999) 'T=',T,' Y=',Y,' F=',F,' U=',U
         FORMAT(4(A,E12.4))
       END IF


C       IF(NPT.EQ.1) THEN
C         WRITE(6,999) 'T=',TIME(2),' Y=',COORDS(2),' F=',F
C999     FORMAT(A,E12.4,A,E12.4,A,E12.4)
C       END IF

C       IF(NPT.EQ.1) THEN
C         WRITE(6,'(3(1X,E12.4E3))')
C      & 'F=',F,'TIME=',TIME(2),'COORDS=',COORDS(2)
C       END IF

       RETURN
       END
```

	TprofileName	T140-1	-	
2	h	0.108	m	
3	b1	0.14	m	
4	b2	0.01904	m	0.136
5	t1	0.01404	m	0.13
6	t2	0.051678	m	0.55
7	t3	0.0131376	m	0.69
	A	0.003505033	m^2	
	Iyy (for ForceX)	3.26772E-06	m^4	
	Ixx (for ForceY)	4.04123E-06	m^4	
	NeutralAxisFromBottomFlange	0.032946362	m	
	Iv_torsion_Estimate	3.13615E-07	m^4	
	E	2E+11	Pa	
	Poisson	0.3	-	
	G	76923076923	Pa	
	rhoSteel	7850	kg/m^3	
	I_polar (Iyy+Ixx)	7.30896E-06	m^4	
	Beamlength	2.97	m	
	LineMass	27.51451002	kg/m	
	BeamTotalMass	81.71809475	kg	
	BracketStiffnessTranslation	1.00E+09	N/m	
	BracketStiffnessRotation	1.00E+06	Nm/rad	
	BendingForceX	291	N	Fx61=291 N MathcadGuide_Paee_38
	BendingForceY	277	N	Fy62=277 N MathcadGuidei_Page_40
	ForceXDistanceFromTProfileTip	0.002	m	
	ForceXExcentricityFromNeutralAxis	0.073053638	m	
	TorsionMoment	21.25860861	Nm	
	MaxDeltaX	1.16E-04	m	
	MaxDeltaY	9.62077E-05	m	
	MaxTorsionPhi (1/4 comp to free end torsion beam)	0.000654302	rad	
	MaxTorsionPhi_Degrees	0.03748873	deg	
	ElevatorCarSpeed	10	m/s	
	GuideRollerWheelRadius	0.15	m/s	
	DynamicForceXAmplitude_Travel	291	N	
	DynamicForceYAmplitude_Travel	277	N	
	DynamicTorsionMomentAmplitude_Travel	21.25860861	Nm	
	DynamicForceFrequency_Travel CHECK THIS!!! 2 or 4 beam lengths???	0.841750842	Hz	Speed/(2 OR 4*Lbeam)
	DampingX_Travel	0.01	-	
	DampingY_Travel	0.01	-	
	DampingTorsion_Travel	0.01	-	
	BeamNaturalFrequencyX (Pin-Pin-Bending)	27.44503706	Hz	
	BeamNaturalFrequencyY (Pin-Pin-Bending)	30.5209435	Hz	
	BeamNaturalFrequencyTorsion (Clamp-Clamp-Torsion)	1843.875423	Hz	check
	FrequencyRatioX_Travel	0.030670421	-	
	FrequencyRatioY_Travel	0.02757945	-	
	FrequencyRatioTorsion_Travel	0.000456512	-	
	DAF_X_Travel	1.000941372	-	
	DAF_Y_Travel	1.000761053	-	
	DAF_Torsion_Travel	1.000000208	-	
	StaticMaxDeltaX_Travel	1.16E-04	m	
	StaticMaxDeltaY_Travel	9.62077E-05	m	
	StaticMaxTorsionPhi_Travel	0.000654302	rad	
	DynamicMaxDeltaX_Travel	0.000116566	m	
	DynamicMaxDeltaY_Travel	9.6281E-05	m	
	DynamicMaxTorsionPhi_Travel	0.000654302	rad	
	DynamicForceAmplitudeX_Roller	0	N	NoWheelExitation
	DynamicForceAmplitudeY_Roller	0	N	NoWheelExitation
	DynamicForceAmplitudeTorsion_Roller	0	Nm	
	DynamicForceFrequency_Roller	10.61032954	Hz	
	DampingX_Roller	0.01	-	
	DampingY_Roller	0.01	-	
	DampingTorsion_Roller	0.01	-	
	FrequencyRatioX_Roller	0.38660283	-	
	FrequencyRatioY_Roller	0.347640942	-	
	FrequencyRatioTorsion_Roller	0.005754304	-	
	DAF_X_Roller	1.175677516	-	
	DAF_Y_Roller	1.137432217	-	
	DAF_Torsion_Roller	1.000033107	-	
	StaticMaxDeltaX_Roller	0		
	StaticMaxDeltaY_Roller	0		
	StaticMaxTorsionPhi_Roller	0		
	DynamicMaxDeltaX_Roller	0	m	
	DynamicMaxDeltaY_Roller	0	m	
	DynamicMaxTorsionPhi_Roller	0	rad	
	StaticMaxDeltaX_Total	0.000116457	m	
	StaticMaxDeltaY_Total	9.62077E-05	m	
	StaticMaxTorsionPhi_Total	0.000654302	rad	
	DynamicMaxDeltaX_Total	0.000116566	m	
	DynamicMaxDeltaY_Total	9.6281E-05	m	
	DynamicMaxTorsionPhi_Total	0.000654302	rad	
	DAF_X_Total	1.000941372	-	
	DAF_Y_Total	1.000761053	-	
	DAF_Trosion_Total	1.000000208	-	
	K_SupportFactor	0.569507839	-	
	BeamBucklingForce_Iyy	2254565.74	N	
	BeamBucklingForce_Ixx	2788246.527	N	
	MinimumBucklingForce	2,254,566	N	
	RideComfortAngle_Total_X_Static	5.23E-05	rad	
	RideComfortAngle_Total_Y_Static	4.31909E-05	rad	
	RideComfortAngle_TotalLimit_AAA (divided by 1000)	0.000188888	rad	
	AnglePerLimit_AAA_X	28%	-	
	AnglePerLimit_AAA_Y	23%	-	






Analytical formulations

The first critical speed of a force travelling across a simply supported beam is shown to occur when the load traverses the length of the beam in the time it takes the first natural frequency to oscillate through one quater of its period. This can be seen in figure (1). Therefore the 1st critical speed of a moving load can be shown to be

$$c_{cr} = \frac{\omega_{01} l}{\pi} \quad (1)$$

Where 'n' is the mode number, 'l' is the length of the beam and ω_n is the nth mode natural frequency in rad/s.

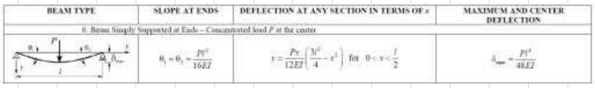
$$\omega_n = \frac{n^2 \pi^2}{l^2} \left(\frac{EI}{\rho A} \right)^{1/2} \quad (2)$$

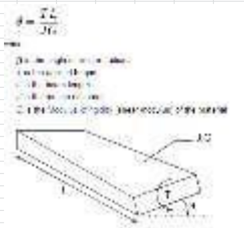
It can be easily shown that the non-dimensional speed parameter of the ratio of load speed to critical speed can be shown to be

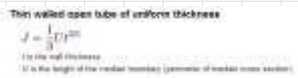
$$\alpha = \frac{c}{c_{cr}} = \frac{cl}{\pi} \left(\frac{\rho A}{EI} \right)^{1/2} \quad (3)$$

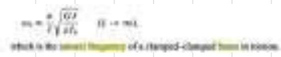
It is also common to have a non-dimensional term for damping which is denoted as

$$\beta = \frac{\delta}{2\pi} = \frac{\omega_b}{\omega_{(1)}} \quad (4)$$

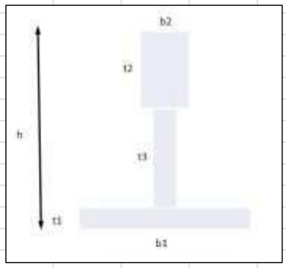


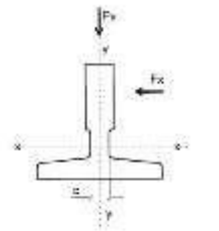


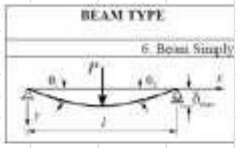




TprofileName	T140-1	-	
h	0.108	m	
b1	0.14	m	
b2	0.01904	m	0.136
t1	0.01404	m	0.13
t2	0.051678	m	0.55
t3	0.0131376	m	0.69
A	0.003505033	m^2	
Iyy (for ForceX)	3.26772E-06	m^4	
Ixx (for ForceY)	4.04123E-06	m^4	
NeutralAxisFromBottomFlange	0.032946362	m	
Iv_torsion_Estimate	3.13615E-07	m^4	
E	2E+11	Pa	
Poisson	0.3	-	
G	76923076923	Pa	
rhoSteel	7850	kg/m^3	
I_polar (Iyy+Ixx)	7.30896E-06	m^4	
Beamlength	2.97	m	
LineMass	27.51451002	kg/m	
BeamTotalMass	81.71809475	kg	
BracketStiffnessTranslation	1.00E+09	N/m	
BracketStiffnessRotation	1.00E+06	Nm/rad	
BendingForceX	291	N	Fx61=291 N MathcadGuide_Page_38
BendingForceY	277	N	Fy62=277 N MathcadGuide_Page_40
ForceXDistanceFromTProfileTip	0.002	m	
ForceXExcentricityFromNeutralAxis	0.073053638	m	
TorsionMoment	21.25860861	Nm	
MaxDeltaX	1.16E-04	m	
MaxDeltaY	9.62077E-05	m	
MaxTorsionPhi (1/4 comp to free end torsion beam)	0.000654302	rad	
MaxTorsionPhi_Degrees	0.03748873	deg	
ElevatorCarSpeed	10	m/s	
GuideRollerWheelRadius	0.15	m/s	







K_SupportFactor	0.569507839	-
BeamBucklingForce_Iyy	2254565.74	N
BeamBucklingForce_Ixx	2788246.527	N
MinimumBucklingForce	2,254,566	N
BendingAngle_Total_X_Static	5.23E-05	rad
BendingAngle_Total_Y_Static	4.31909E-05	rad
BendingAngle_TotalLimit_AAA (divided by 1000 ???)	0.000188888	rad
AnglePerLimit_AAA_X	28%	-
AnglePerLimit_AAA_Y	23%	-