Abstract
This master thesis examines a sandwich lift plate for cargo trucks in order to propose an option for existing lift platforms in aluminium or steel. The wish is to make it lighter but not less stiff or weak. Two different cores in Divinycell with different density; H130 and H250 is examined. Calculations are made with respect to deflection and max stress in both core and faces for the whole plates. But also more local calculations are performed to see how the faces and core behaved on local point loads from pallet lift wheels.

The analytical results successfully meet the deflection requirements causing only 30 mm deflection in the worst scenario with 33% overload on a lift plate built up by 3 mm aluminium faces and H130 Divinycell core measuring a total (incl. faces) 20 mm thickness in the top. Nor did the von Mises stresses exceed 50% of ultimate strength for the aluminium parts or the Divinycell core. The choice between the lighter (H130) and heavier (H250) core only appears to affect the deflection 1-2 mm and the von Mises face stress for point loads are in the order of 10 MPa.

The objective regarding the mass of the lift plate was not met. This is however a matter of further optimisation and is considered to be solved.
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## Nomenclature

<table>
<thead>
<tr>
<th>Factor</th>
<th>Definition</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$t_f$</td>
<td>Thickness of upper sandwich face</td>
<td>mm</td>
</tr>
<tr>
<td>$t_{f2}$</td>
<td>Thickness of lower sandwich face</td>
<td>mm</td>
</tr>
<tr>
<td>$t_f$</td>
<td>Same thickness of upper and lower face</td>
<td>mm</td>
</tr>
<tr>
<td>$t_c$</td>
<td>Thickness of core</td>
<td>mm</td>
</tr>
<tr>
<td>$d$</td>
<td>$t_{f1}/2 + t_{f2}/2 + t_c$</td>
<td>mm</td>
</tr>
<tr>
<td>$P$</td>
<td>Force</td>
<td>N</td>
</tr>
<tr>
<td>$L$</td>
<td>Length</td>
<td>mm</td>
</tr>
<tr>
<td>$R_{p0.2}$</td>
<td>Yield strength</td>
<td>MPa</td>
</tr>
<tr>
<td>$R_m$</td>
<td>Ultimate strength</td>
<td>MPa</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
<td>kg/mm$^3$</td>
</tr>
<tr>
<td>$E_f$</td>
<td>Young’s modulus of the Face</td>
<td>MPa</td>
</tr>
<tr>
<td>$E_c$</td>
<td>Tensile modulus of the Core</td>
<td>MPa</td>
</tr>
<tr>
<td>$G_c$</td>
<td>Shear modulus</td>
<td>MPa</td>
</tr>
<tr>
<td>$\nu$</td>
<td>Poisson’s ratio</td>
<td>-</td>
</tr>
</tbody>
</table>

1. **Introduction**

Tail lifts have many times an unfriendly situation in the very aft of the truck. When it’s raised; dust and water are whirling in the wake and accumulates on the underside and when it’s lowered; stones, dirt and wheels from pallet lifts will wear its faces and also cause point forces.

In order to meet the conditions of today’s demands of lighter and more effective transports will this master thesis contribute with a competitive alternative to the standard tail lifts for smaller cargo trucks and vans. Existing tail lifts are made in steel or aluminium and a profit in weight may be considerable with a sandwich construction.

A sandwich design is also the only design for tail lifts that offers isolation for refrigerated cargo compartments.

1.1. **Background**

Zepro was pioneers in the field of tail lift platforms and has produced tail lifts for decades in a variety of design, sizes and fastening set ups. One of the most successful models is the Z-75 which is designed for smaller trucks and vans, it carries a maximum load of 750 kg.

The original lift plate on model Z-75 is an all-aluminium construction built up by three types of string casted profiles which gives the benefit to easy adjust the length between different
models. The profiles are then welded together. (This is already a relatively light and very
durable design that doesn’t show any structural weaknesses after several year of daily use.

Z-75 is available in different sizes. Three examples for a standard 30 mm thick aluminium
platform; 1200 x 2200 mm has a mass of 58 kg, 1450 x 2200 mm has a mass of 67 kg and
1600 x 2200 mm has a mass of 72 kg. The objective in this work is to match the second plate
in dimensions and mass. Mass and price is two important aspects to compete with.

![Figure 1. The original Z-75 tail lift with lift plate manufactured in string casted aluminium profiles](image)

An earlier manufactured test sandwich lift plate with uniform thickness was made in 2002.
That example did not meet the qualifications, it did carry the loads without failing but with a
considerable trampoline effect and the deflection of 95 mm in the physical static and fatigue
testing was too much. Since fibreglass served as face material in this study some of the
flexibility may be ascribed to this choice.
1.2. Aims and scope
The goal is to provide a light lift plate without any restraints on the stiffness. The approach to succeed with this objective is to build up a platform with the use of a light core between two faces, i.e. a sandwich design.

2. Design, sandwich lift plate concept
The new design of the lift plate is a wedge shaped sandwich design with an top and bottom face with a core in between. The formula of the Divinycell has been improved since the test plate in 2002 and a choice of harder Divinycell will be chosen this time.

This sandwich plate has all sides of the core covered and a lengthwise tapering. It means that the thickest part is closest to the cargo compartment while it has a constant taper towards the tip. The attachment brackets which connect the lift plate to the manoeuvre arms and cylinders have the same design as the original Zepro Z-75. The overall design can be seen in Figure 2.

![Figure 2. Schematic view of the sandwich lift plate](image)

Since it’s beneficial to spread the shear stresses in the core over the biggest possible cross section between the faces, the idea is to use the whole cross section area down to the attachment brackets. This directly leads to the advantage of a totally flat underside, i.e. useful commercial area and less area for dirt to accumulate on.

The dimension of the lift plate is the same as original Z-75. The most outer tip is measuring 20 mm thick and at the bottom part at the attachment brackets close to the cargo compartment is measuring 157 mm. The lift area has the dimensions 1500 x 2200 mm (length x width).

Four imbedded lift arms transfer the loads from the plate to the attachment brackets in order to make the design stiffer These lift arms has the same thickness as today’s all aluminium Z-75 lift plate of 5 mm with the difference that they extended following the taper of the lower
The new design consists of nineteen parts listed in the table in Figure 3. The illustration unveil the inside structure of the sandwich lift plate. Four imbedded lift arms transfer the loads to the attachment brackets and will add strength to the design. Nomenclature of the lift plate’s orientation is also illustrated in Figure 1.

<table>
<thead>
<tr>
<th></th>
<th>Description</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Core mid section, 1 piece</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Lift arm, 4 pieces</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Core between lift arms, 2 pieces</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Core outer part, 2 pieces</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Tip list, 1 piece</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Bottom contact list, 1 piece</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Top plate, 1 piece</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Bottom plate, 1 piece</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Corner list, 1 piece</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>Front plate, outer, 2 pieces</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>Attachment bracket, 2 pieces</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>Front plate mid, 1 piece</td>
<td></td>
</tr>
</tbody>
</table>

Figure 3. Principal built up of the sandwich lift plate with table of parts and definitions of orientations.

3. Requirements
To fulfil the requests for a stiff and reliable lift plate, the objectives can be summarized:

- Applied loads in the analysis shall be maximum allowed loads on the lift plate multiplied by a factor of 1.33. Zepro normally use a test factor of 2.1 in the physical test but this test is performed with the load applied on a Euro pallet and since the load cases in this analysis will be node loads or line loads, i.e. much more conservative, 1.33 is a estimated mean factor.

- Maximum deflection for lift plate shall not exceed 50 mm.

- The von Mises stress shall not exceed the Ultimate strength (Rm) for the faces or other metal parts in the structure.
- The maximum negative principal face stress in sandwich panel shall not exceed half of calculated limit stress for Local buckling. This implies a safety margin of 2 for local buckling.

- One objective is to keep the mass of the lift plate low. The goal is to match the mass of the existing aluminium Z-75 platform.

- Material cost will be examined.

4. Materials

Due to the relative unfriendly environment in the aft of the truck, the exposed parts of the platform have to be insensitive for wet or absorbing, accumulate and transport wet into the core. A precondition is that the joints and gaps in the design are sealed.

4.1. Face material

The main task for the face material is to take care of the compression and tension forces in the faces; consequently the choice of face material has to have high tensile properties. The examined face material are; fibreglass, steel and aluminium.

The fibreglass option is by far the lightest considering tensile strength weighted mass, but earlier delaminating problems and the flexible characteristics leaves room for alternative face materials.

Steel is by far the stiffest but this pays off in higher mass and corrosion problems.

Aluminium is a balanced choice to meet the requirements in terms of mass, stiffness, corrosion and price. Based on these benefits and in the light of the less positive sides of steel and fibreglass the main face material in this examination will be aluminium. Some space will anyhow be given to sandwich analysis with faces of steel and fibreglass, to be compared with.

4.2. Core material

The main task for the core is to take care of the shear forces caused by the lateral travelling of the faces due to the bending of the plate. A material with high shear modulus is consequently to prefer. Well known materials for this task are balsa wood, honeycomb made in different materials, and different types of polymer foams. One well-known brand of polymer foam is Divinycell which has a wide variety of different foam cores for many applications. In this examination two core materials from the Divinycell High-Performance series will be used, H130 and H250. This is a polymer foam product with closed cells from Diab where 130 and 250 is the density in kg/m$^3$. 
4.3. Material data
The chosen materials are selected with information of commonly used materials given from Zepro and Diab. Aluminium AlmgSi0.7 F27 is used in the string casted sections that build up the original Z-75 lift. Steel S375 is used in the main structure for larger tail lifts entirely designed out of steel. Data for each material can be seen in Table 1.

<table>
<thead>
<tr>
<th>Material</th>
<th>Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Divinycell H130</td>
<td>Young’s modulus $E_{C,H130}$ 175 MPa</td>
</tr>
<tr>
<td></td>
<td>Shear modulus $G_{C,H130}$ 50 MPa</td>
</tr>
<tr>
<td></td>
<td>Density 130 kg/m$^3$</td>
</tr>
<tr>
<td>Divinycell H250</td>
<td>Young’s modulus $E_{C,H250}$ 320 MPa</td>
</tr>
<tr>
<td></td>
<td>Shear modulus $G_{C,H250}$ 104 MPa</td>
</tr>
<tr>
<td></td>
<td>Density 250 kg/m$^3$</td>
</tr>
<tr>
<td>Fibreglass</td>
<td>Young’s modulus $E_{Fibreglass}$ 16 GPa</td>
</tr>
<tr>
<td></td>
<td>Poisson’s ratio 0.3 aprx</td>
</tr>
<tr>
<td></td>
<td>Ultimate strength $R_m$ 230 MPa</td>
</tr>
<tr>
<td></td>
<td>Density 1700 kg/m$^3$</td>
</tr>
<tr>
<td>Steel S375</td>
<td>Young’s 210 GPa</td>
</tr>
<tr>
<td></td>
<td>Poisson’s ratio 0.3</td>
</tr>
<tr>
<td></td>
<td>Yield strength $R_{p,0.2}$ 375 MPa</td>
</tr>
<tr>
<td></td>
<td>Ultimate strength $R_m$ 510 MPa</td>
</tr>
<tr>
<td></td>
<td>Density 7700 kg/m$^3$</td>
</tr>
<tr>
<td>Aluminum AlmgSi0.7 F27(6005A T6):</td>
<td>Young’s modulus $E_{6005A,T6}$ 68.9 GPa</td>
</tr>
<tr>
<td></td>
<td>Poisson’s ratio 0.33</td>
</tr>
<tr>
<td></td>
<td>Limit $R_{p,0.2}$ 225 MPa</td>
</tr>
<tr>
<td></td>
<td>Ultimate strength $R_m$ 275 MPa</td>
</tr>
<tr>
<td></td>
<td>Density 2700 kg/m$^3$</td>
</tr>
</tbody>
</table>

*Table 1. Properties for used materials*
5. Methods: Finite Element Analysis (FEA) and analytical approach

Due to the multipart design of the construction, Finite Element Analysis (FEA) is a good choice because of the complexity in the design and the easy ability to change conditions, such as material, loads and geometries. Analytical calculations will be used to verify the accuracy of the FEA model and to examine buckling behavior of the faces.

5.1. Analysis program

To fulfill prescribed requirements of the sandwich plate, a mix of analytical methods and FEA will be consulted.

**Patch test, FEA and analytical calculation**

To confirm that the FE-model is reliable, a patch test has to be performed in order to verify that the used elements in the FE-model give a reasonable result. This will be done using a standard square shapes cross sectioned sandwich cantilever and compare the FEA answers with analytical calculations.

**Sandwich plate deflection, FEA**

This is one of the main parts in the report and the questions of deflection of the plate and stresses in face and core will be answered. The entire sandwich plate with all essential bearing details will be subjected to four different load sets using FEA, chosen FEA-tool is Ansys 10 Multiphysics Chosen elements in the analysis will be the ones evaluated from previous Patch test.

**Lift arm analysis, FEA**

Moreover an analytical method will be consulted to investigate the deflection in one of the imbedded lift arms. Elementary cantilever calculations calculation is to be compared with Ansys results considering deflection and stress.

**Local face pressure analysis, FEA**

In addition, a point load model is also of interest to see how the top face stands the point forces caused by for example the wheels on pallet lifts. The vertical deflection and the stresses transferred down into the core is and how large deformations that transfers down into the core.

**Local buckling, analytical calculation**

The phenomenon Local buckling, which is a structural instability problem and sometimes critical to sandwich designs will be explained and the sandwich lift plate will be examined to be safe from this occurrence.

5.2. General aspects of FEA

FE-analysis is to break down a physical structure to small substructures called *Finite elements*. FE-analysis can mathematically be described as finding approximate solutions to partial differential equations to solve structural or elastic problems, or even thermodynamic, electrical or viscous problems.

Depending on the nature of the structural problem, the design can be modelled either with beams, shells or solids. Beam modelling may be referred to as one dimensional, suitable for frameworks and is not interesting in this case but shells and solids may represent the
sandwich problem in a good way. Shells and solids can be modelled in different geometrical shapes, -triangles, rectangles, tetrahedrals, bricks etc. All these elements are built up with nodes, -in the corners, or also in between the corners. The nodes will describe the boundary and the displacement of the elements and it’s also in the nodes where the elements continuum equation is solved. All elements are consequently forming a mesh within (the solid) and over the entire design, the size of each element is a result how fine the mesh is. For ordinary hyperelastic elements (H-elements), the resolution of the mesh is also a direct tool to control the convergence and accuracy of the solution. A coarse mesh may not pick up displacements in a proper way.

5.2.1. FEA, design

To model and examine a structural design in a FE program some simplifications can often be done since small details are not always necessary for the outcome of the calculations. Advanced models are also very time- and capacity consuming. Removing material and details will often also give a more conservative solution since more parts has a stiffening effect on the design. The rule is to keep the FE representation simple without obtaining a non-conservative result. Care must be taken when geometrical simplifications are done around round corners or holes since sharp corners may cause singularities and stress peaks. When the design is satisfying the model needs a mesh and element description.

5.2.2. FEA, meshing

Meshing is a central, important and sometimes a hard part in FE modelling. The more advanced the design is the harder and more enforcing it will be to create a good mesh. Also, the simpler the shape is, the more straightforward elements can be used. The most straightforward 3D element is a brick and the big advantage with bricks compared with tetrahedrals or pyramids is that the form equations solved within each element is becoming almost theoretical exact due to the bricks’ simple shape. Nevertheless the aspect ratios of the sides on the brick cannot become too large or it will lead to a bad conditioned model with violent shapes when deformed and that will cause very unreliable results.

5.2.3. FEA, element description

Many FE programs offer a wide range of different elements, different geometrical shapes and thus different node set ups.

One dimensional (1D) beam elements may be used to represent frameworks with beam characteristics that describe the geometrical cross section of the beam. One dimensional element will not be further used in this work.

Two dimensional (2D) squared shaped shell elements can be represented with four (corner nodes) or eight (corner nodes + mid) nodes elements. The nodes in a shell element may be able to transfer translational degrees of freedom (membrane element), or both translational and rotational degrees of freedom.

In the same way as the two dimensional elements, three dimensional (3D) brick elements can be represented with eight (corner nodes) or twenty (corner nodes + mid) nodes elements. The advantage with more nodes in each element is to catch more complex geometries. Naturally, the drawback of this is increased computational time. Solids will normally not be able to transfer rotational degrees of freedom, thus the bending capability has to be threatened through layers of brick elements.
The standard type of elements are called hyperelastic elements (H-elements), the form function solved within each element is linear. In order to catch more complex designs without the demand of a fine mesh, it is also possible to reach convergence through a polynomial form function where the polynomial order is automatically increased until convergence is reached. These elements are called P-elements and they can catch a complex design even with a coarse mesh and that advantage is payed of by increased computational time.

These presented elements leave a variety of options to build up a sandwich design with; solids, a combination of solids and shells or even P-elements.

6. **Patch test with standard cantilever**

This section will verify which FE-elements that can be used, alone or in combination with each other. The validation is performed on a sandwich cantilever.

6.1. **Analytical approach for standard cantilever**

The sandwich cantilever has boundary conditions similar to the lift plate, load is per unit width. The cantilever with load can be seen in *Figure 4*.

![Figure 4. Rigid attached and free end sandwich cantilever with line load at the tip](image)

with dimensions

\[
\begin{align*}
    t_{f_1} &= 3 \text{ mm} \\
    t_{f_2} &= t_{f_1} \\
    t_{c} &= 40 \text{ mm} \\
    d &= \frac{t_{f_1}}{2} + \frac{t_{f_2}}{2} + t_{c} \\
    L &= 1500 \text{ mm} \\
    P &= 2.2 \text{ N/mm}.
\end{align*}
\]

All other properties are according to material set up with aluminium and Divinycell H130.

When calculating sandwich structures it is obligatory to calculate the contributions from both bending and shearing of the structure. Practically this means that the major part of the bending stiffness is represented by the faces and the major part of the shear stiffness is represented by the core. This implies it is necessary to calculate the shear stiffness [1] with
Using the shear stiffness $S = \frac{Gd^2}{t_c}$ (1) and the bending stiffness $D = \frac{Et^2}{2}$ (2) with

$$D = \frac{E_t d^2}{2}.$$ 

The shear stiffness and bending stiffness can then be used to calculate the deflection

$$S = \frac{P \bar{E}}{D} + \frac{P L}{3D S}$$

which is, just like the two previous expressions, for a two dimensional case, i.e. per unit width. Hence width can be excluded in further calculations. Actual data can then be used to calculate the deflection for the cantilever to

A limitation for ordinary engineering with sandwich constructions is the core – face ratio, which should be larger than 4.77 ($t_c/t_f > 4.77$) [2]. This gives a minimum thickness of the face to ~2.5 mm for 12 mm core at the tip of the plate.

### 6.2. FE-analysis for standard cantilever

The analytical result is now to be compared with the outputs from four different elements set ups. In this FE-modulation a three dimensional model is likely to represent the real lift plate in a good way since the imbedded lift arms and the asymmetric load are not easy to describe in different ways.

Three different solid elements and two shell elements are to be examined for further analysis. In the first three element set ups, the core is represented by conventional H-elements, eight and twenty node brick elements. The sandwich faces are represented with four and eight node shell elements but also solid elements.

Schematically built up with load can be seen in **Figure 5**.

*Figure 5. Patch test cantilever with loads in nodes at the right side and constraints in the other end. Core element size is 18.75 x 16.66 x 5.375 mm.*
The four element set ups are:
Case 1: Solid 45 and Shell 63 8 node brick element and 4 node shell element
Case 2: Solid 45 8 node brick element
Case 3: Solid 186 and Shell 99 20 node brick element and 8 node shell element
Case 4: Solid 147 20 node brick polynomial element

All elements are compared with the same element layup and the same number of elements in the core (case 2 and 4 has an addition of 240 x 4 for solid faces).

**Case 1** is a representation with solid elements and shell elements. The bending stiffness of the faces will be represented by the moment capability in the shell nodes.

**Case 2** is very much like the previous but is entirely modelled with solid elements. The bending stiffness will be represented by translational degrees of freedom in the brick elements. This will give a hint if a face modelled with one single layer of elements will affect the output in a major extent. The faces are built up with one layers of solid elements (one on each side).

**Case 3** is a representation with solid elements and shell elements but, with mid-nodes (on both solids and shells). This shell element also has the advantageous to define a lay-up sequence if a fibre composite shall be represented. Hence this may be a little heavy and unnecessary when an isotropic face is to be represented and will consequently increase the solution time.

**Case 4** is a representation with P-elements. The main benefit with this model is the usage of a coarse mesh for the faces, which may represent the faces with only one layer of solid elements in a much better way than comparable representation with standard H-element.

### 6.3. Result. FE-model for standard cantilever

The deflections from the four different load set ups are compiled in *Table 2.*

<table>
<thead>
<tr>
<th>Element type</th>
<th>Defl. [mm]</th>
<th>CP [sec]</th>
<th>Nmbr. of elemts</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solid 45 and Shell 63</td>
<td>14.15</td>
<td>14.87</td>
<td>4800</td>
</tr>
<tr>
<td>Solid 45</td>
<td>14.50</td>
<td>16.18</td>
<td>5760</td>
</tr>
<tr>
<td>Solid 186 and Shell 99</td>
<td>14.07</td>
<td>95.61</td>
<td>4800</td>
</tr>
<tr>
<td>Solid 147</td>
<td>14.07</td>
<td>460.59</td>
<td>5760</td>
</tr>
</tbody>
</table>

*Table 2. Deflection of cantilever according to different element combinations in FE runs*

### 6.4. Discussion FE-analysis and standard cantilever

Table 2 shows that all element combination matches the analytical answer very good and difference in solution time can be seen in the *CP* column. This means that the third and fourth option is fairly computer demanding and any of the first two element combination will be good choices but the first will be even less time consuming.

The choice for further FE-modelling will be based on this study and consequently the chosen set up will fall on the combination of Solid45 and Shell63.
6.5. **Element presentation: SOLID45**

This is a 3D brick element. The element is defined by 8 nodes having three degrees of freedom per node: translations in the nodal x, y, and z directions. The element may have any spatial orientation. Solid45 is used for the three-dimensional modelling of solid structures. The element is defined by eight nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions, see figure 4 for details.

This element has plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities. With Solid45 it is also possible to model prisms and tetrahedrals but this task will only use brick (or cub) element because the form equations solved within each element will be solved in the most accurate way with this element due to the cub’s plain and simple shape. *Figure 5* shows the characteristics. The nodes have orthotropic material properties. Orthotropic material directions correspond to the element coordinate directions.

![Element Coordinate System (shown for \eta=1)](image)

*Figure 6. Three element designs that can be modelled using SOLID45 with all nodes placed in the corners.*

6.6. **Element presentation: SHELL 63**

Shell63 has both bending and membrane capabilities. Both in-plane and normal loads are permitted. The element has six degrees of freedom at each node: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z axes. Stress stiffening and large deflection capabilities are included. A consistent tangent stiffness matrix option is available for use in large deflection (finite rotation) analyses.

![Element Coordinate System](image)

*Figure 7. SHELL 63*
7. Lift plate model, presentation FE-analysis

The lift plate described in §1.2. Design, sandwich lift plate and seen in Figure 2 is to be modelled and analysed.

To facilitate the FE analysis simplifications can be made to the model. The most outer tip strip is intended to go in between the two faces to increase the strength and stiffness of the tip of the plate. This will only have a small theoretical influence on the sandwich concept and the addition of this strip will increase the design’s stiffness, illustrated in Figure 9 for details.

![Figure 9. Detailed view of the simplifications made to the design. Left side shows attachment bracket and right side shows the tip strip.]

The curved contact strip towards the cargo compartment (shown at upper left part in Figure 9.) has a stiffening effect on the plate. This detail can be excluded from further FE-analysis since it also will be a conservative representation.

The two attachment brackets that are connected to the lever arms and the maneuver cylinders (also shown at the upper left part of Figure 9.) are of the same sizes as the original Z-75 and there is no reason to look any closer on these since they have not shown any signs of structural weaknesses or fatigue.

Moreover this FE-model is designed without side walls protecting the core from side impacts, this is anyhow a conservative way of modelling and a later add of these will have a stiffening effect even if this analyse now will be made without them.

Four load cases will be modelled in order to calculate the required face thickness. All four load cases are described with line forces acting on the top surface at four different positions. Applied load is the maximum load of 750 kg (according to the data sheet for Z-75) multiplied by the safety margin of 1.3, which gives a load of 10 kN (i.e. ~1000 kg) along element lines. The only constraints in all cases are the four attachment points at the attachment brackets, these are locked in all directions; translation in ux, uy,uz and rotations about the x-, z-, y-axis.

To give an idea of the size of the mesh, the meshed FE model can be seen in Figure 10.
Figure 10. FE model with mesh. This picture shows a finer mesh at the left side since this very capture is dedicated to the left side calculations of the lift plate.

The almost endless combinations of different upper and lower face thickness (tf) and materials can be simplified. Due to the linear elastic element representation Ansys will deliver the same deflection for the same asymmetric representation of face thicknesses (1 mm upper face and 3 mm lower face will give the same result as the opposite face representation). This may not be the real case since the materials in the core and the faces are not behaving identical when subjected to tension or compression. Nor does the FE-model represent nonlinear support effects from the core (or the glue bonding) or local buckling phenomenon of the compression face. The local buckling phenomenon on the design may be critical and will be examined more in detail in § 8. Local Buckling.

(The choice of chosen face material will mainly fall on aluminium (based on discussion in sect) in this analysis. Besides six runs with simultaneously increasing aluminium (both upper and lower) face thickness in steps of 0.5 mm, the design will also be compared with designs with fibreglass or steel faces.)

7.1. Material set ups for FE-analysis
To investigate if the heavier H250 core is much more favourable before the lighter H130 core, both combinations will be examined with aluminium faces and simultaneously increasing face thicknesses for top and bottom face (tf) in steps of 0.5 mm. This will be compared to single examples for steel and fibreglass, the heavier Divinycell H250 core is chosen in combination with steel faces since two stiff materials collaborates better, rather than a combination with a soft and a hard one. In the same way is the lighter Divinycell H130 chosen with the weaker fibreglass combination.

All four load cases will follow the test schedule seen in Table 3.

<table>
<thead>
<tr>
<th>Set 1</th>
<th>Material combination</th>
<th>Alu/H130</th>
<th>Alu/H130</th>
<th>Alu/H130</th>
<th>Alu/H130</th>
<th>Alu/H130</th>
<th>Fibreglass/H130</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>tf [mm]</td>
<td>2</td>
<td>2.5</td>
<td>3</td>
<td>3.5</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Set 2</td>
<td>Material combination</td>
<td>Alu/H250</td>
<td>Alu/H250</td>
<td>Alu/H250</td>
<td>Alu/H250</td>
<td>Alu/H250</td>
<td>Steel/H250</td>
</tr>
<tr>
<td></td>
<td>tf [mm]</td>
<td>2</td>
<td>2.5</td>
<td>3</td>
<td>3.5</td>
<td>4</td>
<td>2</td>
</tr>
</tbody>
</table>

Table 3. Test schedule and material combinations for the four load cases, forty-eight different runs all in all.

7.2. Presentation Load case 1
First load case describes the 10 kN load or 10.5 kN/m line load acting on 70% length from the front of the lift plate and this is the most outer location where loads shall be positioned according to the Z-75 data sheet. Figure 11 visualize Load case 1 with constraints and load.
Figure 11. Load case one, line load at 70%, from the front blue areas are constraints in the lift arms, triangles in figure are the constraints in the lift arms, locked in translation $u_x$, $u_y$ and $u_z$ together with rotations about the $x$-, $y$- and $z$-axes. Drawing is not accurate according to scale.

7.3. Presentation Load case 2
The second load case has the same 10 kN load or 13.9 kN/m line load and is still acting on 70% length from the front but this time it is placed on one of the sides. None of the three following load cases are allowed according to the Z-75 data sheets. The effect of treatment like this is anyhow of interest and tells more about the characteristics of the plate. Figure 12 visualizes load case 2 with constraints and load.

Figure 12. Load case 2. 13.9kN/m line load at 70% from front on outer part, triangles in figure are the constraints in the lift arms, locked in translation $u_x$, $u_y$ and $u_z$ together with rotations about the $x$-, $y$- and $z$-axes.

7.4. Presentation Load case 3
The third load case has the 10 kN load or 10.5 kN/m line load applied on the central rear of the plate just inside the stiffening strip at the very rear of the lift plate. Figure 13 visualize load case 3 with constraints and load.

Figure 13. Load case 3 Line load of 10.5 kN/m at the mid rear tip, triangles in figure are constraints in the lift arms, locked in translation $u_x$, $u_y$ and $u_z$ together with rotations about the $x$-, $y$- and $z$-axes. Drawing is not accurate according to scale.
7.5. Presentation Load case 4
Load case four is the worst scenario, 10kN load or 13.9 kN/m line load placed on the rear of the left section of the lift plate. Figure 14 visualizes load case 4 with constraints and load.

Figure 14. Load case 4 Line load of 13.9kN/m at the very rear tip, triangles in figure are the constraints in the lift arms, locked in translation ux, uy and uz together with rotations about the x-, y- and z-axies.

7.6. Results, lift plate FE-analysis
To determinate the visual and structural effects from the line loads on the whole lift plate, the results is presented with three plots showing: deflection, von Mises face stress and von Mises core stress in three figures. Results are given from the Ansys listed results, the deflection is also presented in a graphical capture for each FEA run, showing the deflection and maximum von Mises stress situation in the entire structure. Note that these graphical captures shows the deflections multiplied 1000 times to show the effect.

7.7. Results Load case 1
Figure 15 from the FEA run shows the deflection and von Mises stress situation for the entire plate for load case 1. Figure 16, Figure 17 and Figure 18 shows vertical deflection, maximum von Mises face stress and maximum von Mises core stress respectively.

Figure 15. Load case 1 Deflection and von Mises stress situation with 10kN load or 10.5 kN/m line load placed on the middle of the plate and 70% from front. White contours showing the undeformed shape (deflection is multiplied 1000 times.).
Figure 16. Load case 1: Maximum vertical deflection located at rear central, 10.5 kN/m line load acting at 70% from front of the lift plate.

Figure 17. Load case 1: Maximum von Mises face stress in the lower face for central 10.5 kN/m line load at 70% from front of the lift plate.
7.8. Results Load case 2

Figure 19 from the FEA run shows the deflection for load case 2. Figure 20, Figure 21 and Figure 22 shows vertical deflection, maximum von Mises face stress and maximum von Mises core stress respectively.

Figure 19. Load case 2. Deflection and von Mises stress situation with 10 kN or 13.9 kN/m line load. The white contours show the undeformed shape. (Deflection is multiplied 1000 times.)
Figure 20. Load case 2 Maximum vertical lift plate deflection at rear corner with 13.5 kN/m line load acting on 70% from front of the lift plate.

Figure 21. Load case 2 Maximum von Mises face stress in the bottom face with edge load 13.5 kN/m line load acting on 70% from front of the lift plate.
Figure 22. Maximum von Mises core stress with 13.5 kN/m line load on 70% from the front of the lift plate.
7.9. Results Load case 3
Figure 23 from the FEA run shows the deflection for load case 3. Figure 23, Figure 24 and Figure 25 shows vertical deflection, maximum von Mises face stress and maximum von Mises core stress respectively.

Figure 23. Load case 3 Deflection and von Mises stress situation with 10kN load or 10.5 kN/m line load placed on the central rear top

Figure 24. Load case 3 Maximum vertical lift plate deflection at rear central with 10.5 kN/m line load placed on central rear top
Figure 25. Load case 3 Maximum von Mises face stress in lower face with 10.5kN/m line load placed on central rear top.

Figure 26. Load case 3 Maximum von Mises core stress with 10.5kN/m line load placed on central rear top.
7.10. Results Load case 4

*Figure 27* from the FEA run shows the deflection for load case 4. *Figure 28, Figure 29, and Figure 30* shows vertical deflection, maximum von Mises face stress and maximum von Mises core stress respectively.

*Figure 27. Load case 4 Deflection and von Mises stress situation with 10kN load or 13.5 kN/m line load located on the outer corner of the lift plate

*Figure 28. Load case 4 Maximum vertical plate deflection at rear corner with 13.5 kN/m line load.*
Figure 29. Load case 4 Maximum von Mises face stress with 13.5 kN/m line load placed at rear corner.

Figure 30. Load case 4 Maximum von Mises core stress with 13.5 kN/m line load.
7.11. Discussion FE-analysis, lift plate
Maximum allowed deflection for the whole Z-75 tail lift with the lift arms and chassis included is 100 mm. There is no exact number expressed for how great part the lift platform should do in this value, but 50 mm is reasonable and somewhere in the region of what would be acceptable. The plots in the earlier section reveals that almost all four load cases with $t_f$ higher than 2 mm is within that region. However, this may anyhow not be the limitation for the design anyway.

Observation of the figures shows that the maximum face stress and the maximum core stress is very close to each other in all cases besides the last LC4. Maximum face stress is fairly high in all runs but is critical or close to critical in LC4.

Combinations with the stiffer H250 core reach smaller von Mises stress values in the core with small numbers of $t_f$ for load cases 1, 2 and 3. This appears however not to be the case for load case 4 where the difference between the stresses are much more significant. The reason for this is a little unclear but it may be related to the rougher load geometry and corresponding stresses.

Coordinates for the highest face and core stress are in the same node or nearby each other, close to the lift arms at the bottom of the plate for all runs. This is nevertheless also a manifestation of singularity in the model since it’s theoretically infinite in sharp corners. The singularity in the model is anyhow not completely providing fictive results since this is the area where the entire moment from the load will be transferred. The von Mises stress situation can be seen in Figure 31.

![Figure 31. Maximum stress for the aluminium alloy occurs in the bottom face or in the lift arms in direct connection to the maximum stress node in the Divinycell core. This example is taken from LC3.](image)

8. Lift arm examination, FEA calculation
Next detail to be examined little more carefully is one of the lift arms. The major reason is that this is the part that will transform a large part of the loads carried on the plate. Figure 32 represents the set up and the geometric definitions.
The geometric numbers are:

\[
L = 1490 \text{ mm} \\
F = 500 \text{ N} \\
H = 155 \text{ mm} \\
h = 20 \text{ mm}
\]

and the thickness of the cantilever is

\[t = 5 \text{ mm}\]

Aluminium according to Table 1

The force, \(F\) is an assumption for how much of the max load of 10 kN that may be transferred to one of the lift arms only. This problem only looks on one part in the whole complex structure.

8.1. FEA approach

The same geometrical shape was implemented in Ansys 10 and brick shaped SOLID45 elements were used for the calculations. The load is 500 N. The arm has been translational supported along the sides since it will experience the support from the surrounding material in the real case and prevent it from buckeling.

8.2. Results for analytical and FE calculations for lift arm

The FE analysis of the wedge shaped arm resulted in a deflection of 24.9 mm. The graphical result is captured in Figure 33.

**Figure 32.** Wedge shaped solid cantilever, i.e. one of the lift arms

**Figure 33.** FE calculation for lift arm, 500N top load, deflection 24.9mm
9. Core impact due to point load

One issue worth to study more in detail is the consequences of the point loads in the core area close to the face. Large vertical deflection and high levels of stress in the core will cause the core to crush (permanent deformation) and even if the bond between the face in the core is good will this lead to almost the same thing, structural failure and may be considered as a sort of delamination.

This analysis will estimate the magnitude of the deflection that one wheel from a pallet lift will cause the core with respect to the face thickness and core density. In this study only the core and the top face has to be represented. The difficult part is to estimate the load area from the wheels in a proper way. Given dimensions on pallet lift wheels are 45 x 175 mm and 75 x 75 mm (width x diameter).

Keeping in mind that these wheels are relatively large in diameter, the contact surface would be rather flat. An estimated rectangular load area of 45x3 mm i.e. 135 mm² and 75 x 2 mm i.e. 150 mm² for the two different wheels is still a guess and estimation but is probably better than the theoretical line case. A little FE model can visualize this in a good way and the result will be used along with the result from Sandwich lift plate deflection to investigate the final face thickness.

9.1. Model description and calculations, core impact due to point load

Since the estimated bigger area for the smaller wheel should leave a smaller impact on the underlying surface was only the bigger wheel treated. The test section of the core and its upper face was modelled fairly large in dimensions of 2500 x 2500 x (50 + t_f) mm. Where t_f is the thickness of the upper face and 50 is the thickness of the core. t_f was to be varied from 1 mm to 5 mm in steps of 0.5 mm. This test section was considered to be large enough to minimize the influences of stresses at the free ends.

The load 10 kN (~1000 kg) was divided on the pallet lift’s four wheels i.e. 2.5 kN on 135 mm² gave 8.51 MPa. Figure 34 shows the meshed test section with at load area of one pallet wheel with dimensions; 45 x 75 mm diameter.
The model is made with brick shaped Solid45 elements, same as in the lift plate model (see §7. Lift plate model, presentation FE-analysis), but this time is also the face represented with solids rather than shells, since the details in the face-core connection is in focus now. The mesh is refined in the middle of the plate and a zoomed in view can be seen in Figure 35.

Figure 35. Detailed view of the central section 400x300x(40+1.5)mm. Red area in the middle shows the load area with the wheel 75 x 3mm area with 8.9MPa pressure on ten elements which corresponds to a load of 2500N.
9.2. Result, core impact due to point load

The deflection on the single faced sandwich was calculated in Ansys 10 and can be seen in the graphical capture in Figure 36 and Figure 37. Values of interest where chosen from Ansys listed results and can be seen in Figure 38 to Figure 40 as deflection, von Mises face stress and von Mises cores stress, all are plotted with respect to increasing $t_f$.

Figure 36. Von Mises stress situation, view from above. Example shows 3mm steel face with H250 core.

Figure 37. Cross section of the face-pressure model where the wheel have been placed with its axis perpendicular to the view.
Figure 38. Maximum vertical plate deflection right under the load as a function of face thickness.

Figure 39. Maximum von Mises face stress as a function of face thickness.
Figure 40. Maximum von Mises core stress as a function of face thickness

9.3. Discussion, core impact due to point load
From Figure 38 it is seen that the difference in deflection between the higher and lower density core material decreases as the $t_f$ increases. If $t_f$ would increase towards infinity would these two lines meet each other, but at this very part of the scale $t_f$ value of 3 to 3.5 mm gives difference in deflection in numbers around 0.02 mm which may be somewhere in the same region as the uncertainty in the model. Also, by looking at Figure 38 it is observable that the greater part of the load is transferred into the face mainly. The von Mises face stresses in Figure 39 is also low, many times lower than the Yield strength (rp0.2) for aluminium.

The local face pressure of the face appears not to be much of an issue and the dimensional thickness of the faces are more likely to come out of the plate deflection analysis or the local buckling calculations. This section of the report shows that the major part of a point load is absorbed in the face mainly and that the difference in the core material doesn’t affect the point load deflection in a great extent.

10. Local buckling
One issue worth looking at is called Local buckling which is a direct consequence of bending sandwich structures, where the compression face will buckle similar to the Euler case for a beam. It can be described as local instability problem, making the compression face either buckle in to the core or out from the core. The later will also cause the glue bounding to delaminate; both cases anyhow share the same sort of critical influence on the structure and can be seen in Figure 41.
10.1. Calculations Local buckling
The method to predict the local buckling phenomena is to determine the largest negative stress in the compression face and then compare with Hoff’s conservative form

\[ \sigma_{cr} = \frac{G_E}{c_f} \]  

(6)

This will provide a value for max negative stress in the compression face. Note that this equation is independent of the face thickness and the core thickness.

10.2. Result Local buckling
The recently presented relationship (6) is used with Young’s modulus and shear modulus for each material setup. The critical numbers that the maximum negative stress shall not be exceeded are given in Table 5.

<table>
<thead>
<tr>
<th>Material</th>
<th>Div</th>
<th>Critical Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alu – Div H130</td>
<td></td>
<td>424MPa</td>
</tr>
<tr>
<td>Alu – Div H250</td>
<td></td>
<td>647MPa</td>
</tr>
<tr>
<td>Steel – Div H250</td>
<td></td>
<td>956MPa</td>
</tr>
<tr>
<td>GF – Div H130</td>
<td></td>
<td>405MPa</td>
</tr>
</tbody>
</table>

*Table 5. Results from Hoff’s formula for the compression face, i.e the lower face of the sandwich.*
The previous Ansys runs for bending of the lift plate also brought values for the max negative stress in the compression face. These values are in the region of one tenth of the derived numbers from Hoff’s formula (6). All maximum negative principal stresses can be seen in Table 6.

<table>
<thead>
<tr>
<th>Material, Thickness [mm]</th>
<th>LC 1 [MPa]</th>
<th>LC 2 [MPa]</th>
<th>LC 3 [MPa]</th>
<th>LC 4 [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Al 2</td>
<td>-54.58</td>
<td>-112.3</td>
<td>-88.49</td>
<td>-147.38</td>
</tr>
<tr>
<td>Al 2.5</td>
<td>-50.93</td>
<td>-97.842</td>
<td>-82.38</td>
<td>-135.59</td>
</tr>
<tr>
<td>Al 3</td>
<td>-47.95</td>
<td>-90.714</td>
<td>-77.37</td>
<td>-126.30</td>
</tr>
<tr>
<td>Al 3.5</td>
<td>-45.43</td>
<td>-85.029</td>
<td>-73.17</td>
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</tr>
<tr>
<td>Al 4</td>
<td>-43.28</td>
<td>-80.36</td>
<td>-69.56</td>
<td>-112.36</td>
</tr>
<tr>
<td>Div H250, Steel 2</td>
<td>-54.67</td>
<td>-122.76</td>
<td>-88.48</td>
<td>-56.52</td>
</tr>
<tr>
<td>Div H130, GF 4</td>
<td>-42.22</td>
<td>-79.75</td>
<td>-69.36</td>
<td>-111.78</td>
</tr>
</tbody>
</table>

Table 6. Largest negative stress values in lower face MPa from FE-calculations for corresponding load case

10.3. Discussion Local buckeling
Results from Ansys runs compared with numbers from Hoff’s formula (6) implies that face buckling not will be an issue in this sandwich design. Since the load placement in LC2 and LC4 only are acting on the outer part of the plate and the load cannot be picked up in the built in arms, these two load cases also give the highest negative principal stress.

11. Manufacturing aspects
Since the lift plate design is a mix of materials with rather different properties (the core is considered to be more porous while the metals are solid) the only conventional way to connect them with each other is by gluing. A more elastic glue, like for example Sikaflex may be preferable to absorb some of the deflection from point loads that locally can be transformed to the core. Divinycell can locally only handle limited strains/deformation.

The attachment to the liftarms at the bracket can be weld, see Figure 42.

Figure 42. One pair of lift arms with attachment bracket. The core part between the lift arms are excluded here. The parts may be joined with a weld from the outside like illustrated.
The bottom and top face plates are made in sheet aluminium. The top side plate is cut out of a tear plate to ensure a good grip.

Other interfaces such as the vertical side cover plates and where the top and bottom faces meets the string casted profiles can be joined to each other by either welding or gluing. Both have their advantage and disadvantages. Welding will probably contribute to a stiffer plate but glue may be more suitable if the interface between the faces and the core is chosen to be somewhat flexible.

12. Final conclusions and discussion

12.1. Mechanical strength
The objectives from § 1.3. Requirements are summarized with the results from the analysis.

  - The requirement for maximum deflection 50 mm (for the lift plate) was not exceed in any of the four load cases. Approximately 100 % margin for Load case 4 with 2 mm faces and H130 core.
  - The core/face ratio at the tip gives a face thickness of minimum 2.5 mm according to §6.1.
  - The von Mises stress for the face material exceeds the Ultimate strength (Rm) 275 MPa Aluminium alloy AlmgSi0.7 F27 (6005A T6) for 2 mm faces considering Load case 4 with 290 MPa and 250 MPa for the H130 and H250 core respectively.
  - The maximum negative principal face stress caused by bending of the plate is small compared with the calculated limiting stress for Local buckling, see § 8 Local buckeling. Smallest margin is 187% for Load case 4 with H130 core.
  - The mass of the new design with 3 mm faces and H130 and H 250 core is 110 kg respectively 158 kg.
  - The mass of the new design with 2 mm faces and H130 and H 250 core is 90 kg respectively 138 kg.
  - The core compression effects due to point loads are small, approximately 0.23 mm for a point load of 2500 N with a 2 mm face and H130 core according to §9. Core impact due to point load.

Based on these results, the sandwich lift plate will be designed with:

- The H130 Divinycell core, to minimize the mass of the plate.
- The 3 mm aluminium faces, to meet the requirement of deflection

With this combination, the requirement for this master thesis considering mechanical strength is achieved. The sandwich lift plate demonstrates a very good result in terms of deflection and stresses. The mass of the lift plate is something that needs to further examined since that objective is not meet yet (Zepro Z-75, 1450 x 2200 mm, 67 kg).

Long terms characteristics like crush and fatigue combination effects for the core are hard to predict or model in a FE analysis. The finally answer to the quest of a reliable, cheap and light lift plate can probably be best answered if a prototype sandwich lift plate model is manufactured.
12.2. Manufacturing cost

Material cost is a relative and flexible charge, also in direct connection to quantity and quality. This design can anyhow be divided into three parts:

- Divinycell expense. Divinycell can be bought in different thicknesses. The price (5th of June 2015) for 50 mm thick H130 is 1529 SEK/m² and the price for a 50 mm H250 is 2947 SEK/m² (no VAT included). For the volume of this core it gives a price of 7649 SEK and 15206 SEK respectively.

  Also, since this master thesis started, new cheaper materials such as Diabs PET-foam PX300 have been developed. It is used as floor in the cargo compartments and the price is approximately half of the Divinycell H-series. Another 20-30% discount is to expect for series production.

- String casted aluminium profiles expense, three pieces. Prices taken from Zepros price list alternate in a price range from 0.9 SEK/cm to 3 SEK/cm (depending on the complexity of the profile). This leads to an approximate cost of 600 SEK to 1980 SEK.

- Aluminium plates. Aluminium flat plate 2500 x 1250 x 3 mm costs approximately 35 SEK and aluminium tear plate 3000 x 1500 x 5/3 costs 37 SEK.
13. Future work

Improvements of the design can be done. Both to keep the quantity of Divinycell down since this is the major cost, but also to suppress the mass of the sandwich platform. It may be worth looking at other shapes, for examples lift plates with uniform thickness where one layer of 50 mm (or 40 mm) Divinycell can be used. One example on a simpler design can be seen in Figure 43.

![Figure 43. Example on a uniform lift plate. Lift arms attached to the outside of the plate only.](image)

This model has also a reduced amount of parts, one single piece of core, two faces, two attachments brackets with lift arms. The areas at the tip and at the bottom (close to the cargo compartment) have to be closed in some way and this time has the tapering towards the tip been designed by thinning out the core part and folding of the bottom face. The other part towards the cargo compartment has been closed with an aluminium profile see Figure 44.

![Figure 44. Side profile of the uniform sandwich lift plate. String casted aluminium profile is](image)

14. References

   http://www.diabgroup.com


[3] Handbok och formellsamling i hållfasthetslära