ID-1489

# SIMONA Research Simulator: Lightweight Design for High Performance and Low Cost Manufacture

Ir. Liesbeth Tromp/Ir. Stefaan de Winter
*Delft University of Technology*
*Centre of Lightweight Structures TUD-TNO*

## Summary

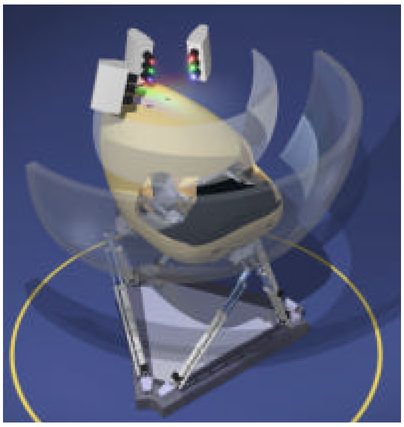


**Fig.1: SIMONA shuttle and VDS**

The Delft University of Technology works on the development of a Research Simulator with high performance through a very small delay time. This is achieved through a lightweight and very stiff structure for which carbon fibre sandwich panels are used in combination with a dedicated carbon/epoxy space frame. In order to achieve a first natural frequency of 20 Hz, the large structure of the Visual Display System is designed for very direct load paths and optimised for high stiffness of structure and joints at a minimum weight. This paper describes the design process from initial hand calculations through FE-analysis and design for manufacture by vacuum infusion at high fibre content.

**Keywords:** design, manufacture, lightweight composite, flight simulator

## 1. Introduction

The Delft University of Technology works on the development of a high performance flight simulator under the name SIMONA. SIMONA is a joint faculty project that will focus on research on SImulation, MOtion and NAvigation of aircraft and other vehicles. Several industrial partners, simulation and non-simulation, participate in this development

**Fig.2: shuttle on motion system**

The flight simulator consists of a composite ‘flight deck’ structure (the shuttle) which is directly attached to the hydraulic motion system. (see figure 2). As such, the simulator has been in operated and tested for one year.

The second stage of the development includes the design and manufacture of control loading, improved software and the Visual Display System (VDS). The VDS features a 180º outside view, which is projected by means of 3 LCD projectors, a back projection screen and a large mirror in front of the flight deck. Such VDS will complete the system to a full flight simulator, similar to FAA approved level D systems, but at much lower weight and dedicated to research

instead of training. The VDS is based on the 10 ft. PANORAMA system by SEOS Displays ltd. The SIMFUSION GP image generator is manufactured by Evans and Sutherland, also a project partner.
The design and manufacturing of the shuttle is described in [ref 1]. This paper will discuss the design, optimisation and manufacture of the Visual Display Support Structure.

Major design requirements of the Research Simulator are:

- A very small delay time of the total system up to 4 Hz (instant response of simulator to pilot actions)
- Good fidelity at high frequency dynamics, such as runway contact and flight through turbulence.

This is achieved by combining a high performance motion system with low mass and a very stiff structure.
The Visual Display System consists of many, large and heavy components placed at a large distance from the center of motion, which makes it critical for the performance of the simulator. This paper describes the design and manufacture of the VDS support structure.

## 2. Design process

The design process was done in two phases. During the first phase a quick FE-analysis was performed on a structure that consists of an all space frame support of carbon fiber tubes, in order to estimate the first natural frequency. This analysis showed:

1) It is difficult to achieve the design goal of 15 Hz with such a large structure.
2) To get a realistic analysis the FEA-model should include:
   - Flexibility of joints
   - 'As built' information of the shuttle
3) Additional load paths are required to increase the stiffness

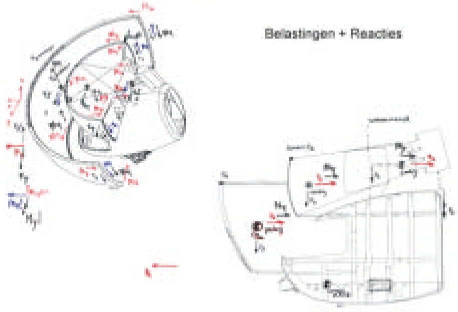
Below the design process of the second phase is discussed in more detail.

### 2.1 Natural frequency requirement

In normal flight, 4 Hz is a reasonable upper limit for the highest bandwidth of the interaction between the pilot and the aircraft. The weight, inertia and flexibility of the shuttle and of the VDS structure will effect motion performance. In order to achieve a 60° phase shift at 4 Hz, the first natural frequency should be at least 2.5 times the 4 Hz
It must be possible to operate the system must be operate the system at high frequencies over 10 Hz. At such high frequencies the structural integrity must be maintained, without phase shift requirement.To assure this, the first natural frequency should be well over 15 Hz.

### 2.2 Concept generation and load path analysis

For all elements of the VDS the specific requirements are listed such as accessibility, light tightening, and effect on image quality and performance of the simulator. Based on the set of requirements, several possible design solutions for the visual display support structure have been generated. Fig. 3 and 4 illustrate the load path analysis.



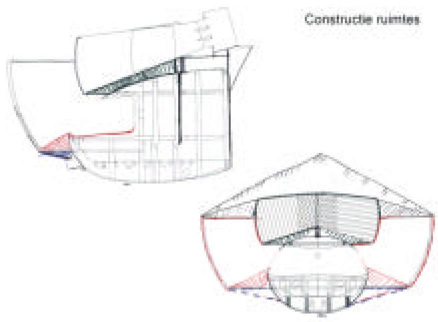




**Fig. 3: load path analysis**

**Fig. 4: identification of construction space available**

The load transfer of the items of the VDS through the shuttle shell to the gimbals (and the floor) is designed to be as direct as possible. This way a stiff suspension is achieved in the most efficient way, i.e. using as little material as possible. Based on this analysis the possible load paths have been determined for each degree of freedom. It was found that pitching (rotation about y-axis) and rolling (rotation about x-axis) are the most critical modes. Accordingly, essential load paths are:

1. The support of the rear of the mirror in y,z-direction
   due to the pitching movement of the mirror cell the ends of the mirror tend to move toward the shuttle (forces in y-direction); rolling results in inertia forces in z-direction.
2. The support of the upper edge of the mirror
   due to the pitching movement of the mirror cell the midfront section of the mirror tends to move in x-direction toward the shuttle.

## 2.3 Structural configurations

The structure consists mainly of large spans which need to be light tight. Therefore suitable structures are either a space frame combined with cloth, or sandwich panels. The generated configurations consist of different combinations of these. Two sketches of structural concepts are presented.

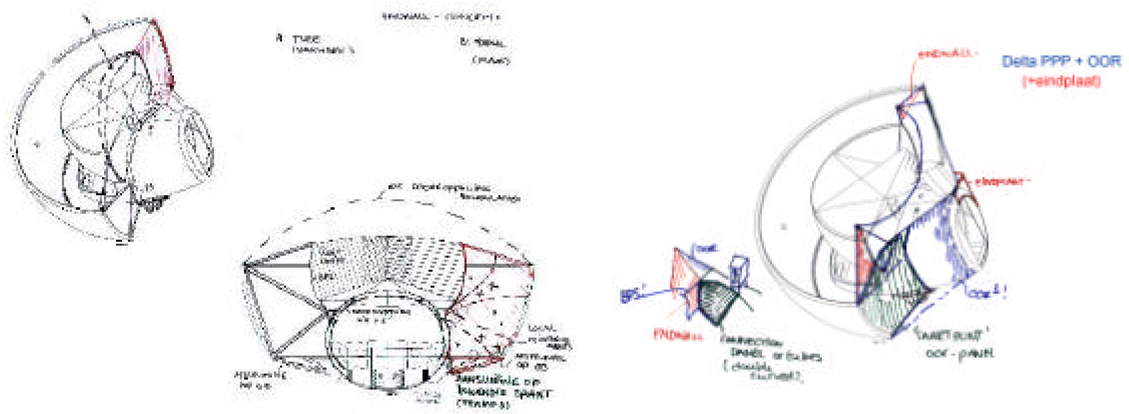



**Fig. 5: examples of structural concept mirror support and projector platform**

The quality of the concepts are judged on the estimated stiffness over weight ratio on functionality and on complexity (costs). Out of the possible solutions, two structural concepts were composed: the canopy-petal concept and the space truss concept. These are modeled in 3D CAD (Pro-engineer) to check geometrical interference with any other parts. In fig. 6-9 the concepts and the denominations for the components of the VDS are shown.

**canopy-petal concept** **space truss concept**

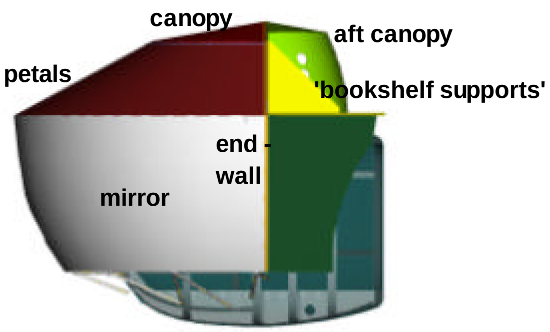



**Fig.6: Side view of the canopy-petal concept**

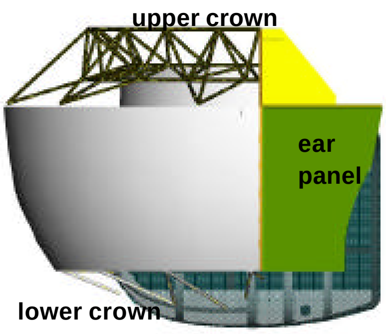



**Fig.7: Side view of the space truss concept**

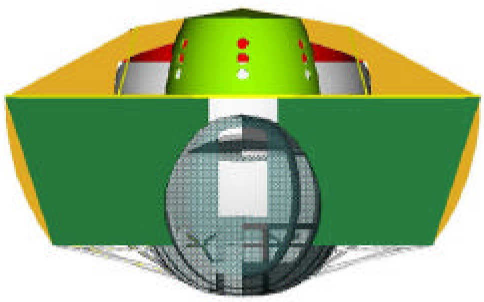


**Fig. 8: Back view of the canopy-petal concept**

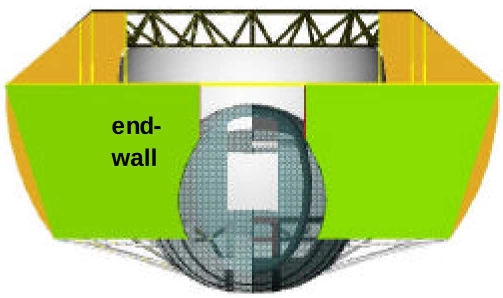



**Fig.9: Back view of the space truss concept**

## Critical aspects

Critical aspects are weight, stiffness, producibility and ease of assembly. Natural frequency and performance of the simulator increase with lower weight and higher stiffness. Manufacturing must be feasible and the cost-performance ratio must be acceptable. The VDS consists of many components that need to be positioned accurately. The ease of assembly influences the accuracy of the positioning of the elements and thus improves the simulation.

These aspects are discussed for the canopy-petal concept and the space truss concept below:

| | |
|---|---|
| Weight | Stiffness over weight is an important criteria. It is expected that the tubular concepts can be lighter than the sandwich panel concepts, but a large weight penalty is imposed by the joints (aluminium balls). The weight will be derived from the 3D CAD model. |
| Stiffness | The most effective way of increasing the stiffness of the structure is by creating direct load paths and reducing the panel span. Carbon reinforced epoxy and sandwich material will be used. |
| Producibility | The composite tubular concept has been elaborated in the first design of the VDS, and a feasible procedure for joining and assembling, though complex, has been found. The sandwich panels require large moulds, but except for the canopy, flat panels can be used. Manufacturing is complicated by the large number of inserts required. The carbon tubes of the lower crown have been produced with carbon prepregs using a blow moulding process. |
| Ease of assembly | The tubular space frame concept is very sensitive to the accuracy of assembly. It highly depends on craftsmanship and is expected to be more time consuming than the assembly of sandwich panels. With sandwich panels the location of the inserts is critical, but this can be controlled by use of reference points in the mould or on the shuttle. |

Based on the above evaluation a choice between the two alternatives could not yet be made. The functional requirement of the natural frequency (‘as light and stiff as possible’) dominates. Using FE-analysis, the eigenmodes of both concepts are compared. A first dimensioning is done analytically, based on the strength and stiffness requirement.

## 2.4 First dimensioning

In the first dimensioning, the basic strength and stiffness criteria are fulfilled. The main loads on the shuttle are inertia loads. The load cases are derived from the capacity of the actuators and are comparable to the load cases used in commercial simulator design.

**Loads**

The maximum inertia loads [ref 2] occur, when all control and safety systems fail and maximum oil pressure is applied in one of the hydraulic actuators. This is an emergency situation and the loads are therefore considered ultimate loads, apart from a safety factor for dynamic effects. The load cases consist of combinations of maximum translational and rotational accelerations of between 15 - 28 $m/s^2$. In the FE-analysis use is made of the actual possible combinations of these loads. For the initial hand calculations, simplified ultimate loads were used of 3g.

## 2.5 Structural analysis

Based on the estimated mass of the items of the VDS, the loads on the joints are determined. Typical maximum loads (compressive and tensile) were found to be 8 kN per connection. The design is clearly dominated by stiffness.

The carbon space frame upper and lower crown have been dimensioned in the first design. In this study the dimensioning of the sandwich panels is discussed. As a first approximation, the sandwich is built up out of quasi-isotropic carbon fiber reinforced epoxy facings and a light PVC foam core.

In case of sandwich panels, thin faces are used for optimum efficiency. The expected critical failure mode is buckling. Thriving design parameter however is the natural frequency. Both buckling as well as natural frequency depend on the free panel size. The analytical evaluation is conducted upon simply supported and clamped panels. A typical maximum free panel size was used of 1.4m x 1.5m.

**Buckling**

The failure modes considered are local as well as global buckling modes. Local modes such as face wrinkling are a point of concern in the joint design and are solved by proper load introduction. For the assumed panel size it was found that a sandwich with 0.5 mm QI-carbon facings and 5 mm thick core, all global buckling loads are over 9 kN.

**Natural frequency**

The minimum required stiffness of the sandwich is not only determined by the buckling criteria, but also by the natural frequency requirement. As in buckling, also the natural frequency is highly dependent on the undisturbed panel size. The panel size is limited by the application of stiffeners or supports. For the canopy concept, panel divisions that are necessary or convenient from a manufacturing or handling point of view are used as a starting point for limiting the undisturbed panel size.

For the first dimensioning, the natural frequency has been considered of rectangular flat panels, either simply supported or clamped on all edges. From [ref 3] the following analytical solutions for isotropic and orthotropic (non-sandwich) panels are given:

For a plate, loaded with a uniformly distributed mass ρ, the first natural frequency is equal to: $f_n = \frac{f_n}{a^2}\sqrt{\frac{D}{r}}$

in which:

$\rho$ = mass per unit area in kg/m$^2$

D = bending stiffness (EI) in Nm

a = length of the panel in m

$\phi_n$ = frequency coefficient

For orthotropic panels the following solution is given:

$$f_{nORTHOSS} = \frac{1.57}{a^2\sqrt{r}}\sqrt{D_{11}m^4 + 2D_0 m^2 n^2\left(\frac{a}{b}\right)^2 + D_{22}n^4\left(\frac{a}{b}\right)^4}$$

with:

D0 = $D_{12}$ + $D_{66}$., the twisting parameter for orthotropic plates

D11, D22 etc = factors from compliance matrix in Nm a /b = length over width ratio of panel
m,n = integers defining mode frequencies
It must be noted that the effect of the shear weakness of the core is not included.

The outcome of the analysis is an initial dimensioning of the cross section, which is used as input of the FE-analysis. The panel analysis is also used for model verification. In order to meet the natural frequency requirement, the free panel size should be reduced to 500mm by 700mm. The sandwich panels were designed according to the following lay up:

-------------- 1mm QI-Carbon facings

10mm PVC foam core

-------------- 1mm QI-Carbon facings

**fig. 9: material build-up**

## FE-evaluation

It was decided to perform a modal FE-analysis, rather than transient dynamic analyses. Such dynamic analyses requires a complete model of the structure including the motion system. This model, which allows predicting the influence of the structure stiffness on the motion delay, will be part of the SIMONA Research Program and is not used as design tool.
In the 3D CAD model of the shuttle, using Pro/Engineer, a surface model has been distracted which is fit for incorporation in the finite element pre-processor MSC/Patran. In MSC/Patran, the imported surfaces are used to establish a finite element model fit for analysis in the finite element code MSC/Nastran.
Based on the experience of the FE-analysis in the first phase, a new more accurate model was built. For all materials in the visual display support structure, both fibre reinforced plies and foam core, the density has been increased with 25% to account for insert, joints, extra resin etc.
In reality, the connections between the components are not completely rigid. The flexibility of the joints has been included by modeling the joints with multiple point constraints [MPC], with which one can explicitly define a certain degree of freedom of a slave node to a degree of freedom of a master node. For bolted joints for example, only translational degrees of freedom have been coupled, and rotations are not restricted. This way of modeling also implies that two panels are connected only at the joint location, whereas straightforward modeling would assume continuous coupling at each coinciding node.
For the modeling of the space truss, from measurements on a real scale component of the space truss, the reduced stiffness due to play in the joints has been determined. The effect was found to be rather large: a reduction of 0.67 has been applied on the material properties. In this way over-estimation of the natural frequencies is avoided.

## 2.6 Effect study

The modes of deformation are identical for both the space truss and canopy-petal concept. They are the same as the first two modes of the shuttle found earlier, see figures 10 and 11.

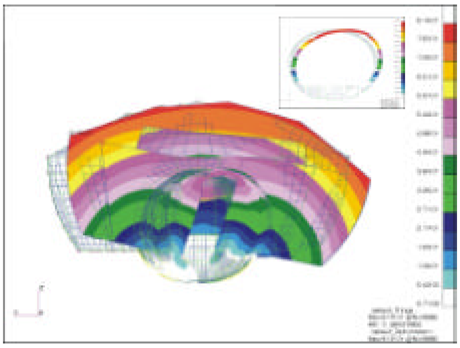


**Fig. 10: First deformation mode of the complete simulator (canopy-petal concept)**

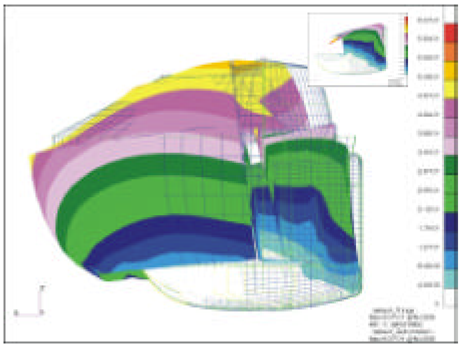


**Fig. 11: second deformation mode of the complete simulator (canopy-petal concept)**

The assumed sandwich built-up proved insufficient for the endwall and ear panels. For these elements the facings are increased from 1mm to 3mm with a core height of 20 mm. With this built up, for the canopy-petal concept the first natural frequency (rolling mode) is 16.7 Hz and the second natural frequency (pitching mode) occurs at 17.8 Hz. For the space truss concepts these values are approximately 2 Hz lower.
An effect study was performed for both concepts, in which the effect of material orientation, sandwich built-up and additional reinforcing elements (beams, rods or frames) on the resulting natural frequency was looked at. It was found that for increased normal or bending stiffness the effect of a higher stiffness is cancelled out by the extra weight added. The application of an external frame on the shuttle frame at the end wall (at the end of the mirror), and reinforcing the edges of the sandwich panels with beams proved effective
The highest resulting maximum natural frequencies are found for the canopy-petal concept: 20.1 Hz for the first mode and 20. 2 Hz for the second mode. The canopy-petal concept is selected and is further elaborated in the detailed design.

## 2.7 Detailed Design

The detailed design mainly involves the design of the connections between the components of the visual display support structure. From the analytical evaluation it was seen that the occurring loads are very low. The material built up as required to fulfill the natural frequency requirement need not further optimised.

**Joints**
The following aspects are of importance in the joint design:

- proper load introduction (local reinforcements)
- detachable or fixed (repair, accessibility )

- assembly of components (assembly sequence positioning, accessibility)

At the joints, compressive loads are introduced locally. Because of the strict weight requirements, very thin facings are used in the sandwich and reinforcements are to be applied only where necessary. Face wrinkling and bearing failure are the critical failure modes at the joints. Because MPC's were used to model the joints, the joint loads are easily derived from the occurring constraint forces of the FE-analysis.
Any play will have a negative influence on the resulting natural frequency of the system. Also, the positioning and stability of the mirror is essential for the image quality. In fact for the image quality, the shuttle -or rather the pilot in the shuttle-, the projectors and the mirror must move as one. The VDS consists of many parts that need to be positioned accurately and assembly is complex. High tolerances are thus required and to compensate for inaccuracies, holes are drilled at assembly.

# 3. Manufacturing

To achieve a low weight, a high fiber volume content is desired. A fiber volume content of at least 50% is aimed at. Large panels are to be manufactured, which excludes the use of an autoclave. The tubes of the lower crown have already been produced using carbon-epoxy prepregs. The shuttle was manufactured using hand laminating in combination with vacuum bagging. For the manufacturing of the VDS the two processes considered are vacuum infusion and low temperature prepregging.
Vacuum infusion was selected, because of the lower costs and the fact that the Centre of Lightweight Structures has ample experience in house.
Qualitiess of vacuum infusion are:

- High quality product
- Low cost material
- Low cost process (no autoclave)
- Clean process (closed moulds)
- High fiber volume content
- Cheap moulds

## 3.1 Vacuum infusion of large sandwich panels

For the manufacturing of sandwich panels, two main aspects to be taken into account are 1) how to prevent large amounts of resin to be absorbed by the core and 2) how to impregnate both facings without air enclosure.
Honeycomb core material is not suitable, since the cells would fill completely with resin. But also in using foam the same effect is seen. At the foam surface the irregularities are filled with resin. This effect could be prevented by sealing the foam surface, but this is impractical.
Even though closed cell foam is used for vacuum infusion, at the surface the cells are no longer intact. For a a typical high performance low density foam with a closed cell structure (for example PVC or PMI) the weight penalty of this can be as much as 1.0 kg/m$^2$. For the application in highly loaded fatigue critical applications, such as wind turbine rotor blades, Rohacell markets a foam with a very high quality material distribution: F-type PMI foam.

For the same density a much smaller cell size was obtained. Figure 12 and 13 show the difference in surface structure. Both panels have a density of 50 kg/m$^3$.

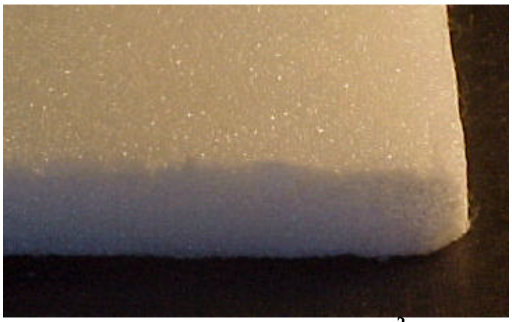


**Fig 12 : 'normal' PMI-foam 50 kg/m$^3$**

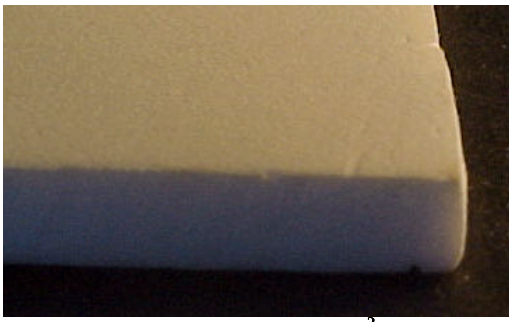


**Fig 13 : F type PMI-foam 50 kg/m$^3$**

Using the HF-type PMI foam, the resin weight due to surface texturewas reduced with 50% to 0.5 kg/m$^2$.

It is very difficult to inject resin at both facings. In the following, single sided injection is considered. When injecting resin in the upper facing, impregnation of the lower facing must be through the core. The technique for through thickness injection of sandwich panels was earlier developed at the Centre of Lightweight Structures for the manufacture of a rudder for the Eaglet, a two-seater composite airtcraft. In fig. 14 and 15 the resulting product is seen.


**Fig14 : rudder of Eaglet**

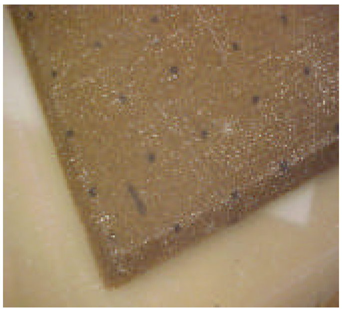


**Fig15: rudder detail: perforated PVC-foam core**

To obtain a coupling of flow between upper and lower facing, the foam is perforated. The hole pattern, or rather the number and size of the holes, depends on the permeability of the facing laminate. The impragnation pattern of the lower facing is a pattern of injection points: a circular flow front will emerge from each hole see fig. 16. If the distance between the holes is too large in relation to the permeability of the facings, the flow fronts of adjacent holes and the flow fronts of the consecutive holes will meet too soon and air is enclosed.
Furthermore, to reduce the risk of air enclosure, it is desired that the holes are filled one after the other. If the permeability of the upper facing is very high, for example by the application of a distibution medium, all holes will fill almost simultaneously which also will result in air enclosure. Preferrably both facings have comparable permeability. This is seen from the simulation of injection process of a sandwich with perforated core using RTM-worx, fig. 16.

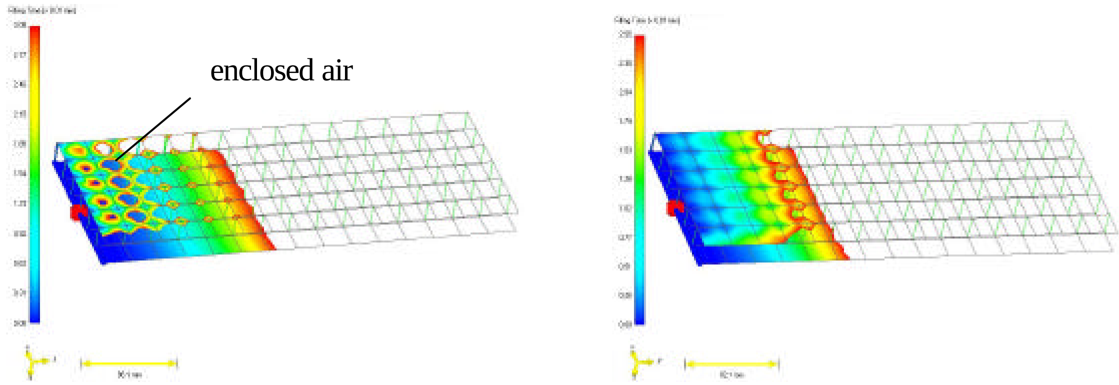



**Fig16 injection of sandwich large (left) difference in permeability and small (right) (no air enclosure)**

For the VDS a large number of double curved sandwich panels have been manufactured. An example is shown in fig 17 and 18: the SIMONA 'mask' to be placed in the window cut out as part of the aircraft simulation interior.

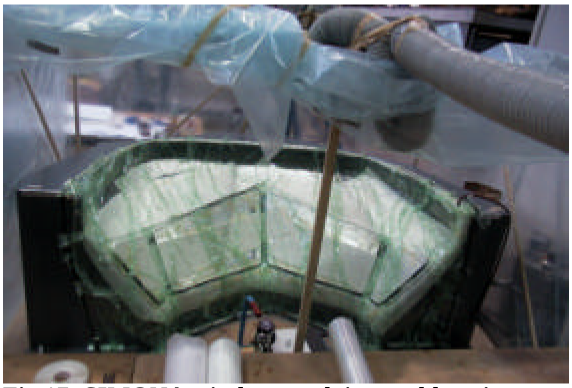


**Fig 17: SIMONA window mask in mould,curing**

**Fig 18 : SIMONA window mask, untrimmed**

# 4. Concluding

With the canopy-petal structure a theoretical first natural frequency of 20 Hz is realised. Eigenfrequency measurements will be performed on the shuttle to determine the accuracy of the FEA-frequency predictions. In the summer of 2001 the structure will be built and mounted to the shuttle. The large double curved sandwich panels of the visual display support system are produced using vacuum infusion. A special process for through thickness injection of sandwich is used. With this process, the manufacture is optimised for high quality, low cost and low weight (high fiber volume content).

# References


Ref 1. Haan de, P.A.J., The Design and Manufacturing of the All Composite Flight Deck of the SIMONA Research Simulator, Proceedings of 20$^{th}$ SAMPE Europe Conference of the Society for the Advancement of Material and Process Engineering, Paris, 1999

Ref 2. Hanssen, L.C.W. Hanssen, " Design of the SIMONA Research Simulator Structure", thesis Structures and Materials Laboratory, Faculty of Aerospace Engineering, Delft University of Technology, Delft, August 1995

Ref 3. 'Structural Plastics Design Manual, Volume 1', American Society of Civil Engineers, New York, 1984