UiT THE ARCTIC UNIVERSITY OF NORWAY

Faculty of Engineering Science and Technology

Optimization of the mass of a sandwich plate that TAM is producing

Final report

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Bjarne Steinulv Jensen

Master thesis in engineering design, June 2017

Preface

On the last semester of the masters in Engineering design is it written a thesis the counts 30 credits. The thesis problem was presented in a list given to the class, where the students could chose after their own desire. The problem *Numerical calc. and optimization of sandwich components (TAM AS)* was chosen since it was for an external company and a good way to see how the industry works.

The candidate gained a significant increase of knowledge in the field of sandwich components. This have the thesis supervisors Dag Lukkassen and Annette Meidell have guided and given good advice to the candidate over the last semester. The meeting with personal at TAM on May 16th 2017 gave a good understanding of the production and challenges with sandwich constructions. The personal at TAM also provided the construction with specific load conditions to optimize that this thesis is based on.

Due to reasons that is not connected to the studies, the project had a slow progress. But the last part of the semester, the progress has been much better.

Acknowledgements

Would like to thank everyone who helped me with this thesis.

____________________ Narvik 06.06.17

Bjarne S. Jensen

Abstract

In this report the possibility to optimize the mass of a sandwich plate that TAM produces have been reviewed. Dimensions of the plate is 2602mm x 2404mm with a core thickness of 40mm, top facing of 3mm and bottom facing of 1mm. To simplify the computations they calculate with uniform facings of 1mm. The function of the sandwich plate is to lift livestock with wires that are fastened in the four corners. The maximum load conditions is set to be a uniformly distributed load of 20.000N and to withstand the impact forces, the top facing has an increased thickness.

The analytical computations gives that a plate that is 11mm thicker, but have a significantly lower density gives a lower mass and less deflection than the original plate. The results given by ANSYS APDL confirms the analytical computations, but the results from ANSYS Workbench is concluded to be unreliable for sandwich constructions.

The increase in thickness should not affect the overall use of the plate since it still fits in the frame, and the frame is significantly thicker than both the new and original plate.

Contents

Introduction

TAM is a small company located at Andslimoen in Troms which is in the northern part of Norway. The name TAM comes from the initials of the founder, Tor Arne Mentzoni [1]. They specialize in production of sandwich constructions to for an example the Norwegian military, for helicopter lifts and other extreme conditions.
The construction that TAM want optimized in this thesis is a plate used to lift livestock into a livestock transport container which also is made up of sandwich plates. Reducing the mass will make the transport able to transport more livestock for each trip, or reduce the fuel consumption for each trip.

Contact person at TAM is Herman Myrvoll.

Thesis supervisors
The thesis supervisors are Professor Dag Lukkassen and Professor Annette Meidell, both are internal supervisors assigned from UiT campus Narvik.

Problem description

The computations in this thesis is based on algorithms from the report *"Optimal stiffness design of sandwich plates with variable core densities"* by Dag Lukkassen, Annette Meidell and Herman Myrvoll [2], this report is attached in appendix C.
The goal for this thesis is to optimize the mass of a sandwich plate that TAM is producing. A sandwich plate supported by a frame that is supported in four points is subjected to uniformly distributed load. It has a length of 2602mm and a width of 2404mm with a divinycell H60 core from Diab and aluminum faces, the top with thickness 3mm and the bottom plate with a thickness of 1mm. The reason the top facing is 3mm thick, is to withstand impact forces from when the animals kick the plate when loading. To simplify the analytical computations for uniformly distributed load, the top facing is reduced to 1mm. In the results, the top facing thickness will the 2mm be added to after all computations are done.
The results given by the analytical computations will then be compared to simulations of the same construction in the numerical calculation tool ANSYS. The optimized construction will then be compared to the original with respect to other general parameters than total mass.
This thesis will be restricted to only consider aluminum facings for the sandwich construction, but the core material will all densities of the core materials Divinycell from Diab or equivalent be considered [3].

Production method

Figure 1 - Core configuration with a roll of aluminum facing in the background.

Figure 2 - Core configuration

The core materials and facings must be cut in to the desired size before the gluing process can begin. This is because the glue has to be set under vacuum within an hour or it will cure prematurely. The layout of core material in figure 1 and 2 is for the floor to the container for transportation of livestock.

In the background of figure 1 it is a roll of aluminum used for facing.

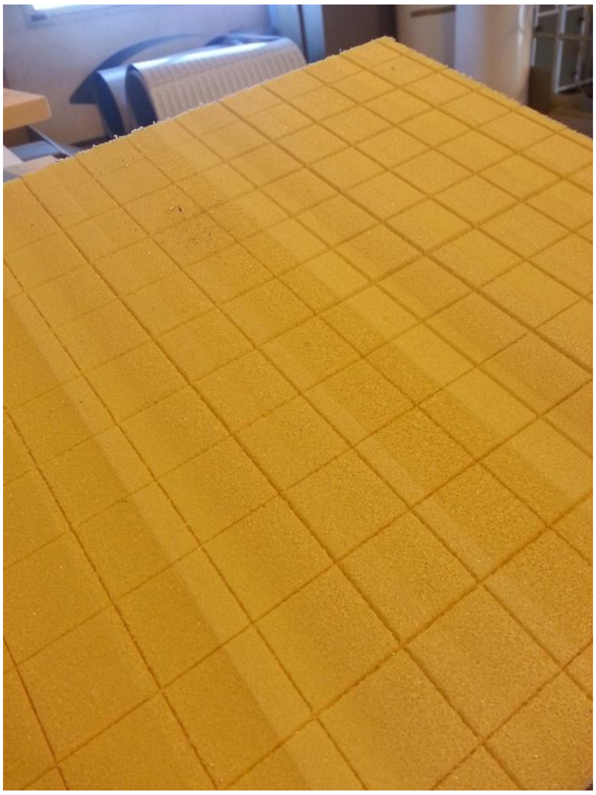

Figure 3 - Close-up of core material

The core material have precut groves in a grid formation like the material in figure 3. This is to ensure that all air is eliminated when the sandwich plate are vacuum pressed. It is kept in vacuum for a minimum of eight hours for the glue to completely cure.

The glue used in the production have higher shear stiffness than the core material. This is to ensure that if the sandwich panel should fail, it is not the glue that fails.

Figure 4 - Complete panels

The sandwich plates in figure 4 is ready to assemble, the final product here is the container for transportation of livestock. The plates are then assembled with aluminum profiles.

Material properties

The tables bellow does not list properties that is non-essential, only properties for generic aluminum and the two different core materials that is used. A list of other core materials from Diab's Divinycell H group is attached in appendix B. Only Divinycell group H is considered since all foams in group H have the same non-relevant properties in regard to mass and shear stiffness. This because if there is a property in this group that is required for this plate that was not given by TAM.

Table 1 - Relevant properties of aluminum [4]

Property	**Value**	**Unit**
Young's modulus	70	[GPa]
Density	2700	[kg/m^3]
Poisson ratio	0,33	-

Table 2 - Relevant properties of divinycell H60 [3]

Property	**Value**	**Unit**
Shear modulus	20	[MPa]
Density	60	[kg/m^3]
Poisson ratio	0,4	-

Table 3 - Relevant properties of divinycell H35 [3]

Property	**Value**	**Unit**
Shear modulus	12	[MPa]
Young's modulus	33,6	[MPa]
Density	38	[kg/m^3]
Poisson ratio	0,4	-

Analytical computations

All computations are in chronological order attached in appendix A. The computations are done in PTC Mathcad Prime 3.0.

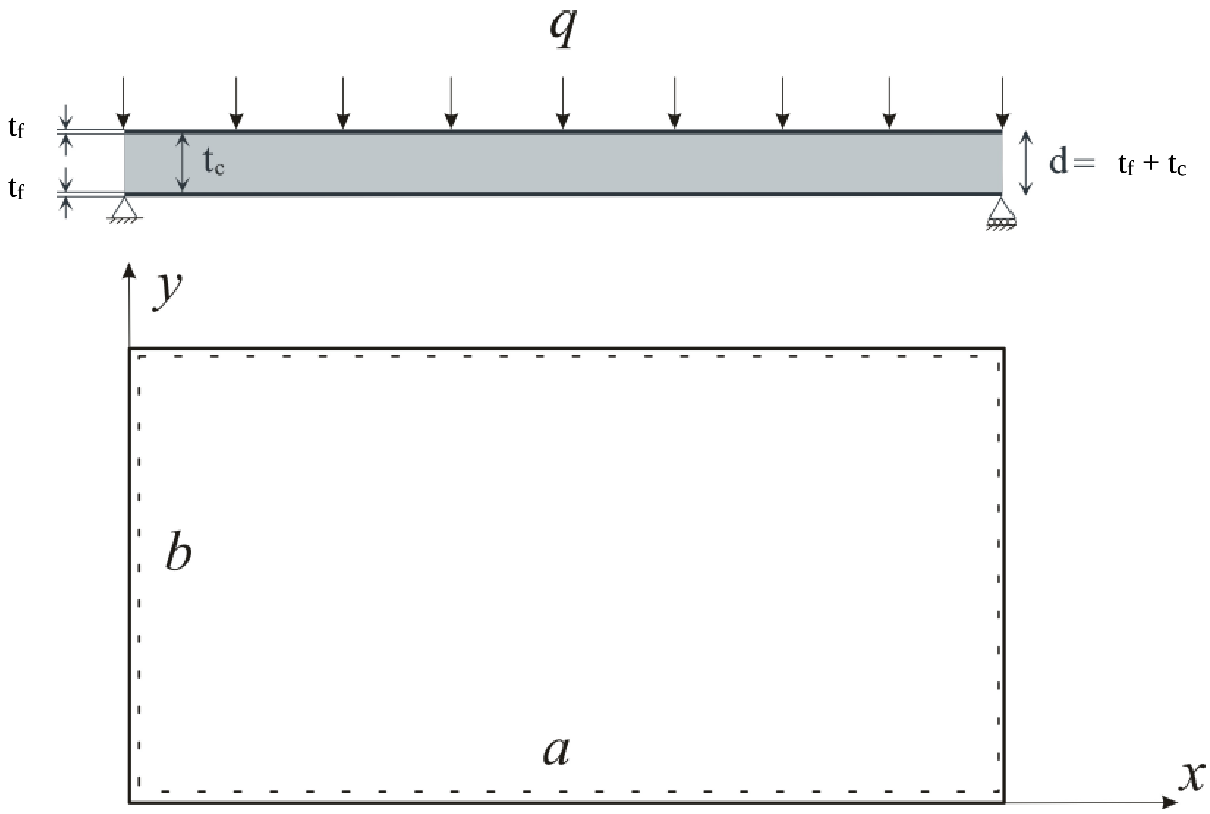

Figure 5 – Illustrating sandwich plate

Figure 5 above illustrates what some of the different variables that occurs later in this chapter. The figure is from *"Optimal stiffness design of sandwich plates with variable core densities"* [2], it is made small alterations to accommodate the denotations in the formulas in this report.

Variables and constants

Table 4 - Units and denotation of variables [5]

Variables	Dentation	Unit
Length	a	[m]
Width	b	[m]
Face thickness	t_f	[m]
Core thickness	t_c	[m]
Poisson ratio for facing	v_f	-
Young's modulus for facing	E_f	[GPa]
Shear Modulus of core	G_c	[MPa]
Uniformly distributed load	q_{mn}	[Pa]
Total deflection	w_{total}	[mm]
Deflection due to pure bending	w_b	[mm]
Deflection due to pure shear deformation	w_s	[mm]
Mass	m	[kg]
Density of core	ρ_c	[kg/m^3]
Density of face	ρ_f	[kg/m^3]

Table 5 - Value of constants [2]

Constants	Value	Unit
k	6080/1533	[s^2/m^2]
l	17/1533	[1/MPa]
v	194.198*10^{-3}	[1/kg]

Computations and description of them

There is used several formulas from "*Optimal Stiffness Design of Sandwich Plates with Variable Core Densities"* [2] to analyze and optimize the mass of the sandwich plate. There is assumed thin faces for all analytical computations.

The deflection can be computed with the formulas as shown below where w_b is the deflection from bending and w_s is from shear deformation. Sum up w_b and w_s to get the total maximum deflection w_{total}.

$$w_b = \frac{1 - v_f^2}{D} \sum_{n=1}^{\infty} \sum_{m=1}^{\infty} \frac{q_{mn} \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right)}{\left(\left(\frac{m\pi}{a}\right)^2 + \left(\frac{n\pi}{b}\right)^2\right)^2}$$

$$w_s = \frac{1}{S} \sum_{n=1}^{\infty} \sum_{m=1}^{\infty} \frac{q_{mn} \sin\left(\frac{m\pi x}{a}\right) \sin\left(\frac{n\pi y}{b}\right)}{\left(\frac{m\pi}{a}\right)^2 + \left(\frac{n\pi}{b}\right)^2}$$

$$D = \frac{E_f t_f d^2}{2}, \quad S = \frac{G_c d^2}{t_f}$$

Dan Zenkert's work [5], An Introduction to sandwich Constructions, states that:

> *The series converge rather quickly for the deflections and bending moments… The maximum deflection and bending moments appear in the middle of the plate at (x,y)=(a/2,b/2)…* [5].

From "*Optimal Stiffness Design of Sandwich Plates with Variable Core Densities"* [2], the uniformly distributed load on a plate where the load $q_{mn} > 0$ are:

$$q_{mn} = \frac{16 q_{mn}}{mn\pi^2}$$

Optimization of sandwich plates

$$w_b = \frac{q_{mn}(1-v^2)b^4}{D} f\left({}^{a}\!/_{b}\right), \qquad w_s = \frac{q_{mn}b^2}{S} g\left({}^{a}\!/_{b}\right)$$

Maximum deflection accurse at the center of the plate since it is an uniformly distributed load, at:

$$x = \frac{a}{2}, \ \ y = \frac{b}{2}$$

This gives that:

$$f\left({}^{a}\!/_{b}\right) = \sum_{n=0}^{27}\sum_{m=0}^{27} \frac{16\sin\left(\frac{(2m+1)\pi}{2}\right)\sin\left(\frac{(2n+1)\pi}{2}\right)}{\pi^6(2m+1)(2n+1)\left(\left(\frac{(2m+1)}{{}^{a}\!/_{b}}\right)^2 + (2n+1)^2\right)^2}$$

$$= 4{,}728 * 10^{-3}$$

$$g\left({}^{a}\!/_{b}\right) = \sum_{n=0}^{27}\sum_{m=0}^{27} \frac{16\sin\left(\frac{(2m+1)\pi}{2}\right)\sin\left(\frac{(2n+1)\pi}{2}\right)}{\pi^4(2m+1)(2n+1)\left(\left(\frac{(2m+1)}{{}^{a}\!/_{b}}\right)^2 + (2n+1)^2\right)}$$

$$= 79{,}452 * 10^{-3}$$

Note that $f\left({}^{a}\!/_{b}\right)$ and $g\left({}^{a}\!/_{b}\right)$ is denoted $f_{a.b}$ and $g_{a.b}$ to accommodate PTC Mathcad Prime 3.0 as attached in appendix A.

$$D = \frac{E_f * t_f * d^2}{2} \qquad S = \frac{G_{cH60} * d^2}{t_c} \qquad v = \frac{\frac{w_{total}}{q_{mn}}}{g\left({}^{a}\!/_{b}\right) * k * a * b^3} + \frac{l}{a * b * d * k}$$

The variables shown above is used to shorten the mathematical expressions that follows in the report.

Optimization of sandwich plates

The total deformation of the original plate is 9,4mm, as shown by the computations done in the equations bellow:

$$w_b = \frac{q_{mn} * \left(1 - v_f^2\right) * b^4}{D} f\left({}^{a}/_{b}\right) = 7{,}647mm$$

$$w_s = \frac{q_{mn} * b^2}{S} g\left({}^{a}/_{b}\right) = 1{,}747mm$$

$$w_{total} = w_s + w_b = 9{,}394mm$$

The extremal value of t = t_{f0} = 1,074mm as seen bellow, which is thicker than the original thickness of the facings that is in the analytical computations. Since the top facing in reality is 3mm, this should be sufficient. The formula bellow is only valid when t_{f0} is significantly smaller than d [2].

$$t_{f0} = \frac{1}{v}\left(\sqrt{\frac{\left(1 - v_f^2\right)}{g\left({}^{a}/_{b}\right) * E_f * d^2 * a^2 * \rho_f * k} f\left({}^{a}/_{b}\right)} + \frac{2 * \left(1 - v_f^2\right) * b}{g\left({}^{a}/_{b}\right) * E_f * d^2 * a * k} f\left({}^{a}/_{b}\right)\right)$$

$$= 1{,}074$$

The formula for density based of t_0 with variable core thickness is then used to make the graph bellow to evaluate the best density choice. The formula is given bellow and d is ranging from 10mm to 65mm with an increment of 5mm per point made in excel. The graph shows that a divinycell H core with a density of 38 $\frac{kg}{m^3}$ and thickness d of 50mm is the best match [3].

$$\rho_{t0} = \frac{1}{a * b * d * v}\left(1 - \left(\sqrt{\frac{1}{\frac{4 * \left(1 - v_f^2\right) * b^2 * \rho_f}{g\left({}^{a}/_{b}\right) * E_f * d^2 * k} f\left({}^{a}/_{b}\right)}} + 1\right)^{-1}\right)^{-1}$$

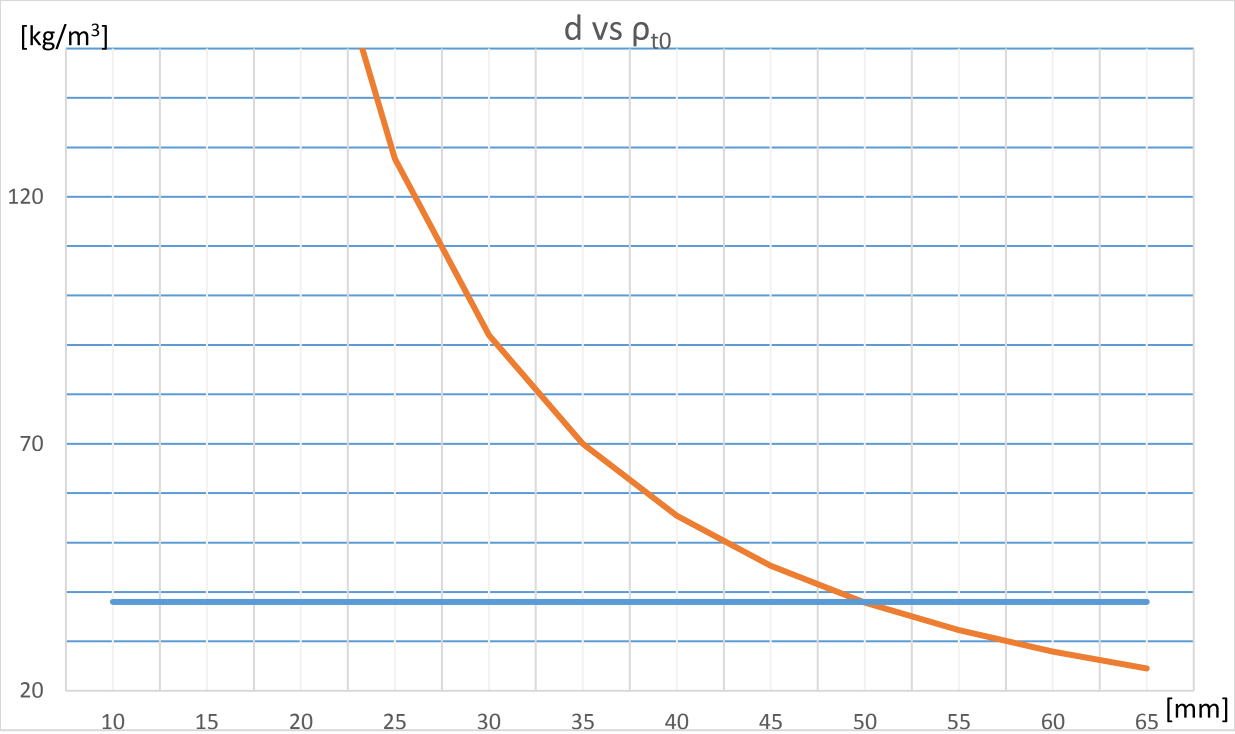

Figure 6 - Graph that show where divinycell H35's density intersects with $\rho_{t0}(d)$

Figure 5 shows that to use divinycell H35, d needs to be around 50mm. This is confirmed by the computation bellow as well.

$$\rho_{t050} = \frac{1}{a * b * d_{50} * v}\left(1 - \left(\sqrt{\frac{1}{\frac{4 * (1 - v_f^2) * b^2 * \rho_f}{g\left({}^{a}/_{b}\right) * E_f * {d_{50}}^2 * k} f\left({}^{a}/_{b}\right)}} + 1\right)^{-1}\right)^{-1} = 37{,}868 \frac{kg}{m^3}$$

The new minimum facing thickness (t_{f050}) then becomes:

$$t_{f050} = \frac{1}{v}\left(\sqrt{\frac{(1 - v_f^2)}{g\left({}^{a}/_{b}\right) * E_f * {d_{50}}^2 * a^2 * \rho_f * k} f\left({}^{a}/_{b}\right)} + \frac{2(1 - v_f^2) * b}{g\left({}^{a}/_{b}\right) E_f * {d_{50}}^2 * a * k} f\left({}^{a}/_{b}\right)\right)$$

$$= 765{,}665 \mu m$$

Optimization of sandwich plates

Finally, when the original and the new plate compared, the mass of the new plate has been reduced by 6,9% compared to the original. This can be seen in the equations bellow.

$$m_{original} = a * b * 2 * t_f * \rho_f + a * b * t_c * \rho_c = 48{,}791kg$$

$$m_{new} = a * b * 2 * t_{f50} * \rho_f + a * b * \left(d_{50} - t_{f50}\right) * \rho_{c50} = 45{,}425kg$$

$$m_{improvement} = 100 - \frac{m_{new}}{m_{original}} * 100 = 6{,}9\%$$

In addition to the improvement in mass, the deflection of the plate is reduced significantly. The improvement is 19,7%, this can be seen by the equations bellow.

$$w_{total50} = w_{s50} + w_{b50} = 7{,}54mm$$

$$w_{total} = w_s + w_b = 9{,}394mm$$

$$w_{improvement} = 100 - \frac{w_{total50}}{w_{total}} * 100 = 19{,}736\%$$

Numerical computations

The numerical computations three different ways to evaluate the best way of compute the deflection of the sandwich plate. The geometry needed to do the numerical computations in ANSYS Workbench is made in SolidWorks 2015. The drawings is attached in appendix H.

ANSYS Workbench

The computations is done twice with ANSYS Workbench due to not unexpected deflection results of 20,5mm in the first simulation. This is much more that the analytical result, just as predicted in the meeting at TAM. To compensate for the deflection, the frame that the sandwich plate is supposed to rest in is added to make the sides more rigid for the second simulation.

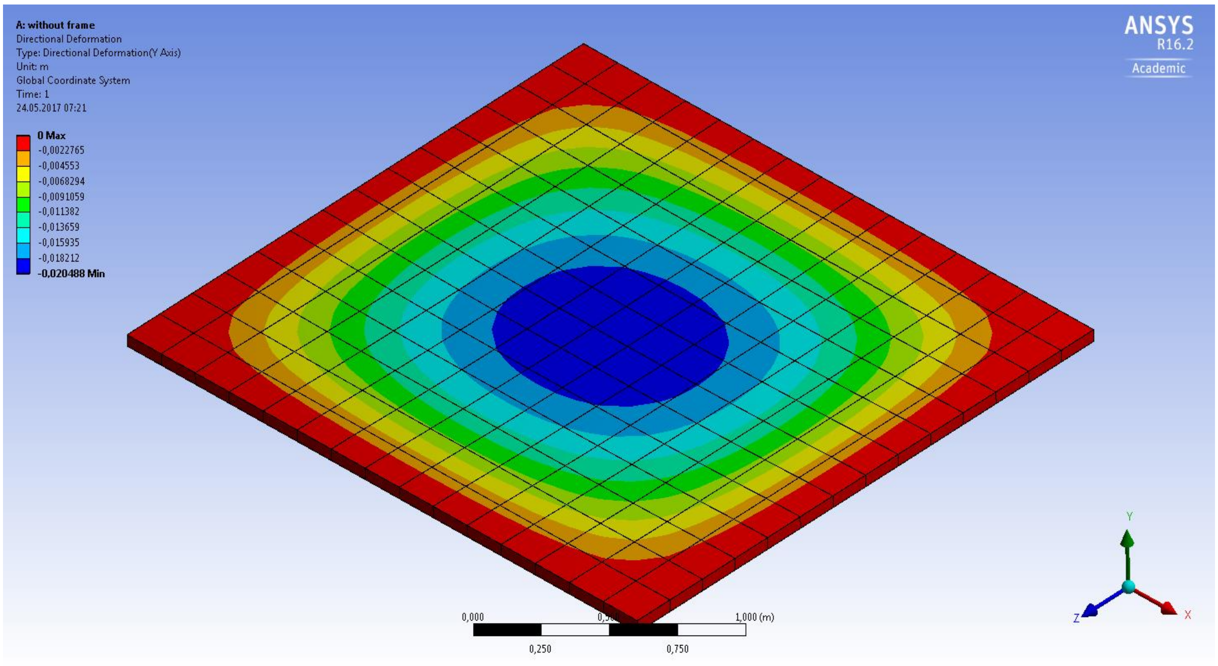

Figure 7 - ANSYS Workbench simulation without frame

Simulation in ANSYS Workbench of the sandwich panel when it is subjected to the same uniformly distributed load as in the analytical computations is shown in figure 6. With fixed supported sides, the sandwich panel has a maximum deflection of 20,5mm. The ANSYS project report that ANSYS generates attached in appendix E.

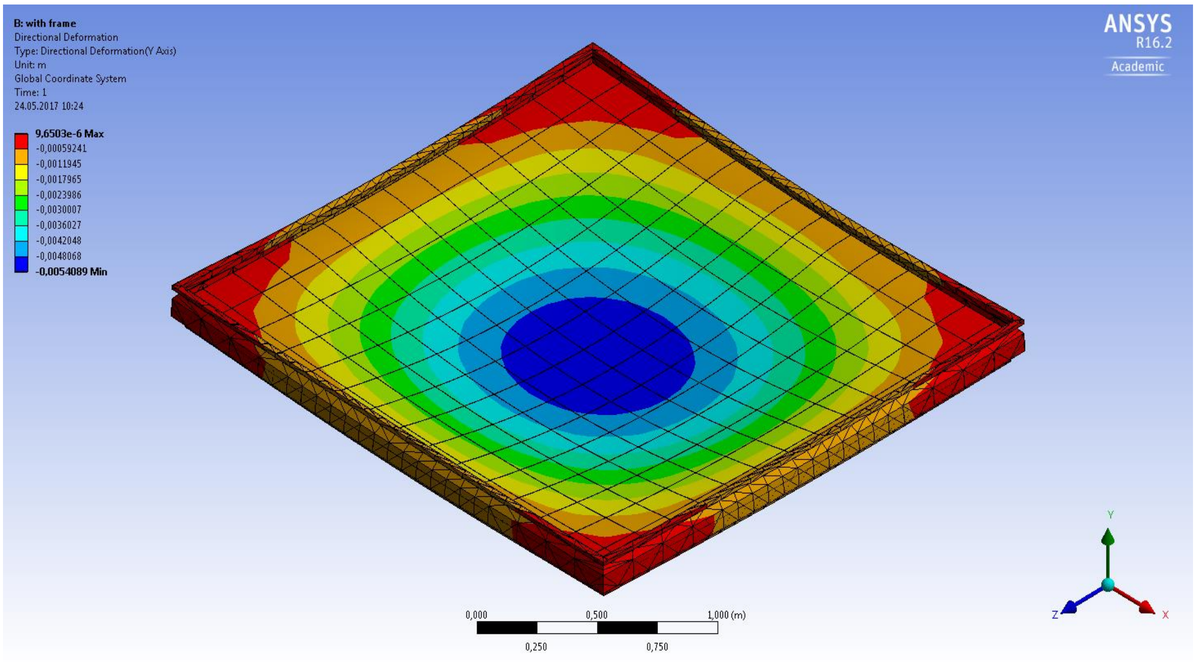

Figure 8 - ANSYS Workbench simulation with frame

Simulation in ANSYS Workbench of the sandwich panel when it is subjected to the same uniformly distributed load as in the analytical computations is shown in figure 7. In addition, the frame that’s supporting the panel is fixed in is added to make the sides more rigid. The plate is supported in a manner such that one corner is fixed in all directions and the other tree is only fixed in the y-direction. With this configuration, the deflection is only 5,4mm. The ANSYS project report that ANSYS generates attached in appendix F.

ANSYS APDL

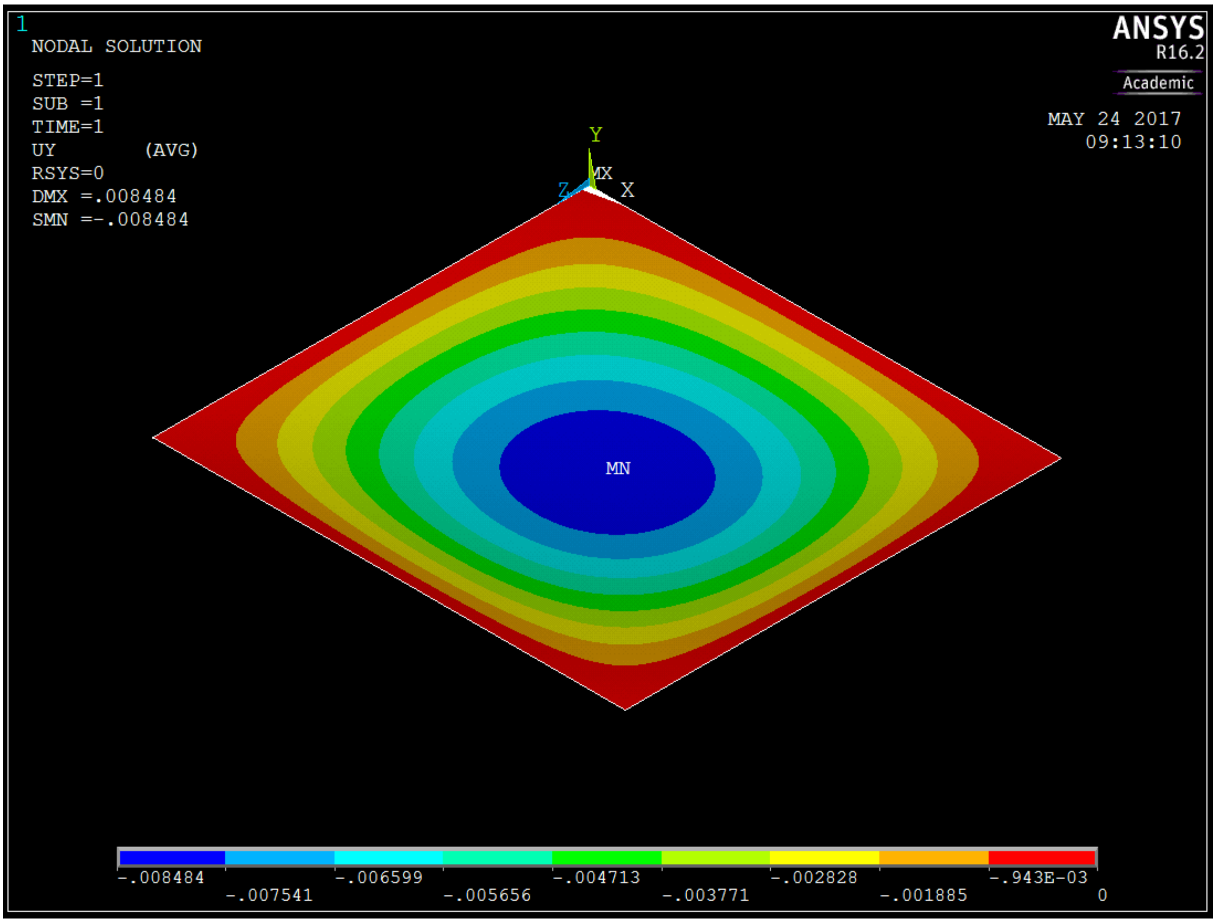

Figure 9 - ANSYS APDL simulation

Simulation in ANSYS APDL of the sandwich panel when it is subjected to the same uniformly distributed load as in the analytical computations is shown in figure 8. The plate is supported in a manner such that one corner is fixed in all directions and the other tree is only fixed in the y-direction. With this configuration, the deflection is 8,5mm. To reconstruct the simulation, the log file is attached in appendix D.

Results

The results is listed in Table 6 - Results bellow. They shows a significant decrease of mass in the plate with the divinycell H35 core compared to the Divinycell H60, 6,9% less mass.

In addition, the deflection is also decreased significantly, in the analytical computation the deflection is reduced by 19,7%. The numerical results varies some, this is due to the different conditions of the geometry in the ANSYS workbench computations and that ANSYS Workbench is not as well set up for simulating sandwich constructions as ANSYS APDL.

Table 6 - Results

	Analytical		**Numerical computations with Divinycell H35**		
	Divinycell H60	Divinycell H35	ANSYS Workbench		ANSYS APDL
			With frame	Without frame	Without frame
Mass [kg]	48,791	45,425	45,425	45,425	45,425
Deflection [mm]	9,39	7,54	5,4	20,5	8,5

Conclusion

The analytical computation and the ANSYS APDL results are relatively close, and more importantly both shows that the new plate is stiffer than the original one.

The result from workbench is less reliable, the result without a frame gives a much higher deflection than all the other results. This was predicted by the personnel at TAM and in their inquiry to their similar result with ANSYS support. They suggested to add a simple frame to stiffen the sides to counter ANSYS Workbench inadequate boundary condition settings for sandwich construction. Therefore the frame the plate was supposed to be fixed inn was added in the final simulation in ANSYS Workbench, resulting in a significantly less deflection than any of the other results.

In all computations the top facing is 1mm, but it should be 3mm to be able to withstand impact forces, but the plate should only get less deflection and the same increase in mass for both core materials. Concluding that it only improves the construction.

From this the conclusion is that the results from ANSYS Workbench is inadequate to use to simulate sandwich constructions. But the analytical and the simulation in ANSYS APDL shows that it is possible to optimize the mass of the sandwich plate.

The new plate is 11mm thicker, but it still fits in the frame and therefore does not affect the overall thickness of the construction.

Assuming the new plate can withstand the impact forces it will be subjected to, there is no negative properties compared to the original plate.

Recommendations for future work

Future work should be to analyze the impact forces to if the thickness of the top facing can be reduced to improve the mass.

Optimize the mass of the rest of the livestock transport should also be done to reduce the fuel consumption or/and increase the transport capacity of the livestock transport.

Also make ANSYS Workbench better suited for simulating sandwich panels if possible.

List of figures

List of tables

References

[1] V. Lekang, "nye-troms.no," Nye Troms, 16 08 2015. [Online]. Available: http://nye-troms.no/nyheter/blir-herrer-i-eget-hus/19.3428. [Accessed 02 06 2017].

[2] A. M. a. H. M. Dag Lukkassen, "Optimal stiffness design of sandwich plates with variable core densities," AIP Publishing, Narvik, 2014.

[3] Diab, "www.diabgroup.com," Diab, February 2016. [Online]. Available: http://www.diabgroup.com/en-GB/Products-and-services/Core-Material/Divinycell-H. [Accessed 11 May 2017].

[4] MIT, "www.mit.edu," [Online]. Available: http://www.mit.edu/~6.777/matprops/aluminum.htm . [Accessed 15 may 2017].

[5] D. Zenkert, An introduction to Sandwich construction, Worcester, UK: Engineering Materials Advisory Servicec Ltd., 1995.

Appendix A – Analytical computations

Dimentions of plate

$a:=2602\ mm \quad b:=2404\ mm \quad t_c:=40\ mm \quad t_f:=1\ mm \quad Areal:=a\cdot b$

$G_{cH60}:=20\ MPa \quad E_f:=70\ GPa \quad v_f:=0.33 \quad F:=20000\ N$

$\rho_f:=2700\ \frac{kg}{m^3} \quad \rho_c:=60\ \frac{kg}{m^3}$

$$q_{mn}:=\frac{F}{Areal}=3.197\ kPa$$

$$d:=t_f+t_c$$

$$D:=\frac{E_f\cdot t_f\cdot d^2}{2} \qquad S:=\frac{G_{cH60}\cdot d^2}{t_c}$$

$$f_{a.b}:=\sum_{n=0}^{27}\sum_{m=0}^{27}\frac{16\sin\left(\frac{(2m+1)\pi}{2}\right)\sin\left(\frac{(2n+1)\pi}{2}\right)}{\pi^6\ (2m+1)\ (2n+1)\left(\left(\frac{2m+1}{\frac{a}{b}}\right)^2+(2n+1)^2\right)^2}=4.728\cdot 10^{-3}$$

$$g_{a.b}:=\sum_{n=0}^{27}\sum_{m=0}^{27}\frac{16\sin\left(\frac{(2m+1)\pi}{2}\right)\sin\left(\frac{(2n+1)\pi}{2}\right)}{\pi^4\ (2n+1)\ (2m+1)\left(\left(\frac{2m+1}{\frac{a}{b}}\right)^2+(2n+1)^2\right)}=79.452\cdot 10^{-3}$$

$$w_b:=\frac{q_{mn}\cdot\left(1-v_f^2\right)\ b^4}{D}\ f_{a.b}=7.647\ mm$$

$$w_s:=\frac{q_{mn}\cdot b^2}{S}\ g_{a.b}=1.747\ mm$$

$$w_{total}:=w_s+w_b=9.394\ mm$$

Optimization of sandwich plates

$$k := \frac{6080\ kg}{1533\ m^3 \cdot MPa} = \left(3.966 \cdot 10^{-6}\right) \frac{s^2}{m^2} \qquad l := \frac{17}{1533\ MPa}$$

$$v := \frac{\left(\frac{w_{total}}{q_{mn}}\right)}{g_{a.b} \cdot k \cdot a \cdot b^3} + \frac{l}{a \cdot b \cdot d \cdot k} = \left(268.816 \cdot 10^{-3}\right) \frac{1}{kg}$$

$$\left(\frac{w_{total}}{q_{mn}}\right) = \left(2.938 \cdot 10^3\right) \frac{mm}{MPa}$$

$$t_0 := \frac{1}{\frac{\frac{w_{total}}{q_{mn}}}{g_{a.b} \cdot k \cdot a \cdot b^3} + \frac{l}{a \cdot b \cdot d \cdot k}} \left(\sqrt[2]{\frac{\left(1 - v_f^2\right)}{g_{a.b} \cdot E_f \cdot d^2 \cdot a^2\ \rho_f \cdot k}\ f_{a.b}} + \frac{2\left(1 - v_f^2\right) b}{g_{a.b} \cdot E_f \cdot d^2 \cdot a \cdot k}\ f_{a.b}\right) = 1.074\ mm$$

$d_{10} := 10\ mm$ $\quad d_{15} := 15\ mm$ $\quad d_{20} := 20\ mm$ $\quad d_{25} := 25\ mm$

$d_{30} := 30\ mm$ $\quad d_{35} := 35\ mm$ $\quad d_{40} := 40\ mm$ $\quad d_{45} := 45\ mm$

$d_{50} := 50\ mm$ $\quad d_{55} := 55\ mm$ $\quad d_{60} := 60\ mm$ $\quad d_{65} := 65\ mm$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{10} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1 - v_f^2\right) b^2 \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{10}^{\ 2} \cdot k}\ f_{a.b}}} + 1\right)^{-1}\right)^{-1} = 708.806\ \frac{kg}{m^3}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{15} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1 - v_f^2\right) b^2 \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{15}^{\ 2} \cdot k}\ f_{a.b}}} + 1\right)^{-1}\right)^{-1} = 328.24\ \frac{kg}{m^3}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{20} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1 - v_f^2\right) b^2 \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{20}^{\ 2} \cdot k}\ f_{a.b}}} + 1\right)^{-1}\right)^{-1} = 192.069\ \frac{kg}{m^3}$$

Optimization of sandwich plates

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{25} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{2}\right) b^{2} \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{25}^{2} \cdot k} f_{a.b}}} + 1 \right)^{-1} \right)^{-1} = 127.682 \, \frac{kg}{m^{3}}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{30} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{2}\right) b^{2} \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{30}^{2} \cdot k} f_{a.b}}} + 1 \right)^{-1} \right)^{-1} = 91.972 \, \frac{kg}{m^{3}}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{35} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{2}\right) b^{2} \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{35}^{2} \cdot k} f_{a.b}}} + 1 \right)^{-1} \right)^{-1} = 69.999 \, \frac{kg}{m^{3}}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{40} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{2}\right) b^{2} \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{40}^{2} \cdot k} f_{a.b}}} + 1 \right)^{-1} \right)^{-1} = 55.451 \, \frac{kg}{m^{3}}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{45} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{2}\right) b^{2} \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{45}^{2} \cdot k} f_{a.b}}} + 1 \right)^{-1} \right)^{-1} = 45.282 \, \frac{kg}{m^{3}}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{50} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{2}\right) b^{2} \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{50}^{2} \cdot k} f_{a.b}}} + 1 \right)^{-1} \right)^{-1} = 37.868 \, \frac{kg}{m^{3}}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{55} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{2}\right) b^{2} \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{55}^{2} \cdot k} f_{a.b}}} + 1 \right)^{-1} \right)^{-1} = 32.278 \, \frac{kg}{m^{3}}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{60} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{2}\right) b^{2} \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{60}^{2} \cdot k} f_{a.b}}} + 1 \right)^{-1} \right)^{-1} = 27.949 \, \frac{kg}{m^{3}}$$

$$\rho_{t0} := \frac{1}{a \cdot b \cdot d_{65} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{\,2}\right) b^2 \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{65}^{\;2} \cdot k} f_{a.b}} + 1} \right)^{-1} \right)^{-1} = 24.518 \; \frac{kg}{m^3}$$

$$d_{50} := 50 \; mm$$

$$\rho_{t050} := \frac{1}{a \cdot b \cdot d_{50} \cdot v} \left(1 - \left(\sqrt{\frac{1}{\frac{4\left(1-v_f^{\,2}\right) b^2 \cdot \rho_f}{g_{a.b} \cdot E_f \cdot d_{50}^{\;2} \cdot k} f_{a.b}} + 1} \right)^{-1} \right)^{-1} = 37.868 \; \frac{kg}{m^3}$$

$$t_{050} := \frac{1}{\frac{\frac{w_{total}}{q_{mn}}}{g_{a.b} \cdot k \cdot a \cdot b^3} + \frac{l}{a \cdot b \cdot d_{50} \cdot k}} \left(\sqrt[2]{\frac{\left(1-v_f^{\,2}\right)}{g_{a.b} \cdot E_f \cdot d_{50}^{\;2} \cdot a^2 \; \rho_f \cdot k} f_{a.b}} + \frac{2\left(1-v_f^{\,2}\right) b}{g_{a.b} \cdot E_f \cdot d_{50}^{\;2} \cdot a \cdot k} f_{a.b} \right) = 771.296 \; \mu m$$

$$t_{f50} := 1 \; mm \qquad \rho_{50} := 38 \; \frac{kg}{m^3}$$

$$m_{original} := a \cdot b \cdot 2 \cdot t_f \cdot \rho_f + a \cdot b \cdot t_c \cdot \rho_c = 48.791 \; kg$$

$$m_{new} := a \cdot b \cdot 2 \cdot t_{f50} \cdot \rho_f + a \cdot b \cdot \left(d_{50} - t_{f50}\right) \cdot \rho_{50} = 45.425 \; kg$$

$$m_{Improvement} := 100 - \frac{m_{new}}{m_{original}} \, 100 = 6.897 \quad \text{\% less mass}$$

$G_{cH35} := 12\ MPa \qquad t_{c50} := d_{50} - t_{f50} = 49\ mm$

$D_{50} := \dfrac{E_f \cdot t_{f50} \cdot d_{50}^{\ 2}}{2} \qquad S_{50} := \dfrac{G_{cH35} \cdot d_{50}^{\ 2}}{t_{c50}}$

$w_{b50} := \dfrac{q_{mn} \cdot \left(1 - v_f^{\ 2}\right)\ b^4}{D_{50}}\ f_{a.b} = 5.142\ mm$

$w_{s50} := \dfrac{q_{mn} \cdot b^2}{S_{50}}\ g_{a.b} = 2.398\ mm$

$w_{total50} := w_{s50} + w_{b50} = 7.54\ mm$

$w_{total} := w_s + w_b = 9.394\ mm$

$w_{Improvment} := 100 - \dfrac{w_{total50}}{w_{total}} \cdot 100 = 19.736$ % less deflection

Appendix B – Mechanical properties for Divinycell H [3].

Mechanical properties Divinycell® H

Property	Test Procedure	Unit		H35	H45	H60	H80	H100	H130	H200	H250
Compressive Strength[1]	ASTM D 1621	MPa	Nominal	0.5	0.6	0.9	1.4	2.0	3.0	5.4	7.2
			Minimum	0.3	0.5	0.7	1.15	1.65	2.4	4.5	6.1
Compressive Modulus[1]	ASTM D1621-B-73	MPa	Nominal	40	50	70	90	135	170	310	400
			Minimum	29	45	60	80	115	145	265	350
Tensile Strength[1]	ASTM D 1623	MPa	Nominal	1.0	1.4	1.8	2.5	3.5	4.8	7.1	9.2
			Minimum	0.8	1.1	1.5	2.2	2.5	3.5	6.3	8.0
Tensile Modulus[1]	ASTM D 1623	MPa	Nominal	49	55	75	95	130	175	250	320
			Minimum	37	45	57	85	105	135	210	260
Shear Strength	ASTM C 273	MPa	Nominal	0.4	0.56	0.76	1.15	1.6	2.2	3.5	4.5
			Minimum	0.3	0.46	0.63	0.95	1.4	1.9	3.2	3.9
Shear Modulus	ASTM C 273	MPa	Nominal	12	15	20	27	35	50	73	97
			Minimum	9	12	16	23	28	40	65	81
Shear Strain	ASTM C 273	%	Nominal	9	12	20	30	40	40	45	45
Density	ISO 845	kg/m³	Nominal	38	48	60	80	100	130	200	250

All values measured at +23°C

1. Properties measured perpendicular to the plane

Nominal value is an average value of a mechanical property at a nominal density
Minimum value is a minimum guaranteed mechanical property a material has independently of density

Divinycell H is type approved by:

CERTIFIED BY DNV·GL MARITIME

Technical Characteristics Divinycell® H

Characteristics[1]	Unit	H35	H45	H60	H80	H100	H130	H200	H250	Test method
Density variation	%	-10% to +20%	± 10	± 10	± 10	± 10	± 10	± 10	± 10	-
Thermal conductivity[2]	W/(m-K)	0.028	0.028	0.029	0.031	0.033	0.036	0.044	0.049	EN 12667
Coeff, linear heat expansion	$x10^{-6}/°C$	40	40	40	40	40	40	40	40	ISO 4897
Heat Distortion Temperature	°C	+125	+125	+125	+125	+125	+125	+125	+125	DIN 53424
Continous temp range	°C	-200 to +70	-200 to +70	-200 to +70	-200 to +70	-200 to +70	-200 to +70	-200 to +70	-200 to +70	-
Max process temp	°C	+90	+90	+90	+90	+110	+110	+110	+110	-
Dissipation factor	-	0.0001	0.0002	0.0003	0.0005	0.0006	0.0009	0.0015	0.0019	ASTM D 2520
Dielectric constant	-	1.04	1.05	1.06	1.09	1.11	1.15	1.23	1.29	ASTM D 2520
Poissons ratio[3]	-	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	D638-08

1. Typical values
2. Thermal conductivity at +20°C
3. Standard deviation is 0.045

Appendix C – Optimal stiffness Design of Sandwich Plates with Variable Core Densities.

Optimal stiffness design of sandwich plates with variable core densities

Dag Lukkassen, Annette Meidell, and Herman Myrvoll

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Optimal Stiffness Design of Sandwich Plates with Variable Core Densities

The paper is dedicated to professor Lars-Erik Persson, on the occasion of his 70th birthday

Dag Lukkassen*, Annette Meidell† and Herman Myrvoll**

*Narvik University College and
NORUT Narvik, Norway
†Narvik University College and
NORUT Narvik, Norway
**Narvik University College, Norway

Abstract. We consider optimal design of sandwich plates with variable core densities. Standard methods usually involve the numerical solution of complicated polynomial equations. Our method is much simpler and often leads to simple closed form expressions with even higher accuracy.

Keywords: Sandwich plates, core densities, optimal design
PACS: 87.10.-e, 87.10.Pq.

INTRODUCTION

The company TAM at Andslimoen, north in Norway, has for 33 years designed and produced many types of mobile military lightweight shelter, using a self-produced, professional, glued, self-supporting sandwich system. All delivered shelters are still in good condition except for a few ruined in accidents. The research which led to this paper was initiated for the purpose of investigating whether it is possible to reduce the weight of these shelters even more, without reducing the stiffness of the walls, roofs and floors. The most elementary model for analyzing the stiffness of each component of the shelter is to consider a simply supported sandwich plate with sides a and b subjected to a given vertical load $q(x,y)$ (see below). In this paper we consider a new method for minimizing the weight of such sandwich plates where the core density (and the corresponding shear modulus) can be chosen from a set of available core materials. Existing methods (see e.g. [1] and [2]) involve constraint nonlinear programming and the numerical solution of complicated polynomial equations. Our method is much simpler and reduces to simple closed form expressions with even higher accuracy. The method is inspired by the method described in [3].

BASIC FORMULAE

Let us consider a simply supported, isotropic sandwich plate with sides a and b subjected to a vertical load $q(x,y)$. We assume thin faces of thickness t with Young's modulus E and Poissons ratio ν and weak core with shear modulus G of thickness t_c. The deflection of the plate w is then the sum of deflection due to pure bending and pure shear deformation, $w = w_b + w_b$, where

$$w_b = \frac{(1-\nu_f^2)}{D}\sum_{n=1}^{\infty}\sum_{m=1}^{\infty}\frac{q_{mn}\sin\left(\frac{m\pi x}{a}\right)\sin\left(\frac{n\pi x}{b}\right)}{\left(\left(\frac{m\pi}{a}\right)^2+\left(\frac{n\pi}{b}\right)^2\right)^2} \text{ and } w_s = \frac{1}{S}\sum_{n=1}^{\infty}\sum_{m=1}^{\infty}\frac{q_{mn}\sin\left(\frac{m\pi x}{a}\right)\sin\left(\frac{n\pi x}{b}\right)}{\left(\frac{m\pi}{a}\right)^2+\left(\frac{n\pi}{b}\right)^2}.$$

Here,

$$D = \frac{Etd^2}{2},\ S = \frac{Gd^2}{t_c} \approx G_c d,\ d = t_c + t$$

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Optimization of sandwich plates

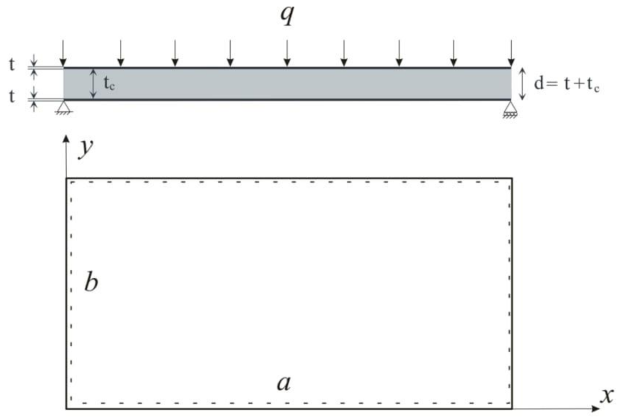

FIGURE 1. Rectangular simply supported sandwich-plate.

and q_{mn} are given by the loading conditions. In case of uniformly distributed load $q(x,y) = q$, where $q > 0$ is a constant (see Figure 1), it holds that

$$q_{mn} = \frac{16q}{mn\pi^2},$$

whenever m and n are odd, otherwise $q_{mn} = 0$ (see [1]). Hence, we may replace m by $2m+1$ and n by $2n+1$ in the above expressions and obtain that

$$w_b = \frac{16q(1-\nu^2)}{D} \sum_{n=0}^{\infty}\sum_{m=0}^{\infty} \frac{\sin\left(\frac{(2m+1)\pi x}{a}\right)\sin\left(\frac{(2n+1)\pi x}{b}\right)}{(2m+1)(2n+1)\left(\left(\frac{(2m+1)\pi}{a}\right)^2+\left(\frac{(2n+1)\pi}{b}\right)^2\right)^2},$$

$$w_s = \frac{16q}{S} \sum_{n=0}^{\infty}\sum_{m=0}^{\infty} \frac{\sin\left(\frac{(2m+1)\pi x}{a}\right)\sin\left(\frac{(2n+1)\pi x}{b}\right)}{(2m+1)(2n+1)\left(\left(\frac{(2m+1)\pi}{a}\right)^2+\left(\frac{(2n+1)\pi}{b}\right)^2\right)}.$$

The maximum deflections are in this case obtained at the midpoint $x = a/2$, $y = b/2$. At this point

$$w_b = \frac{q(1-\nu^2)b^4}{D} f(a/b) \text{ and } w_s = \frac{qb^2}{S} g(a/b),$$

where

$$f(a/b) = \sum_{n=0}^{\infty}\sum_{m=0}^{\infty} \frac{16\sin\left(\frac{(2m+1)\pi}{2}\right)\sin\left(\frac{(2n+1)\pi}{2}\right)}{\pi^6 (2m+1)(2n+1)\left(\left(\frac{(2m+1)}{a/b}\right)^2+((2n+1))^2\right)^2}$$

and

$$g(a/b) = \sum_{n=0}^{\infty}\sum_{m=0}^{\infty} \frac{16\sin\left(\frac{(2m+1)\pi}{2}\right)\sin\left(\frac{(2n+1)\pi}{2}\right)}{\pi^4 (2m+1)(2n+1)\left(\left(\frac{(2m+1)}{a/b}\right)^2+((2n+1))^2\right)}.$$

Optimization of sandwich plates

The stiffness of the plate (with respect to uniformly distributed loads) may be defined as the ratio w/q, where w is the maximum deflection, i.e.

$$w/q = \frac{(1-\nu^2)b^4}{D} f(a/b) + \frac{b^2}{S} g(a/b) = \frac{2(1-\nu^2)b^4}{Etd^2} f(a/b) + \frac{b^2}{Gd} g(a/b).$$

Hence,

$$G = \frac{g(a/b)b^2}{d\left((w/q) - \frac{2(1-\nu^2)b^4}{Etd^2} f(a/b)\right)}. \tag{1}$$

MODELLING THE SHEAR MODULUS

Let us now assume that the core material belongs to some specific class of cellular materials where the shear modulus G is uniquely determined by the density ρ, i.e. $G = G(\rho)$. We will also assume that $G(0) = 0$ and that $G(\cdot)$ is strictly increasing, continuous and piecewise differentiable. In particular, this means that its inverse function $\rho(G)$ exists. The total mass of the sandwich plate is given by

$$m = ab2t\rho_f + abd\rho(G).$$

Given the required stiffness $(w/q)^{-1}$, our main objective is to find the density $\rho \in [\rho_{\min}, \rho_{\max}]$ and the design parameters t and d which minimizes this mass. Here, $\rho_{\min}$ and $\rho_{\max}$ denote the minimal and maximal density of the core materials available in our class $\rho_{\max}$. One of the most crucial remarks in this paper is that it turns out that the shear stiffness $G(\rho)$ very often can be approximated by some rational function of the form

$$G(\rho) = \frac{1}{k\rho^{-1} - l}. \tag{2}$$

For example, according to DIAB (Divinycell), their H-type forms have the following relation between ρ and $G(\rho)$:

ρ [kg/m^3]	38	48	60	80	100	130	160
$G(\rho)$ [MPa]	12	15	20	27	35	50	73

The function of the type (2) which coincide with these table values at $\rho = 100$ and $\rho = 160$ is

$$G(\rho) = \frac{1}{\frac{6080}{1533}\rho^{-1} - \frac{17}{1533}}. \tag{3}$$

The graph of this function and the above table values are compared in Figure 2.

This is a substantial improvement compared with well known optimization methods. For example, the simplest known method is to approximate the shear stiffness by some power of the density, i.e. of the form

$$G(\rho) = k\rho^n \tag{4}$$

for some positive constants k and n. The approximation of this type which fits the above table values best possible, and also coincide with table value at the highest density $\rho = 160$, seems to be the following function:

$$G(\rho) = 73\left(\frac{\rho}{160}\right)^{3/2}. \tag{5}$$

The corresponding curve is illustrated in Figure 3. Note that in this case the rational approximation (3) is clearly much closer to the measured values and is therefore a better approximation than any power law formula.

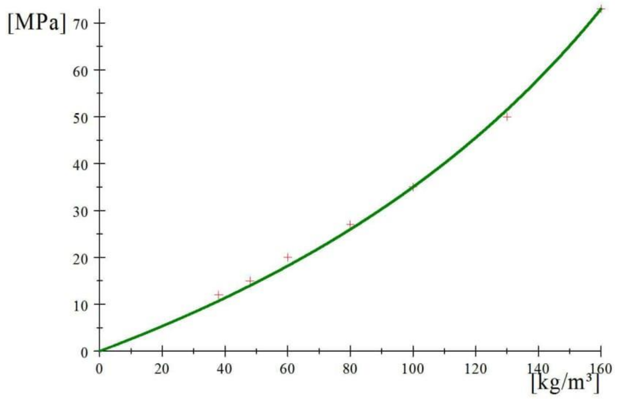

FIGURE 2. The shear modulus $G(\rho)$ as function of the density ρ.

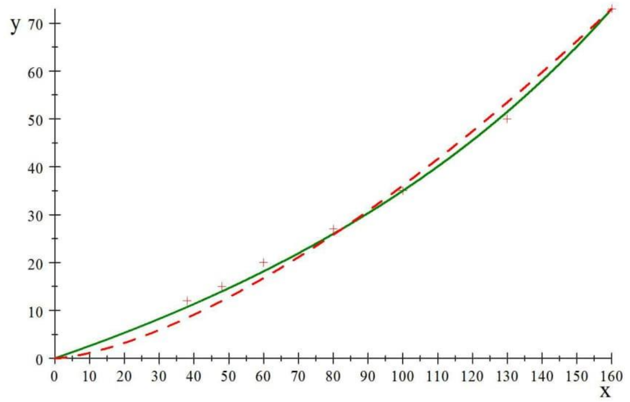

FIGURE 3. Comparison of the rational approximation (3) (solid curve) with the best possible power law approximation (5) (dashed curve).

OPTIMAL FACE THICKNESS

As demonstrated in this section, there is another property of the rational approximation (3) which is even more striking than its accuracy, namely that it is surprisingly suitable for implementation in optimization algorithms. By (1) and (2) we find that

$$\frac{1}{k\rho^{-1}-l} = \frac{g(a/b)b^2}{d\left((w/q) - \frac{2(1-\nu^2)b^4}{Etd^2} f(a/b)\right)}.$$

Hence,

$$\rho^{-1} = \frac{d\left((w/q) - \frac{2(1-\nu^2)b^4}{Etd^2} f(a/b)\right)}{g(a/b)b^2k} + l. \tag{6}$$

From this we see that the mass is given by

$$m = abt\rho_f + abd\rho =$$

$$abt\rho_f + abd\rho = abt\rho_f + abd\left(\frac{d\left((w/q) - \frac{2(1-\nu^2)b^4}{E_f td^2} f(a/b)\right)}{g(a/b)b^2k} + l\right)^{-1}.$$

Hence, for a given values of a, b, d and stiffness w/q, the mass can be expressed in the form

$$m(t) = rt + \left(v - \frac{c}{t}\right)^{-1}, \tag{7}$$

where

$$r = ab\rho_f,\ v = \frac{(w/q)}{g(a/b)kab^3} + \frac{l}{abd} \text{ and } c = \frac{2(1-\nu^2)b}{g(a/b)E_f d^2 ak} f(a/b),$$

which is a very suitable form since the derivative with respect to t is given by

$$m'(t) = r - \frac{c}{(tv-c)^2}.$$

Therefore, the extremal value $t = t_o$ is easy to find and is given by the simple expression,

$$t_o = \frac{1}{v}\left(\sqrt{\frac{c}{r}} + c\right) = \frac{1}{\frac{(w/q)}{g(a/b)kab^3} + \frac{l}{abd}}\left(\sqrt{\frac{\frac{2(1-\nu^2)b}{g(a/b)E_f d^2 ak} f(a/b)}{ab\rho_f}} + \frac{2(1-\nu^2)b}{g(a/b)E_f d^2 ak} f(a/b)\right) = \tag{8}$$

$$\frac{1}{\frac{(w/q)}{g(a/b)kab^3} + \frac{l}{abd}}\left(\sqrt{\frac{2(1-\nu^2)}{g(a/b)E_f d^2 a^2 \rho_f k} f(a/b)} + \frac{2(1-\nu^2)b}{g(a/b)E_f d^2 ak} f(a/b)\right)$$

This is very striking since a similar treatment with the use of the power law approximation (4) will lead to nontrivial polynomial equations. For example, the use of (5) gives a 6th order equation. The expression obtained from (8) is particularly useful if we also want to optimize the thickness of the sandwich d in the next step.

Note that our formula is based on the assumption of thin faces. The above formula for the optimal thickness is therefore only valid if $t_o \ll d$. From the above expressions we obtain that

$$\rho(t_o) = \frac{1}{abd}\left(v - \frac{c}{t_o}\right)^{-1} = \frac{1}{abd}\left(v - \frac{c}{\frac{1}{v}\left(\sqrt{\frac{c}{r}} + c\right)}\right)^{-1} = \frac{1}{abdv}\left(1 - \frac{1}{\left(\sqrt{\frac{1}{cr}} + 1\right)}\right)^{-1} =$$

$$\frac{1}{abdv}\left(1 - \left(\sqrt{\frac{1}{\frac{2(1-\nu^2)b^2\rho_f}{g(a/b)E_f d^2 k} f(a/b)}} + 1\right)^{-1}\right)^{-1}.$$

We have to make sure that the minimum value of $m(t)$ falls into the class of those values which are physically realizable i.e. such that

$$\rho_{\min} \le \rho(t) \le \rho_{\max}.$$

Based on this we obtain the following: The optimal value of $t = t_{opt}$ is given by

$$t_{opt} = \begin{cases} t_o & \text{if} \quad \rho_{\min} \le \rho(t_o) \le \rho_{\max} \\ t_- & \text{if} \quad \rho(t_o) < \rho_{\min} \\ t_+ & \text{if} \quad \rho_{\max} < \rho(t_o) \end{cases},$$

Optimization of sandwich plates

where t_- and t_+ are given by

$$w/q = \frac{2(1-\nu^2)b^4}{Ed^2 t_-} f(a/b) + \frac{b^2}{G(\rho_{\min})\,d} g(a/b),$$

and

$$w/q = \frac{2(1-\nu^2)b^4}{Ed^2 t_+} f(a/b) + \frac{b^2}{G(\rho_{\max})\,d} g(a/b).$$

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Appendix D – APDL log file

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MPDATA,PRXY,1,,0.33
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TBCOPY,ALL,1,2
MPTEMP,,,,,,,,
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MPDE,PRXY,2
MPDATA,EX,2,,33.6E+06
MPDATA,PRXY,2,,0.4
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secdata, 0.001,1,0.0,3
secdata, 0.049,2,0.0,3
secdata, 0.001,1,0.0,3
secoffset,MID
seccontrol,,,, , , ,
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K,2,2.602,0,0,
K,3,2.602,0,2.404,
K,4,0,0,2.404,
K,4,0,0,2.404,
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FITEM,2,2
FITEM,2,3
FITEM,2,4
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CMSEL,S,_Y
AMESH,_Y1
CMDELE,_Y
CMDELE,_Y1
CMDELE,_Y2
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/SOL
FINISH
/PREP7
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FITEM,2,-4
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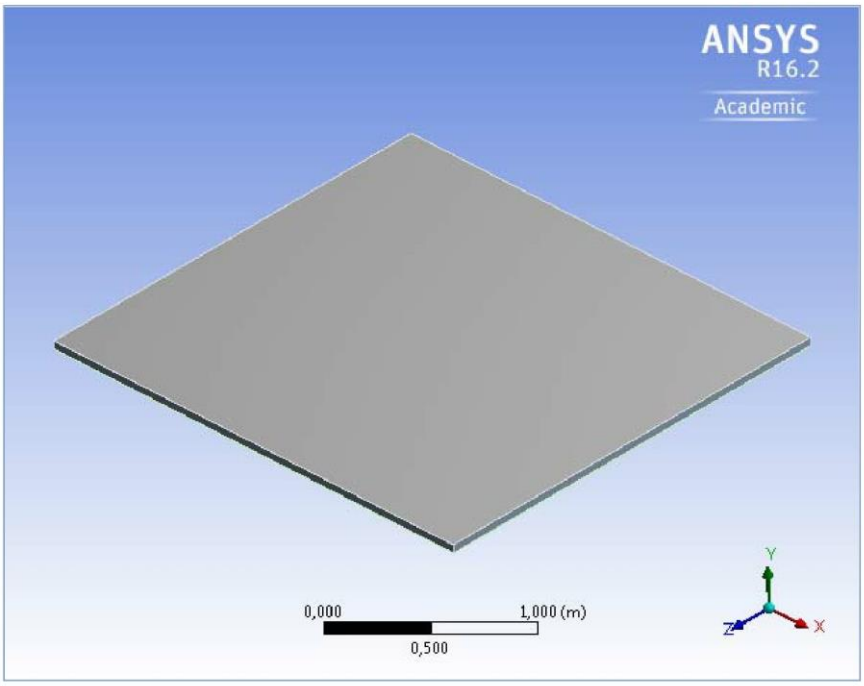
/GO
DL,P51X, ,UY,0
FLST,2,1,5,ORDE,1
FITEM,2,1
/GO
SFA,P51X,1,PRES,3197
FINISH
/SOL
SOLVE
FINISH
/POST1
```

Appendix E – ANSYS report – Simulation without frame

Project

First Saved	Tuesday, May 23, 2017
Last Saved	Tuesday, May 23, 2017
Product Version	16.2 Release
Save Project Before Solution	No
Save Project After Solution	No



Contents

Units

TABLE 1

Unit System	Metric (m, kg, N, s, V, A) Degrees rad/s Celsius
Angle	Degrees
Rotational Velocity	rad/s
Temperature	Celsius

Model (A4)

Geometry

TABLE 2
Model (A4) > Geometry

Object Name	*Geometry*
State	Fully Defined
Definition	
Source	D:\simulations\Master\sim_files\dp0\SYS\DM\SYS.agdb
Type	DesignModeler
Length Unit	Meters
Element Control	Program Controlled
Display Style	Body Color
Bounding Box	
Length X	2,602 m
Length Y	5,1e-002 m
Length Z	2,404 m
Properties	
Volume	0,31902 m³
Mass	42,36 kg
Scale Factor Value	1,
Statistics	

Bodies	3
Active Bodies	3
Nodes	4854
Elements	630
Mesh Metric	None
Basic Geometry Options	
Parameters	Yes
Parameter Key	DS
Attributes	No
Named Selections	No
Material Properties	No
Advanced Geometry Options	
Use Associativity	Yes
Coordinate Systems	No
Reader Mode Saves Updated File	No
Use Instances	Yes
Smart CAD Update	No
Compare Parts On Update	No
Attach File Via Temp File	Yes
Temporary Directory	C:\Users\bje012\AppData\Local\Temp
Analysis Type	3-D
Decompose Disjoint Geometry	Yes
Enclosure and Symmetry Processing	Yes

TABLE 3
Model (A4) > Geometry > Parts

Object Name	*face*	*core*	*face*
State	Meshed		
Graphics Properties			
Visible	Yes		
Transparency	1		
Definition			
Suppressed	No		
Stiffness Behavior	Flexible		
Coordinate System	Default Coordinate System		
Reference Temperature	By Environment		
Material			
Assignment	Aluminum Alloy	Divinycell H35	Aluminum Alloy
Nonlinear Effects	Yes		
Thermal Strain Effects	Yes		
Bounding Box			
Length X	2,602 m		
Length Y	1,e-003 m	4,9e-002 m	1,e-003 m
Length Z	2,404 m		
Properties			
Volume	6,2552e-003 m³	0,30651 m³	6,2552e-003 m³
Mass	16,889 kg	8,5821 kg	16,889 kg
Centroid X	0,8913 m		
Centroid Y	1,3491 m	1,3241 m	1,2991 m
Centroid Z	1,5642 m		
Moment of Inertia Ip1	8,1342 kg·m²	4,1349 kg·m²	8,1342 kg·m²
Moment of Inertia Ip2	17,663 kg·m²	8,9752 kg·m²	17,663 kg·m²
Moment of Inertia Ip3	9,5293 kg·m²	4,8438 kg·m²	9,5293 kg·m²
Statistics			
Nodes	1618		
Elements	210		

Mesh Metric	None

Coordinate Systems

TABLE 4
Model (A4) > Coordinate Systems > Coordinate System

Object Name	*Global Coordinate System*
State	Fully Defined
Definition	
Type	Cartesian
Coordinate System ID	0,
Origin	
Origin X	0, m
Origin Y	0, m
Origin Z	0, m
Directional Vectors	
X Axis Data	[1, 0, 0,]
Y Axis Data	[0, 1, 0,]
Z Axis Data	[0, 0, 1,]

Connections

TABLE 5
Model (A4) > Connections

Object Name	*Connections*
State	Fully Defined
Auto Detection	
Generate Automatic Connection On Refresh	Yes
Transparency	
Enabled	Yes

TABLE 6
Model (A4) > Connections > Contacts

Object Name	*Contacts*
State	Fully Defined
Definition	
Connection Type	Contact
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Auto Detection	
Tolerance Type	Slider
Tolerance Slider	0,
Tolerance Value	8,8573e-003 m
Use Range	No
Face/Face	Yes
Face/Edge	No
Edge/Edge	No
Priority	Include All
Group By	Bodies
Search Across	Bodies
Statistics	
Connections	2
Active Connections	2

TABLE 7

Model (A4) > Connections > Contacts > Contact Regions

Object Name	*Contact Region*	*Contact Region 2*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Contact	1 Face	
Target	1 Face	
Contact Bodies	face	core
Target Bodies	core	face
Definition		
Type	Bonded	
Scope Mode	Automatic	
Behavior	Program Controlled	
Trim Contact	Program Controlled	
Trim Tolerance	8,8573e-003 m	
Suppressed	No	
Advanced		
Formulation	Program Controlled	
Detection Method	Program Controlled	
Penetration Tolerance	Program Controlled	
Elastic Slip Tolerance	Program Controlled	
Normal Stiffness	Program Controlled	
Update Stiffness	Program Controlled	
Pinball Region	Program Controlled	
Geometric Modification		
Contact Geometry Correction	None	
Target Geometry Correction	None	

Mesh

TABLE 8
Model (A4) > Mesh

Object Name	*Mesh*
State	Solved
Display	
Display Style	Body Color
Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Use Advanced Size Function	Off
Relevance Center	Coarse
Element Size	Default
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge Length	1,e-003 m
Inflation	
Use Automatic Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0,272
Maximum Layers	5
Growth Rate	1,2
Inflation Algorithm	Pre
View Advanced Options	No

Patch Conforming Options	
Triangle Surface Mesher	Program Controlled
Patch Independent Options	
Topology Checking	No
Advanced	
Number of CPUs for Parallel Part Meshing	Program Controlled
Shape Checking	Standard Mechanical
Element Midside Nodes	Program Controlled
Straight Sided Elements	No
Number of Retries	Default (4)
Extra Retries For Assembly	Yes
Rigid Body Behavior	Dimensionally Reduced
Mesh Morphing	Disabled
Defeaturing	
Pinch Tolerance	Please Define
Generate Pinch on Refresh	No
Automatic Mesh Based Defeaturing	On
Defeaturing Tolerance	Default
Statistics	
Nodes	4854
Elements	630
Mesh Metric	None

Static Structural (A5)

TABLE 9
Model (A4) > Analysis

Object Name	*Static Structural (A5)*
State	Solved
Definition	
Physics Type	Structural
Analysis Type	Static Structural
Solver Target	Mechanical APDL
Options	
Environment Temperature	22, °C
Generate Input Only	No

TABLE 10
Model (A4) > Static Structural (A5) > Analysis Settings

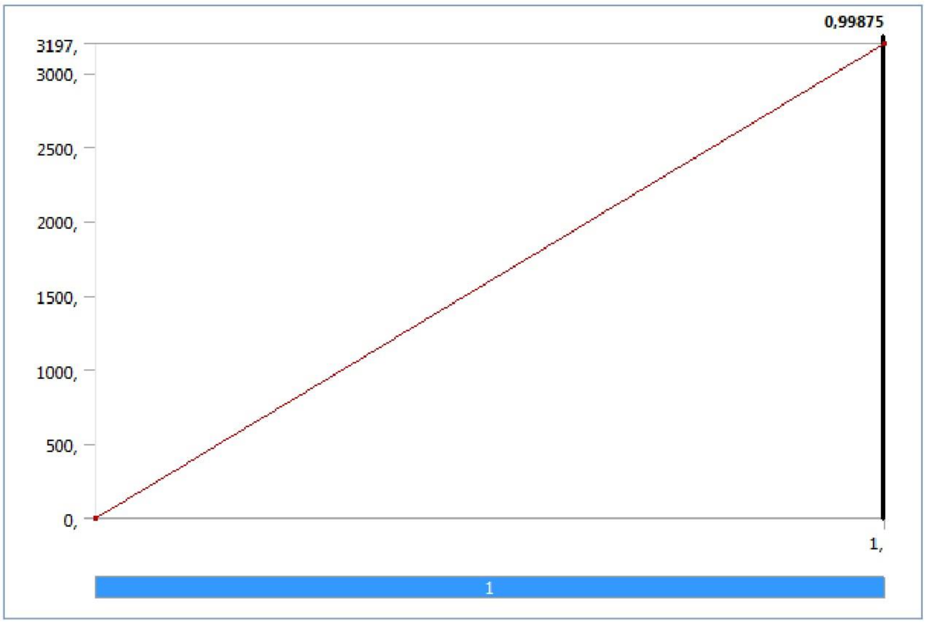
Object Name	*Analysis Settings*
State	Fully Defined
Step Controls	
Number Of Steps	1,
Current Step Number	1,
Step End Time	1, s
Auto Time Stepping	Program Controlled
Solver Controls	
Solver Type	Program Controlled
Weak Springs	Program Controlled
Solver Pivot Checking	Program Controlled
Large Deflection	On
Inertia Relief	Off
Restart Controls	
Generate Restart Points	Program Controlled
Retain Files After Full Solve	No
Nonlinear Controls	

Newton-Raphson Option	Program Controlled
Force Convergence	Program Controlled
Moment Convergence	Program Controlled
Displacement Convergence	Program Controlled
Rotation Convergence	Program Controlled
Line Search	Program Controlled
Stabilization	Off
Output Controls	
Stress	Yes
Strain	Yes
Nodal Forces	No
Contact Miscellaneous	No
General Miscellaneous	No
Store Results At	All Time Points
Analysis Data Management	
Solver Files Directory	D:\simulations\Master\sim_files\dp0\SYS\MECH\
Future Analysis	None
Scratch Solver Files Directory	
Save MAPDL db	No
Delete Unneeded Files	Yes
Nonlinear Solution	Yes
Solver Units	Active System
Solver Unit System	mks

TABLE 11
Model (A4) > Static Structural (A5) > Loads

Object Name	*Fixed Support*	*Pressure*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	12 Faces	1 Face
Definition		
Type	Fixed Support	Pressure
Suppressed	No	
Define By		Normal To
Magnitude		3197, Pa (ramped)

FIGURE 1
Model (A4) > Static Structural (A5) > Pressure



Solution (A6)

TABLE 12
Model (A4) > Static Structural (A5) > Solution

Object Name	*Solution (A6)*
State	Solved
Adaptive Mesh Refinement	
Max Refinement Loops	1,
Refinement Depth	2,
Information	
Status	Done
Post Processing	
Calculate Beam Section Results	No

TABLE 13
Model (A4) > Static Structural (A5) > Solution (A6) > Solution Information

Object Name	*Solution Information*
State	Solved
Solution Information	
Solution Output	Solver Output
Newton-Raphson Residuals	0
Update Interval	2,5 s
Display Points	All
FE Connection Visibility	
Activate Visibility	Yes
Display	All FE Connectors
Draw Connections Attached To	All Nodes
Line Color	Connection Type
Visible on Results	No
Line Thickness	Single
Display Type	Lines

TABLE 14
Model (A4) > Static Structural (A5) > Solution (A6) > Results

Object Name	*Directional Deformation*	*Total Deformation*
State	Solved	
Scope		
Scoping Method	Geometry Selection	
Geometry	All Bodies	
Definition		
Type	Directional Deformation	Total Deformation
Orientation	Y Axis	
By	Time	
Display Time	Last	
Coordinate System	Global Coordinate System	
Calculate Time History	Yes	
Identifier		
Suppressed	No	
Results		
Minimum	-2,0488e-002 m	0, m
Maximum	0, m	2,0488e-002 m
Minimum Occurs On	face	
Maximum Occurs On	face	
Information		
Time	1, s	
Load Step	1	
Substep	1	
Iteration Number	5	

FIGURE 2
Model (A4) > Static Structural (A5) > Solution (A6) > Directional Deformation

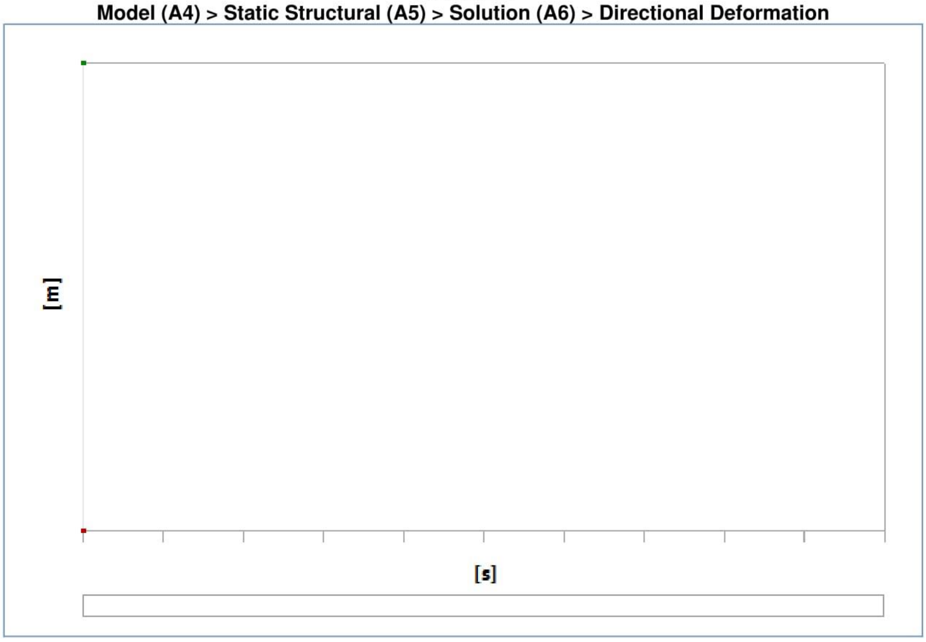


TABLE 15
Model (A4) > Static Structural (A5) > Solution (A6) > Directional Deformation

Time [s]	Minimum [m]	Maximum [m]

1,	-2,0488e-002	0,

FIGURE 3
Model (A4) > Static Structural (A5) > Solution (A6) > Total Deformation

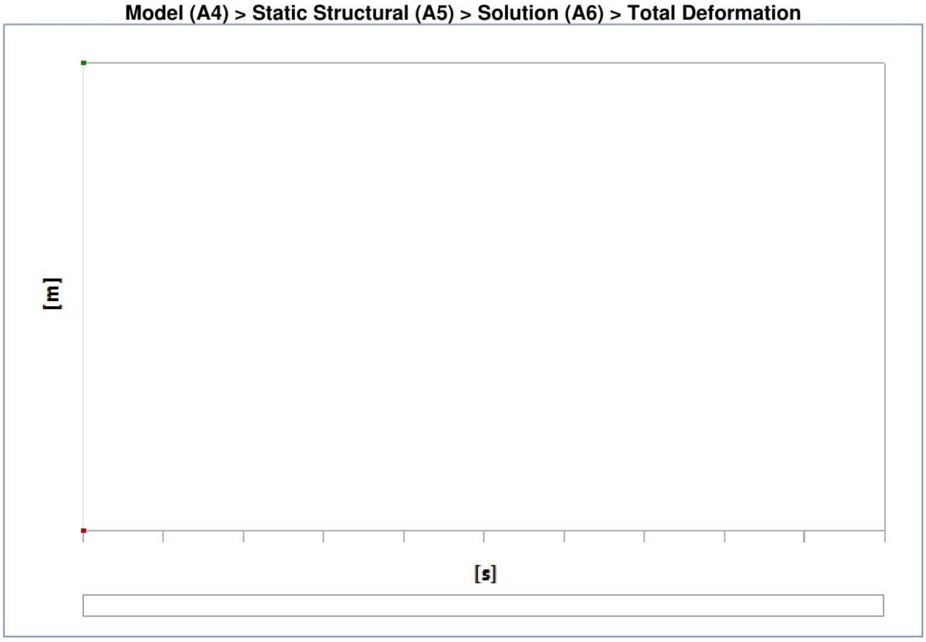


TABLE 16
Model (A4) > Static Structural (A5) > Solution (A6) > Total Deformation

Time [s]	Minimum [m]	Maximum [m]
1,	0,	2,0488e-002

Material Data

Aluminum Alloy

TABLE 17
Aluminum Alloy > Constants

Density	2700, kg m^-3
Coefficient of Thermal Expansion	2,3e-005 C^-1
Specific Heat	875, J kg^-1 C^-1

TABLE 18
Aluminum Alloy > Compressive Ultimate Strength

Compressive Ultimate Strength Pa
0,

TABLE 19
Aluminum Alloy > Compressive Yield Strength

Compressive Yield Strength Pa
2,8e+008

TABLE 20
Aluminum Alloy > Tensile Yield Strength

Tensile Yield Strength Pa

2,8e+008

TABLE 21
Aluminum Alloy > Tensile Ultimate Strength

Tensile Ultimate Strength Pa
3,1e+008

TABLE 22
Aluminum Alloy > Isotropic Secant Coefficient of Thermal Expansion

Reference Temperature C
22,

TABLE 23
Aluminum Alloy > Isotropic Thermal Conductivity

Thermal Conductivity W m^-1 C^-1	Temperature C
114,	-100,
144,	0,
165,	100,
175,	200,

TABLE 24
Aluminum Alloy > Alternating Stress R-Ratio

Alternating Stress Pa	Cycles	R-Ratio
2,758e+008	1700,	-1,
2,413e+008	5000,	-1,
2,068e+008	34000	-1,
1,724e+008	1,4e+005	-1,
1,379e+008	8,e+005	-1,
1,172e+008	2,4e+006	-1,
8,963e+007	5,5e+007	-1,
8,274e+007	1,e+008	-1,
1,706e+008	50000	-0,5
1,396e+008	3,5e+005	-0,5
1,086e+008	3,7e+006	-0,5
8,791e+007	1,4e+007	-0,5
7,757e+007	5,e+007	-0,5
7,239e+007	1,e+008	-0,5
1,448e+008	50000	0,
1,207e+008	1,9e+005	0,
1,034e+008	1,3e+006	0,
9,308e+007	4,4e+006	0,
8,618e+007	1,2e+007	0,
7,239e+007	1,e+008	0,
7,412e+007	3,e+005	0,5
7,067e+007	1,5e+006	0,5
6,636e+007	1,2e+007	0,5
6,205e+007	1,e+008	0,5

TABLE 25
Aluminum Alloy > Isotropic Resistivity

Resistivity ohm m	Temperature C
2,43e-008	0,
2,67e-008	20,
3,63e-008	100,

TABLE 26
Aluminum Alloy > Isotropic Elasticity

Temperature C	Young's Modulus Pa	Poisson's Ratio	Bulk Modulus Pa	Shear Modulus Pa
	7,e+010	0,33	6,8627e+010	2,6316e+010

TABLE 27
Aluminum Alloy > Isotropic Relative Permeability

Relative Permeability
1,

Divinycell H35

TABLE 28
Divinycell H35 > Constants

Density	28, kg m^-3
Coefficient of Thermal Expansion	2,3e-004 C^-1
Specific Heat	296, J kg^-1 C^-1
Thermal Conductivity	0,28 W m^-1 C^-1

TABLE 29
Divinycell H35 > Compressive Ultimate Strength

Compressive Ultimate Strength Pa
5,e+005

TABLE 30
Divinycell H35 > Compressive Yield Strength

Compressive Yield Strength Pa
1,e+006

TABLE 31
Divinycell H35 > Tensile Yield Strength

Tensile Yield Strength Pa
4,9e+007

TABLE 32
Divinycell H35 > Tensile Ultimate Strength

Tensile Ultimate Strength Pa
4,e+007

TABLE 33
Divinycell H35 > Isotropic Secant Coefficient of Thermal Expansion

Reference Temperature C
22,

TABLE 34
Divinycell H35 > Isotropic Elasticity

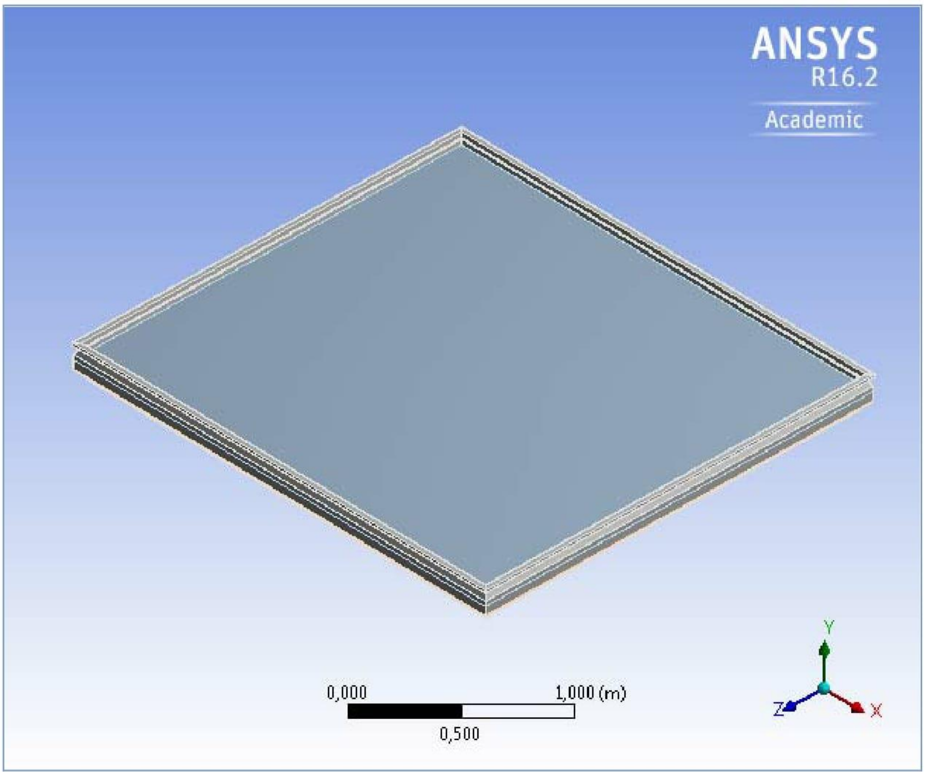
Temperature C	Young's Modulus Pa	Poisson's Ratio	Bulk Modulus Pa	Shear Modulus Pa
	3,36e+007	0,4	5,6e+007	1,2e+007

Appendix F - ANSYS report – Simulation with frame

ANSYS

Project

First Saved	Tuesday, May 23, 2017
Last Saved	Wednesday, May 24, 2017
Product Version	16.2 Release
Save Project Before Solution	No
Save Project After Solution	No



Contents

Units

TABLE 1

Unit System	Metric (m, kg, N, s, V, A) Degrees rad/s Celsius
Angle	Degrees
Rotational Velocity	rad/s
Temperature	Celsius

Model (B4)

Geometry

TABLE 2
Model (B4) > Geometry

Object Name	*Geometry*
State	Fully Defined
Definition	
Source	D:\simulations\Master\sim_files\dp0\SYS-1\DM\SYS-1.agdb
Type	DesignModeler
Length Unit	Meters
Element Control	Program Controlled
Display Style	Body Color
Bounding Box	
Length X	2,6137 m
Length Y	0,19815 m
Length Z	2,4157 m
Properties	
Volume	0,32843 m³
Mass	67,787 kg
Scale Factor Value	1,
Statistics	

Bodies	4
Active Bodies	4
Nodes	15731
Elements	5880
Mesh Metric	None
Basic Geometry Options	
Parameters	Yes
Parameter Key	DS
Attributes	No
Named Selections	No
Material Properties	No
Advanced Geometry Options	
Use Associativity	Yes
Coordinate Systems	No
Reader Mode Saves Updated File	No
Use Instances	Yes
Smart CAD Update	No
Compare Parts On Update	No
Attach File Via Temp File	Yes
Temporary Directory	C:\Users\bje012\AppData\Local\Temp
Analysis Type	3-D
Decompose Disjoint Geometry	Yes
Enclosure and Symmetry Processing	Yes

TABLE 3
Model (B4) > Geometry > Parts

Object Name	*flens*	*face*	*core*	*face*
State	Meshed			
Graphics Properties				
Visible	Yes			
Transparency	1			
Definition				
Suppressed	No			
Stiffness Behavior	Flexible			
Coordinate System	Default Coordinate System			
Reference Temperature	By Environment			
Material				
Assignment	Aluminum Alloy		Divinycell H35	Aluminum Alloy
Nonlinear Effects	Yes			
Thermal Strain Effects	Yes			
Bounding Box				
Length X	2,6137 m	2,602 m		
Length Y	0,19815 m	1,e-003 m	4,9e-002 m	1,e-003 m
Length Z	2,4157 m	2,404 m		
Properties				
Volume	9,4174e-003 m³	6,2552e-003 m³	0,30651 m³	6,2552e-003 m³
Mass	25,427 kg	16,889 kg	8,5821 kg	16,889 kg
Centroid X	0,89131 m	0,8913 m		
Centroid Y	1,3454 m	1,3491 m	1,3241 m	1,2991 m
Centroid Z	1,5642 m			
Moment of Inertia Ip1	24,377 kg·m²	8,1342 kg·m²	4,1349 kg·m²	8,1342 kg·m²
Moment of Inertia Ip2	51,638 kg·m²	17,663 kg·m²	8,9752 kg·m²	17,663 kg·m²
Moment of Inertia Ip3	27,482 kg·m²	9,5293 kg·m²	4,8438 kg·m²	9,5293 kg·m²
Statistics				
Nodes	10877	1618		
Elements	5250	210		

Mesh Metric	None

Coordinate Systems

TABLE 4
Model (B4) > Coordinate Systems > Coordinate System

Object Name	*Global Coordinate System*
State	Fully Defined
Definition	
Type	Cartesian
Coordinate System ID	0,
Origin	
Origin X	0, m
Origin Y	0, m
Origin Z	0, m
Directional Vectors	
X Axis Data	[1, 0, 0,]
Y Axis Data	[0, 1, 0,]
Z Axis Data	[0, 0, 1,]

Connections

TABLE 5
Model (B4) > Connections

Object Name	*Connections*
State	Fully Defined
Auto Detection	
Generate Automatic Connection On Refresh	Yes
Transparency	
Enabled	Yes

TABLE 6
Model (B4) > Connections > Contacts

Object Name	*Contacts*
State	Fully Defined
Definition	
Connection Type	Contact
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Auto Detection	
Tolerance Type	Slider
Tolerance Slider	0,
Tolerance Value	8,9113e-003 m
Use Range	No
Face/Face	Yes
Face/Edge	No
Edge/Edge	No
Priority	Include All
Group By	Bodies
Search Across	Bodies
Statistics	
Connections	5
Active Connections	5

TABLE 7

Model (B4) > Connections > Contacts > Contact Regions

Object Name	*Contact Region*	*Contact Region 2*	*Contact Region 3*	*Contact Region 4*	*Contact Region 5*
State	Fully Defined				
Scope					
Scoping Method	Geometry Selection				
Contact	4 Faces	8 Faces	12 Faces	1 Face	
Target	4 Faces	5 Faces	6 Faces	1 Face	
Contact Bodies	flens			face	core
Target Bodies	face	core	face	core	face
Definition					
Type	Bonded				
Scope Mode	Automatic				
Behavior	Program Controlled				
Trim Contact	Program Controlled				
Trim Tolerance	8,9113e-003 m				
Suppressed	No				
Advanced					
Formulation	Program Controlled				
Detection Method	Program Controlled				
Penetration Tolerance	Program Controlled				
Elastic Slip Tolerance	Program Controlled				
Normal Stiffness	Program Controlled				
Update Stiffness	Program Controlled				
Pinball Region	Program Controlled				
Geometric Modification					
Contact Geometry Correction	None				
Target Geometry Correction	None				

Mesh

TABLE 8
Model (B4) > Mesh

Object Name	*Mesh*
State	Solved
Display	
Display Style	Body Color
Defaults	
Physics Preference	Mechanical
Relevance	0
Sizing	
Use Advanced Size Function	Off
Relevance Center	Coarse
Element Size	Default
Initial Size Seed	Active Assembly
Smoothing	Medium
Transition	Fast
Span Angle Center	Coarse
Minimum Edge Length	1,e-003 m
Inflation	
Use Automatic Inflation	None
Inflation Option	Smooth Transition
Transition Ratio	0,272
Maximum Layers	5
Growth Rate	1,2

Inflation Algorithm	Pre
View Advanced Options	No
Patch Conforming Options	
Triangle Surface Mesher	Program Controlled
Patch Independent Options	
Topology Checking	No
Advanced	
Number of CPUs for Parallel Part Meshing	Program Controlled
Shape Checking	Standard Mechanical
Element Midside Nodes	Program Controlled
Straight Sided Elements	No
Number of Retries	Default (4)
Extra Retries For Assembly	Yes
Rigid Body Behavior	Dimensionally Reduced
Mesh Morphing	Disabled
Defeaturing	
Pinch Tolerance	Please Define
Generate Pinch on Refresh	No
Automatic Mesh Based Defeaturing	On
Defeaturing Tolerance	Default
Statistics	
Nodes	15731
Elements	5880
Mesh Metric	None

Static Structural (B5)

TABLE 9
Model (B4) > Analysis

Object Name	*Static Structural (B5)*
State	Solved
Definition	
Physics Type	Structural
Analysis Type	Static Structural
Solver Target	Mechanical APDL
Options	
Environment Temperature	22, °C
Generate Input Only	No

TABLE 10
Model (B4) > Static Structural (B5) > Analysis Settings

Object Name	*Analysis Settings*
State	Fully Defined
Step Controls	
Number Of Steps	1,
Current Step Number	1,
Step End Time	1, s
Auto Time Stepping	Program Controlled
Solver Controls	
Solver Type	Program Controlled
Weak Springs	Program Controlled
Solver Pivot Checking	Program Controlled
Large Deflection	Off
Inertia Relief	Off
Restart Controls	
Generate Restart Points	Program Controlled

Retain Files After Full Solve	No
Nonlinear Controls	
Newton-Raphson Option	Program Controlled
Force Convergence	Program Controlled
Moment Convergence	Program Controlled
Displacement Convergence	Program Controlled
Rotation Convergence	Program Controlled
Line Search	Program Controlled
Stabilization	Off
Output Controls	
Stress	Yes
Strain	Yes
Nodal Forces	No
Contact Miscellaneous	No
General Miscellaneous	No
Store Results At	All Time Points
Analysis Data Management	
Solver Files Directory	D:\simulations\Master\sim_files\dp0\SYS-1\MECH\
Future Analysis	None
Scratch Solver Files Directory	
Save MAPDL db	No
Delete Unneeded Files	Yes
Nonlinear Solution	No
Solver Units	Active System
Solver Unit System	mks

TABLE 11
Model (B4) > Static Structural (B5) > Loads

Object Name	*Pressure*	*Displacement*	*Displacement 2*
State	Fully Defined		
Scope			
Scoping Method	Geometry Selection		
Geometry	1 Face	3 Edges	1 Edge
Definition			
Type	Pressure	Displacement	
Define By	Normal To	Components	
Magnitude	3197, Pa (ramped)		
Suppressed	No		
Coordinate System		Global Coordinate System	
X Component		Free	0, m (ramped)
Y Component		0, m (ramped)	
Z Component		Free	0, m (ramped)

FIGURE 1
Model (B4) > Static Structural (B5) > Pressure

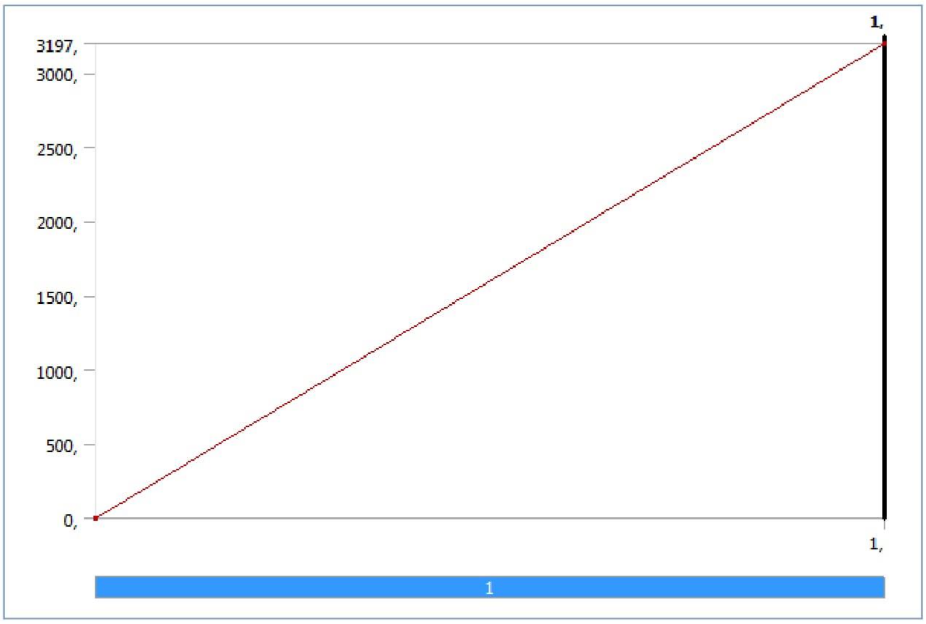


FIGURE 2
Model (B4) > Static Structural (B5) > Displacement

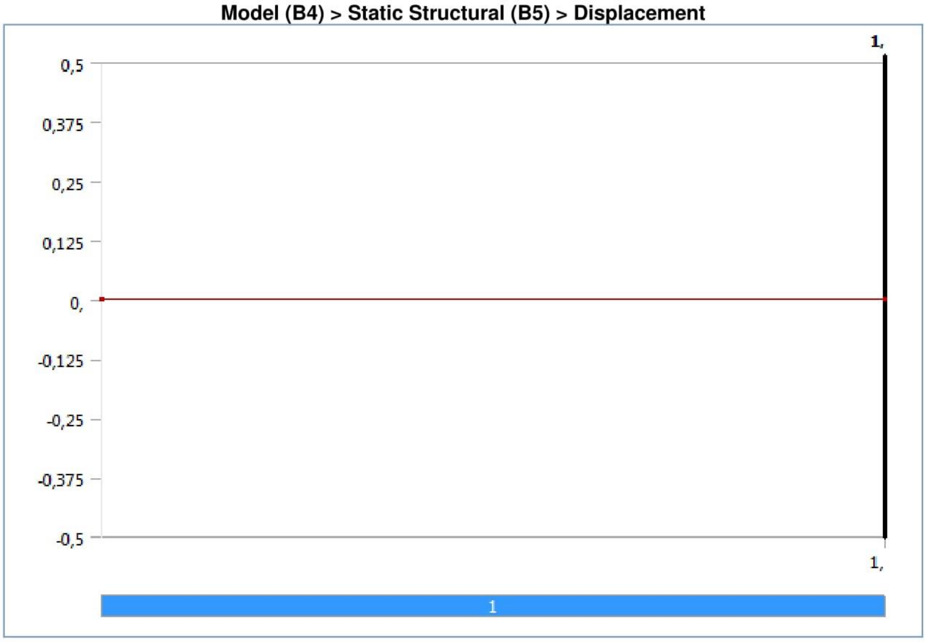
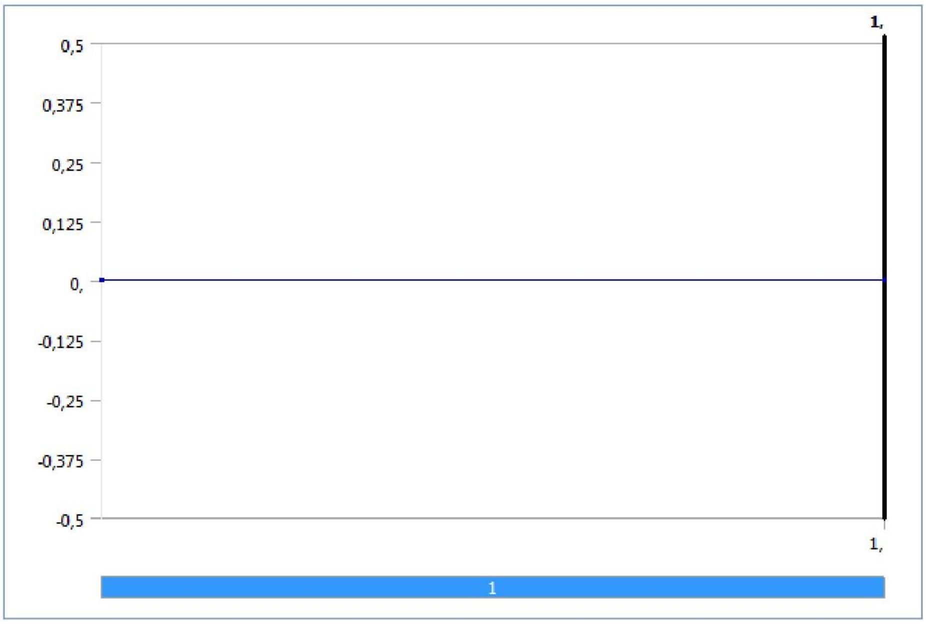


FIGURE 3
Model (B4) > Static Structural (B5) > Displacement 2



Solution (B6)

TABLE 12
Model (B4) > Static Structural (B5) > Solution

Object Name	*Solution (B6)*
State	Solved
Adaptive Mesh Refinement	
Max Refinement Loops	1,
Refinement Depth	2,
Information	
Status	Done
Post Processing	
Calculate Beam Section Results	No

TABLE 13
Model (B4) > Static Structural (B5) > Solution (B6) > Solution Information

Object Name	*Solution Information*
State	Solved
Solution Information	
Solution Output	Solver Output
Newton-Raphson Residuals	0
Update Interval	2,5 s
Display Points	All
FE Connection Visibility	
Activate Visibility	Yes
Display	All FE Connectors
Draw Connections Attached To	All Nodes
Line Color	Connection Type
Visible on Results	No
Line Thickness	Single
Display Type	Lines

TABLE 14
Model (B4) > Static Structural (B5) > Solution (B6) > Results

Object Name	*Total Deformation*	*Directional Deformation*
State	Solved	
Scope		
Scoping Method	Geometry Selection	
Geometry	All Bodies	
Definition		
Type	Total Deformation	Directional Deformation
By	Time	
Display Time	Last	
Calculate Time History	Yes	
Identifier		
Suppressed	No	
Orientation		Y Axis
Coordinate System		Global Coordinate System
Results		
Minimum	0, m	-5,4089e-003 m
Maximum	5,6071e-003 m	9,6503e-006 m
Minimum Occurs On	flens	core
Maximum Occurs On	core	flens
Information		
Time	1, s	
Load Step	1	
Substep	1	
Iteration Number	1	

FIGURE 4
Model (B4) > Static Structural (B5) > Solution (B6) > Total Deformation

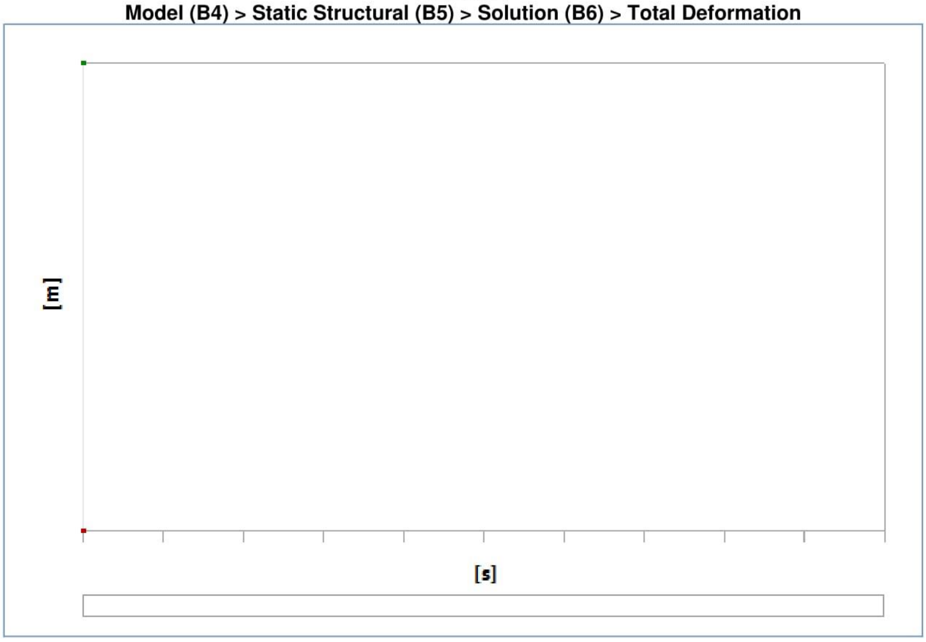


TABLE 15
Model (B4) > Static Structural (B5) > Solution (B6) > Total Deformation

Time [s]	Minimum [m]	Maximum [m]

1,	0,	5,6071e-003

FIGURE 5
Model (B4) > Static Structural (B5) > Solution (B6) > Directional Deformation

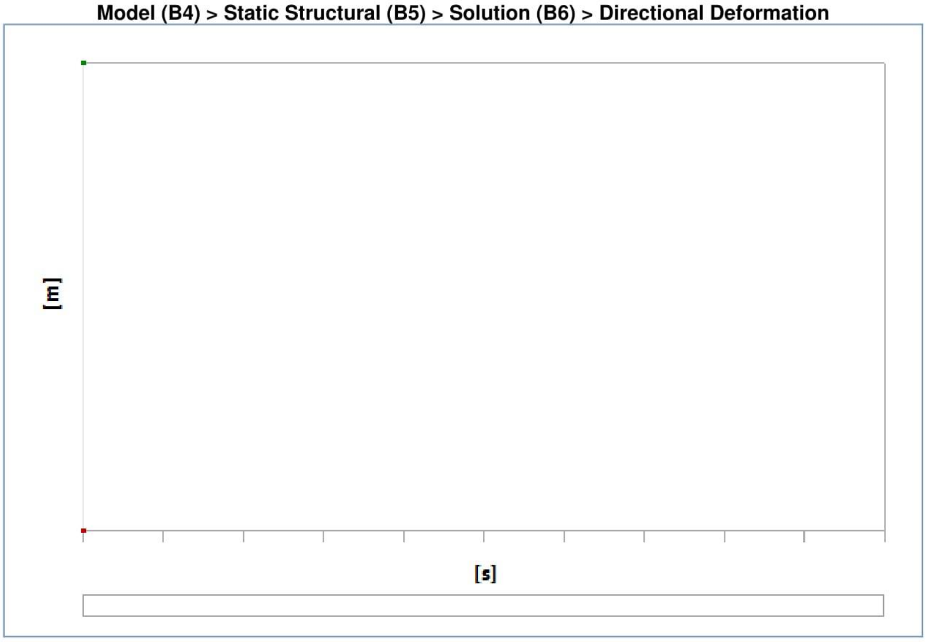


TABLE 16
Model (B4) > Static Structural (B5) > Solution (B6) > Directional Deformation

Time [s]	Minimum [m]	Maximum [m]
1,	-5,4089e-003	9,6503e-006

Material Data

Aluminum Alloy

TABLE 17
Aluminum Alloy > Constants

Density	2700, kg m^-3
Coefficient of Thermal Expansion	2,3e-005 C^-1
Specific Heat	875, J kg^-1 C^-1

TABLE 18
Aluminum Alloy > Compressive Ultimate Strength

Compressive Ultimate Strength Pa
0,

TABLE 19
Aluminum Alloy > Compressive Yield Strength

Compressive Yield Strength Pa
2,8e+008

TABLE 20
Aluminum Alloy > Tensile Yield Strength

Tensile Yield Strength Pa

2,8e+008

TABLE 21
Aluminum Alloy > Tensile Ultimate Strength

Tensile Ultimate Strength Pa
3,1e+008

TABLE 22
Aluminum Alloy > Isotropic Secant Coefficient of Thermal Expansion

Reference Temperature C
22,

TABLE 23
Aluminum Alloy > Isotropic Thermal Conductivity

Thermal Conductivity W m^-1 C^-1	Temperature C
114,	-100,
144,	0,
165,	100,
175,	200,

TABLE 24
Aluminum Alloy > Alternating Stress R-Ratio

Alternating Stress Pa	Cycles	R-Ratio
2,758e+008	1700,	-1,
2,413e+008	5000,	-1,
2,068e+008	34000	-1,
1,724e+008	1,4e+005	-1,
1,379e+008	8,e+005	-1,
1,172e+008	2,4e+006	-1,
8,963e+007	5,5e+007	-1,
8,274e+007	1,e+008	-1,
1,706e+008	50000	-0,5
1,396e+008	3,5e+005	-0,5
1,086e+008	3,7e+006	-0,5
8,791e+007	1,4e+007	-0,5
7,757e+007	5,e+007	-0,5
7,239e+007	1,e+008	-0,5
1,448e+008	50000	0,
1,207e+008	1,9e+005	0,
1,034e+008	1,3e+006	0,
9,308e+007	4,4e+006	0,
8,618e+007	1,2e+007	0,
7,239e+007	1,e+008	0,
7,412e+007	3,e+005	0,5
7,067e+007	1,5e+006	0,5
6,636e+007	1,2e+007	0,5
6,205e+007	1,e+008	0,5

TABLE 25
Aluminum Alloy > Isotropic Resistivity

Resistivity ohm m	Temperature C
2,43e-008	0,
2,67e-008	20,
3,63e-008	100,

TABLE 26
Aluminum Alloy > Isotropic Elasticity

Temperature C	Young's Modulus Pa	Poisson's Ratio	Bulk Modulus Pa	Shear Modulus Pa
	7,e+010	0,33	6,8627e+010	2,6316e+010

TABLE 27
Aluminum Alloy > Isotropic Relative Permeability

Relative Permeability
1,

Divinycell H35

TABLE 28
Divinycell H35 > Constants

Density	28, kg m^-3
Coefficient of Thermal Expansion	2,3e-004 C^-1
Specific Heat	296, J kg^-1 C^-1
Thermal Conductivity	0,28 W m^-1 C^-1

TABLE 29
Divinycell H35 > Compressive Ultimate Strength

Compressive Ultimate Strength Pa
5,e+005

TABLE 30
Divinycell H35 > Compressive Yield Strength

Compressive Yield Strength Pa
1,e+006

TABLE 31
Divinycell H35 > Tensile Yield Strength

Tensile Yield Strength Pa
4,9e+007

TABLE 32
Divinycell H35 > Tensile Ultimate Strength

Tensile Ultimate Strength Pa
4,e+007

TABLE 33
Divinycell H35 > Isotropic Secant Coefficient of Thermal Expansion

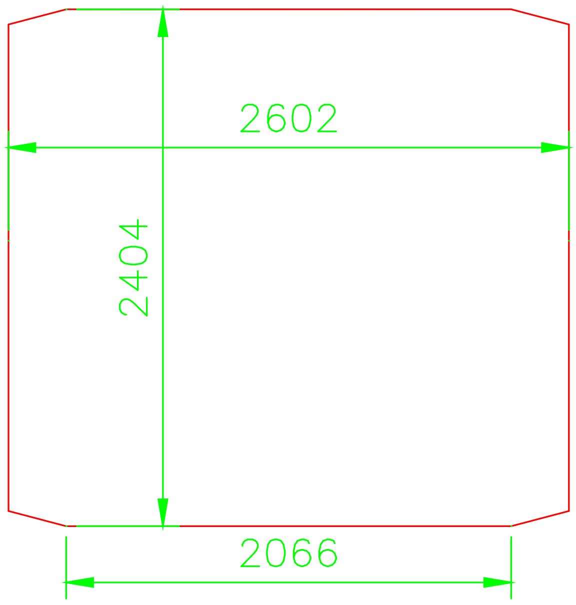
Reference Temperature C
22,

TABLE 34
Divinycell H35 > Isotropic Elasticity

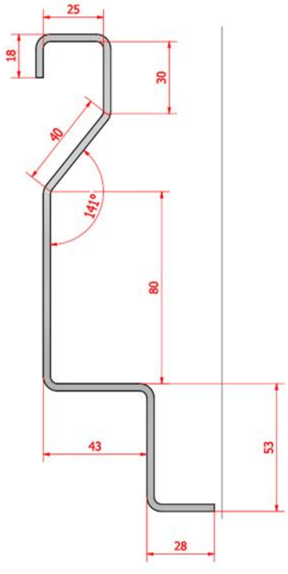
Temperature C	Young's Modulus Pa	Poisson's Ratio	Bulk Modulus Pa	Shear Modulus Pa
	3,36e+007	0,4	5,6e+007	1,2e+007

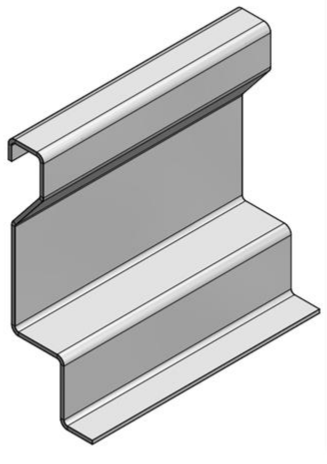
Appendix G – Specifications of sandwich panel from TAM

The panel is 2602mm x 2404mm and can be regarded as rectangular.



The panel is today constructed with a 1mm aluminum plate in the bottom that rests on an aluminum frame. The resting point is where the 80mm and 53mm measurements meets. The core is a 40mm H60 divinycell from DIAB AS [3] and the top is a 3mm aluminum plate.



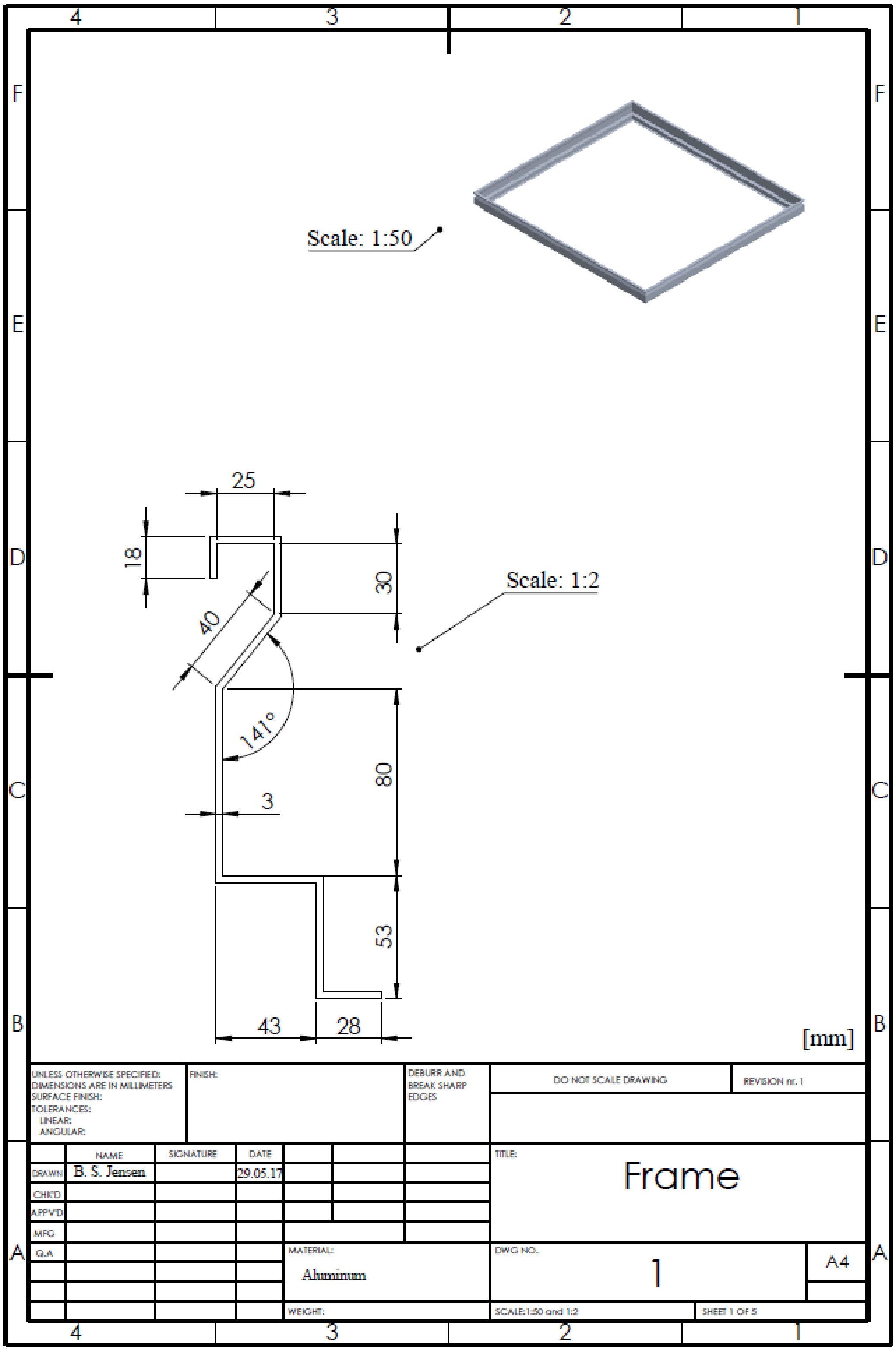


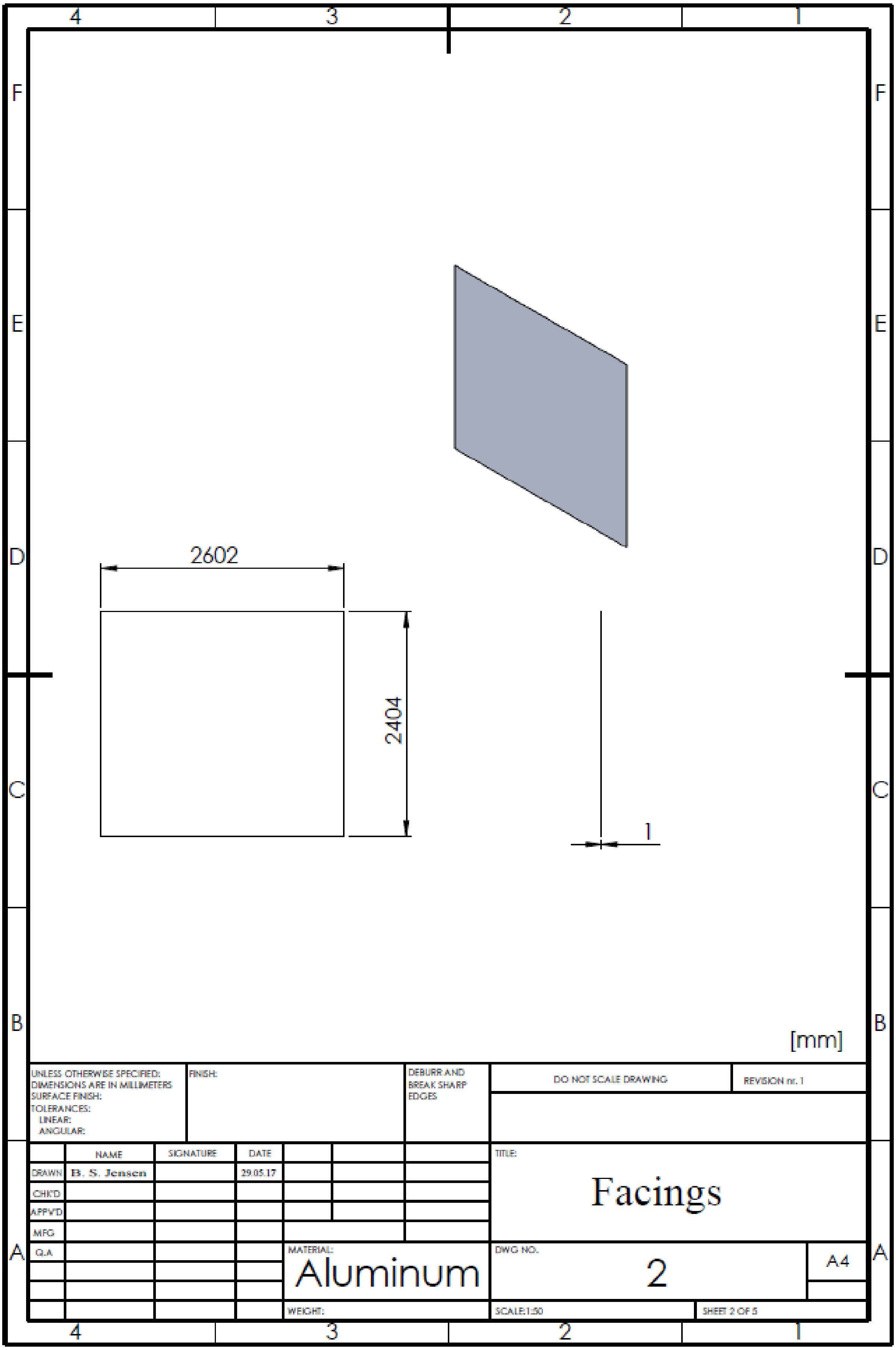
The panel is suspended from the corners by wires and shall withstand an uniformly distributed load of 20.000N.

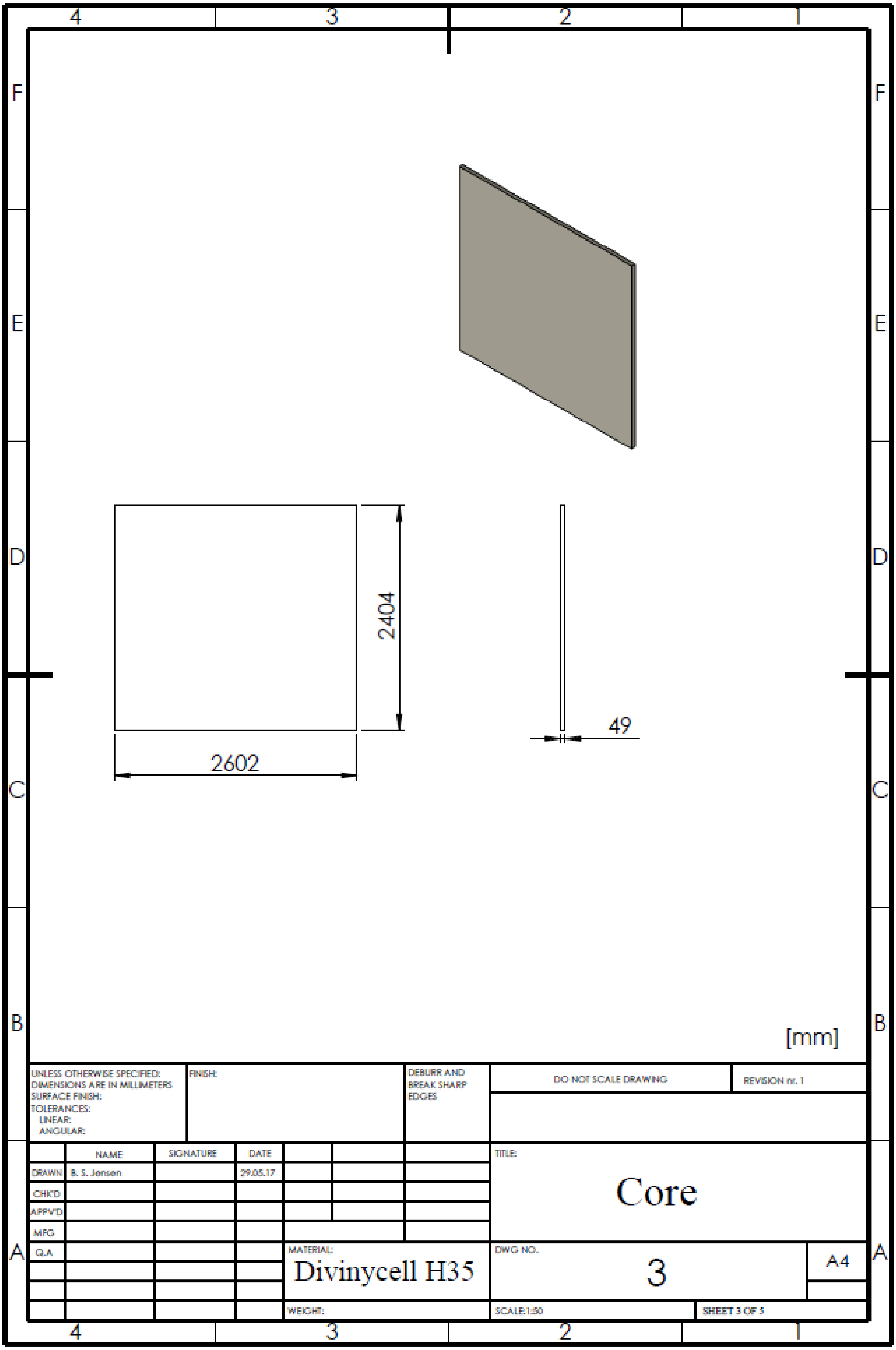
Text is translated by author of this report from Norwegian.

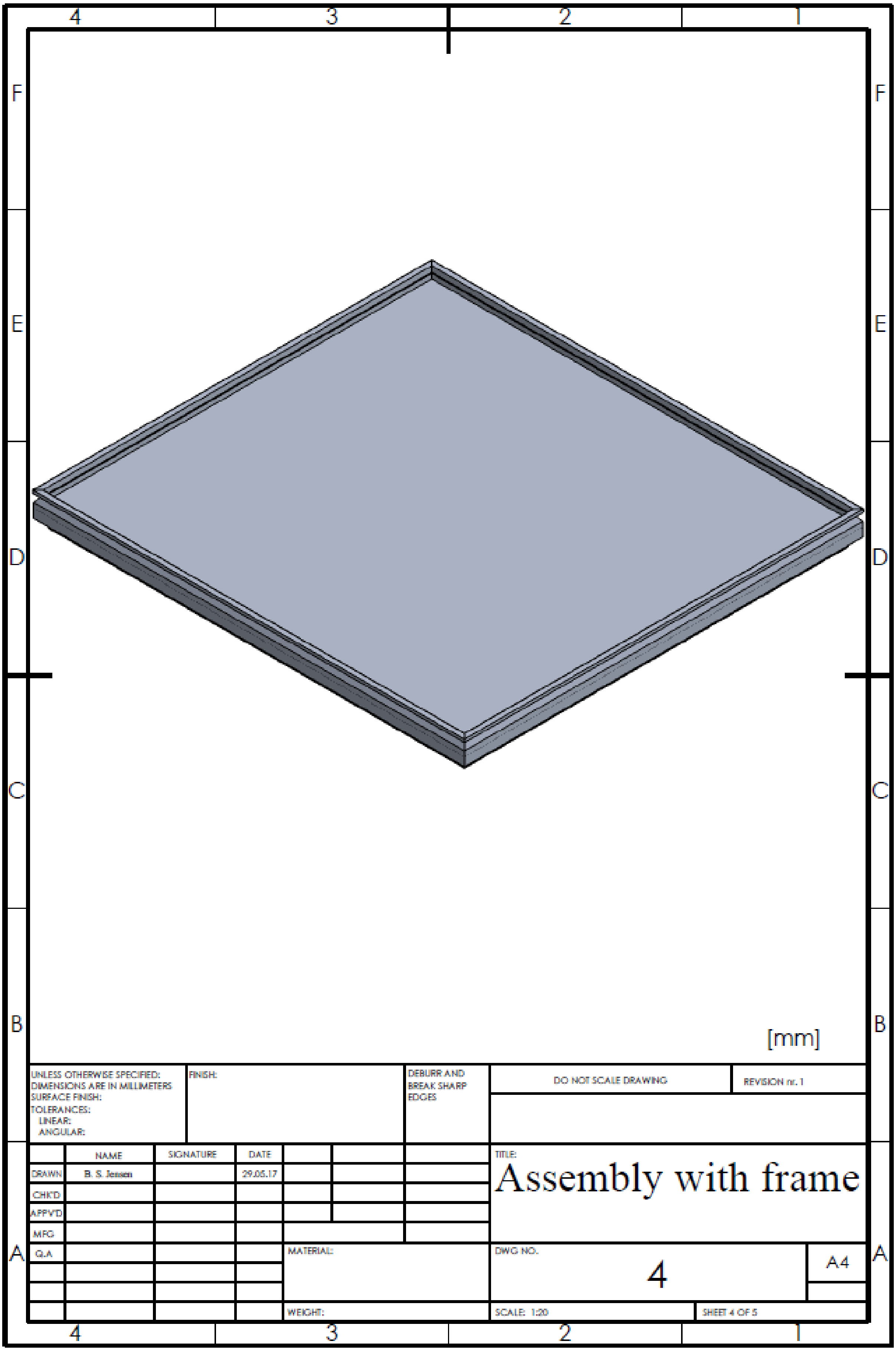
Appendix H – CAD drawings

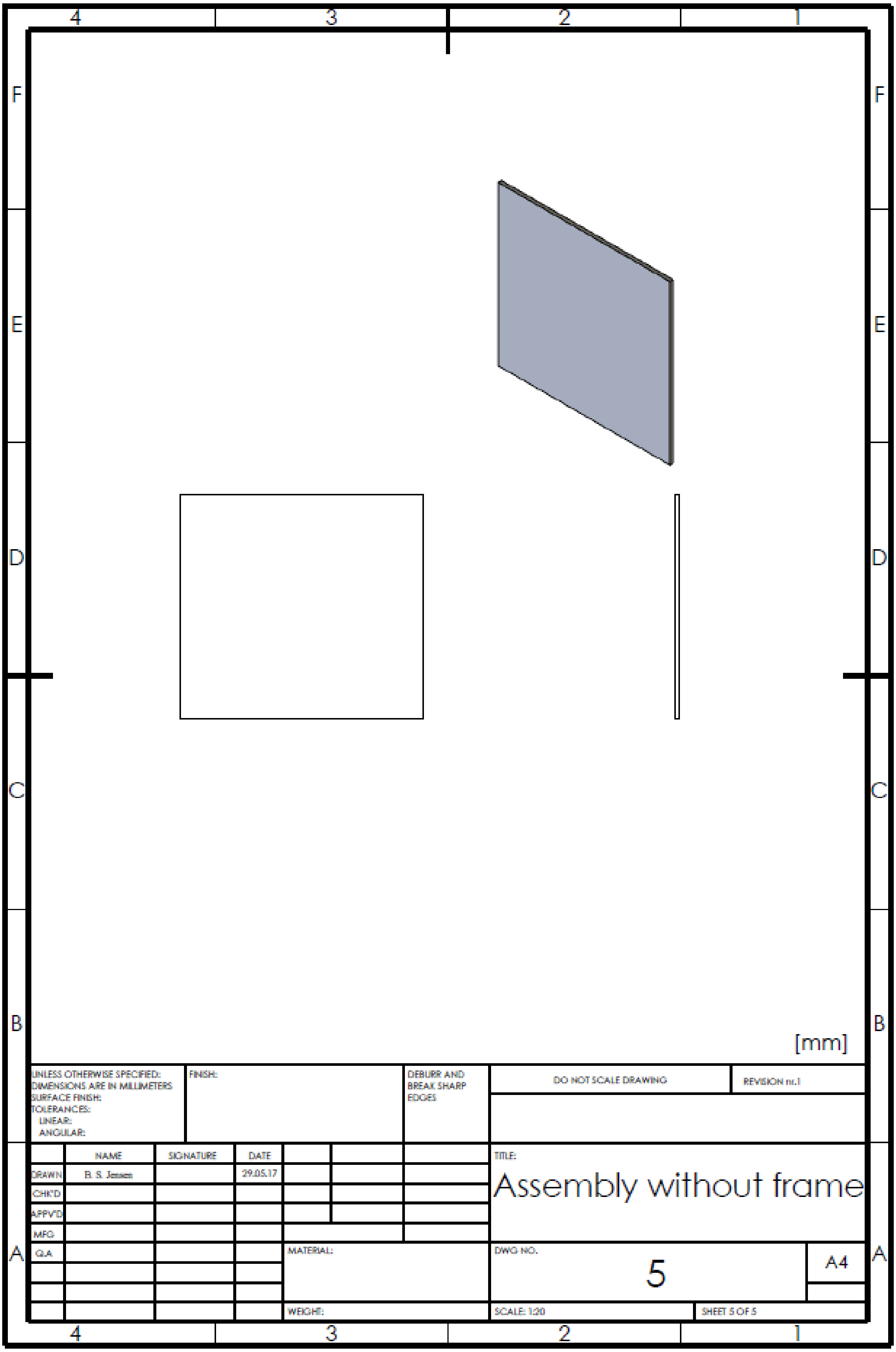
In drawing 1, the corners of the part in the drawing that is in scale 1:2 has been simplified due to lack of dimensions in the original drawing. The thickness of the entire profile is 3mm.




2602
2404
1
[mm]
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION nr. 1
NAME
SIGNATURE
DATE
DRAWN
B. S. Jensen
29.05.17
CHK'D
APPV'D
MFG
Q.A
TITLE:
Facings
MATERIAL:
Aluminum
DWG NO.
2
A4
WEIGHT:
SCALE:1:50
SHEET 2 OF 5


2404
2602
49
[mm]
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION nr. 1
NAME
SIGNATURE
DATE
DRAWN
B. S. Jensen
29.05.17
CHK'D
APPV'D
MFG
Q.A
TITLE:
Core
MATERIAL:
Divinycell H35
DWG NO.
3
A4
WEIGHT:
SCALE:1:50
SHEET 3 OF 5


[mm]
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION nr. 1
NAME
SIGNATURE
DATE
DRAWN
B. S. Jensen
29.05.17
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
WEIGHT:
TITLE:
Assembly with frame
DWG NO.
4
A4
SCALE: 1:20
SHEET 4 OF 5


[mm]
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:
FINISH:
DEBURR AND BREAK SHARP EDGES
DO NOT SCALE DRAWING
REVISION nr.1
NAME
SIGNATURE
DATE
DRAWN
B. S. Jensen
29.05.17
CHK'D
APPV'D
MFG
Q.A
MATERIAL:
WEIGHT:
TITLE:
Assembly without frame
DWG NO.
5
A4
SCALE: 1:20
SHEET 5 OF 5