Potential of fibre-reinforced components for lightweight construction machines with low noise emission

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ABSTRACT

The emission of sound and noise of machines especially for construction companies is increasingly considered as important. This is caused by the strict limitations of the overall noise emission on construction areas. Thus, machines with low noise emission are more and more required by the manufactures of construction equipment. Furthermore, construction machines have to be transported between different construction areas, wherefore they have to be light and easy to handle. Besides, a flexible and fast maintenance is required. A possibility to reach these targets is the use of modern fibre-reinforced composites with their adjustable property profile regarding stiffness, strength and sound emission [1].

Using the example of an upper side hood of a cooling air outlet, the potential of glass fibre-reinforced composite was analysed. Besides the main focus of reducing the weight of the hood, the noise emission was considered in the design process. Starting with an analysis of the acoustical, thermal and mechanical behaviour of a reference hood made of steel, different fibre-reinforced concepts were developed.

As a result, the usage of glass fibre-reinforced epoxy in combination with a material adapted design allows a reduction of the overall mass of the hood up to 85 %, achieving the requirements concerning stiffness, deformation, impact and sound. Based on the preliminary investigations, a prototype was build showing a significantly reduced weight. Subsequently, the prototype was installed at a construction machine and sound pressure measurements were done with both configurations. The resulting noise emission of the construction machine was kept constant and consequently additional noise absorbers got obsolete.

Keywords: Low-Noise-Emission, Lightweight, Hood, Silencer

I-INCE Classification of Subjects Number(s): 34

1. INTRODUCTION

Modern construction machines are characterised by a high power density, huge work capacity and high efficiency. During the last years, the noise emission was kept constant while the work capacity was raised. But in future, the installation of more and more powerful engines and additional aggregates will result in higher sound and noise emissions. To avoid this, e.g. special housing elements and air hoods decreasing the sound emission are added to the machines.

There are special challenges for these housings and hoods caused by the rough environment on the construction areas. The main challenges focused here are:

1. Wide temperature range between -40 °C and +120 °C
2. Resistance against oil, petrol and coolant
3. Impact resistance and walk-on stability according to DIN-EN-ISO 2867 [2]
4. Easy to handle weight and size (according to ISO 11228) for maintenance and service [3]
5. High sound absorption and sound enclosure capability to meet the noise emission target according to european directive 2000/14/EG [4]

In common design these hoods are made of steel. To reach the acoustical target, additional absorption elements are installed based on wire wool and perforated metal plates (see Figure 1).

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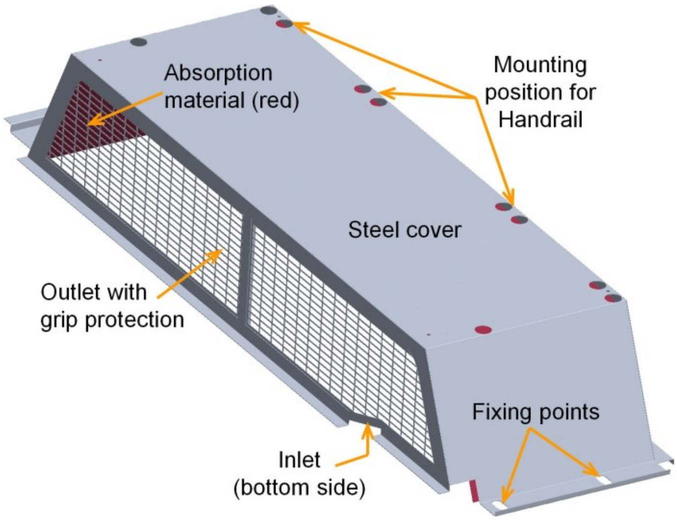

Figure 1 - Model of chosen reference hood made of steel with mounted absorption panels

Consequently, the weight of the hoods is very high - up to 600 kg. To mount, dismount and remove the hood for maintenance or transport reasons, technical equipment (e.g. a forklift) is required. Since for future machine generations these hoods should be handled without additional technical equipment, the weight has to be significantly reduced without negative influences on the other qualities. Because of their high stiffness-to-weight ratio in combination with the high material damping fibre-reinforced composites offers a high potential for this application [1]. Starting with an acoustical, thermal and mechanical analysis of a reference machine and a reference hood made of steel, different fibre-reinforced concepts were derived.

2. FUNDAMENTALS

The main functions of the investigated hoods are the sound insulation and to provide a high flow-rate for used air. The resulting problem includes a combination of acoustical and fluid mechanical effects leading to an aeroacoustic problem. The discipline of aeroacoustics arised within the middle of the 20th century, despite some first phenomenological investigations were started much earlier [5]. In the field of aeroacoustics three main parts are distinguished: the generation, the dissipation and the transportation of noise and sound inside a fluid flow.

The generation of sound inside a fluid is caused by turbulent effects which lead to pressure variations which finally could be recognized as sound (Figure 2).

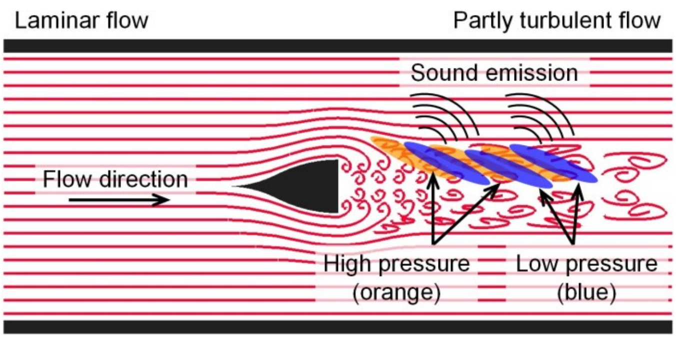

Figure 2 - Schematic draft of sound generation within a fluid flow

The dissipation of sound is based on the damping behaviour of the surrounding fluid. So at one hand a pressure difference inside a fluid could be equalised by small flowing effect between high and low pressured areas and on the other hand by internal and external friction within the elements of the fluid as well as between the fluid elements and connected wall elements.

Finally, the fluid could also work as a transportation vehicle for acoustic waves, neglecting its generating and dissipation capability. Here, additional elements like absorbers or resonators are needed to influence the sound transmission behaviour. For aeroacoustical problems different absorber types are available: absorption silencers, relaxing silencers, reflection silencers, resonance silencers and interference silencers [6].

Absorption silencers are based on the absorption capacity of a porous surface or structure like foams. For aeroacoustical applications the silencer panels are mainly orientated in flow direction. The carried sound energy is absorbed by friction effects inside the soft surface or structure (Figure 3).

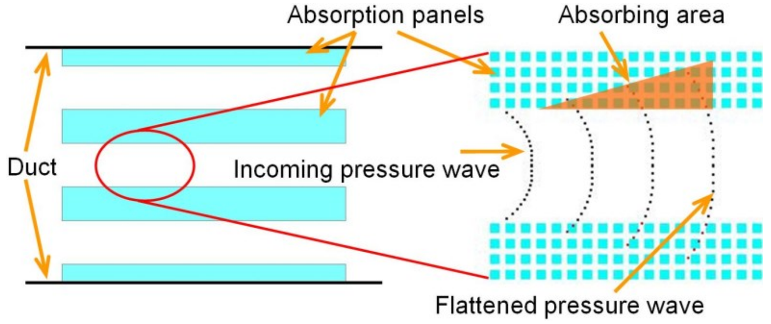

Figure 3 - Schematic draft of sound absorbing mechanism in an absorption silencer

These silencers are able to work in a wide frequency range but just for lower sound energies. To improve the damping effectivity for higher sound energies, relaxation silencers where designed. Here, a thin surface e.g. a metal film acts as cover for a soft core. The general effectiveness is equal to the absorption silencers, but the hard cover offers the possibility to deal with higher energies (Figure 4). Furthermore, closed surfaces are less susceptible for mud and gutter and could be easier cleaned using high-pressure cleaners.

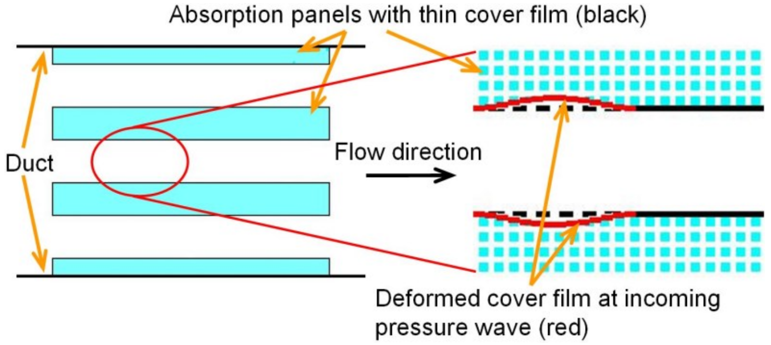

Figure 4 - Schematic draft of sound energy harvesting in a relaxation silencer

Since the principle is based on resonance effects, the silencer shows a limited frequency range which is mainly controlled by the stiffness and thickness of the cover layer. Thus, for wide frequency and energy ranges, a combination of both silencer types, absorption and relaxing silencers, is needed.

Beside these both structural absorbers also geometrical absorbers are available to influence the aeroacoustic behaviour of ducts. E.g. reflection silencers are based on a widening of the flow section. The length of this widening defines in combination with the flow speed the effected frequency (Figure 5).

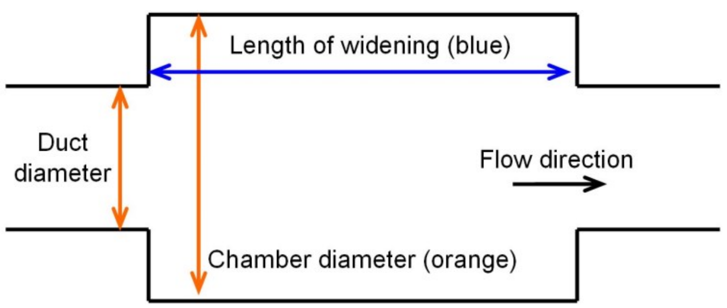

Figure 5 - Schematic draft of a reflection silencer

Resonance absorbers are well known from other acoustical applications as well. Two of the most used type are the so called Helmholtz resonator and $\lambda/4$ resonator. In both cases additional chambers are working as buffers to equalise the pressure amplitudes (Figure 6).

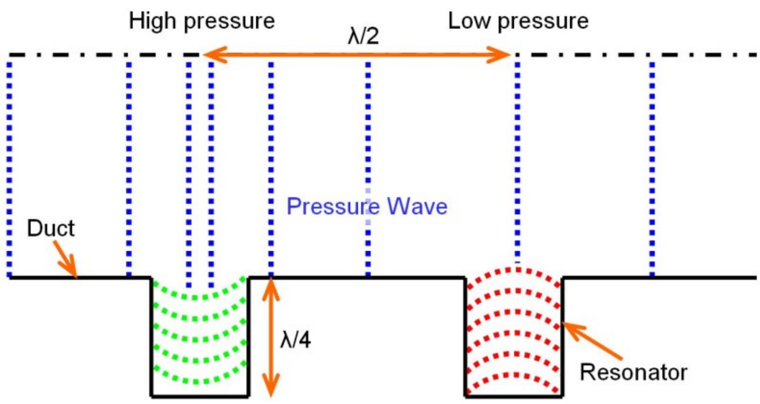

Figure 6 - Schematic draft of a λ/4 resonance silencer

In general these silencers just work for one frequency, but by a combination of different buffer sizes and distances, also a defined range of frequencies could be absorbed [3]. Finally a last possibility of geometric based silencers should be shown – interference silencers. Caused by the geometric creation of interferences at the end of the silencer a wide range of frequencies and sound power levels could be influenced (Figure 7).

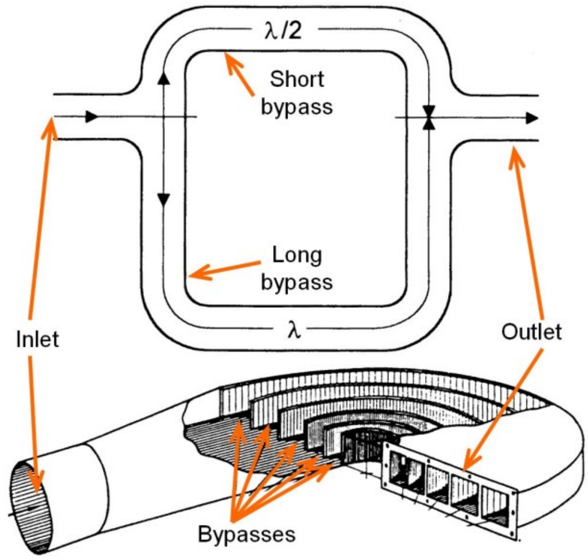

Figure 7 - Schematic draft and example of an interference silencer [3]

All shown silencer types are leading to a bigger construction space of the duct system. The calibration of the geometric based silencers is manageable for a single or a low number of frequencies and consequently not usable as broadband absorbers. Here, the material based absorber types offer good possibilities. The effectivity of these two types is mainly influenced by the effective surface and thickness of the absorber mats. So an elongation or widening of the silencer or an increase of the absorber thickness are common possibilities to adapt the sound absorbing capability.

3. DESIGN AND CONSTRUCTION OF A REPLACEABLE GLASS FIBRE-REINFORCED HOOD

Based on a chosen steel made hood (Figure 1) a replaceable glass fibre-reinforced polymer hood (GFRP-hood) was designed. The resulting construction is driven by the interchangeability of the hoods. Also the exterior design should not be changed within this first step. This approach offers the possibility to analyse the effect of the material without considering any other effects like geometry or interface changes.

3.1 Free cross section for used air flow

At the Beginning of the investigations, the minimum required free cross section for the airflow was determined. Starting with a theoretical approach based on the power and working capacity of the mounted fan system and the thermodynamic behaviour of the engine and hydraulic aggregates the required free cross section was calculated. For that, four measurement points (MP) inside the air flow where defined (Figure 8).

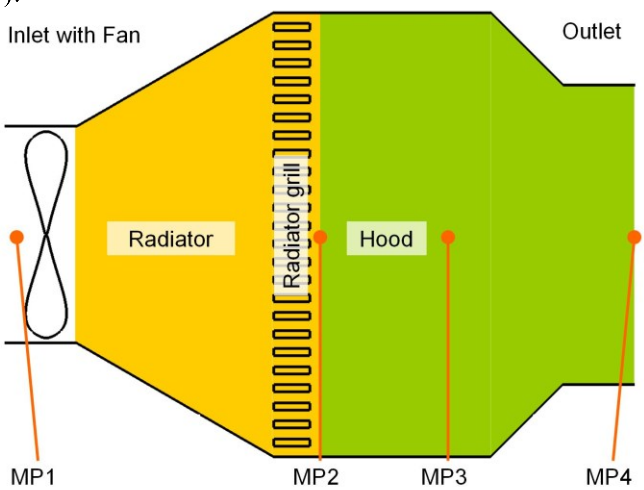

Figure 8 - Schematic draft of the measuring point for the calculation of the needed free cross section at the outlet

Using the fluid properties at MP 1 and the geometrical parameters at MP 2 the required free cross section at MP 3 and MP 4 could be calculated. So, a significant reduction of approx. 50 % for the free cross section inside the hood and at the outlet was estimated (Table 1).

Table 1. Current used cross sections and calculated minimum required cross sections of different machines

Machine	Current cross section	Required cross section
A	approx. 1.2 m²	approx. 0.55 m²
B	approx. 0.6 m²	approx. 0.33 m²
C	approx. 1.1 m²	approx. 0.49 m²

To evaluate these theoretical results, some measurement with the current hood and a modified steel hood were performed and the resulting temperature of the oil circuit was documented. Figure 9 shows the measured oil temperatures during the maximum load case as a function of working time.

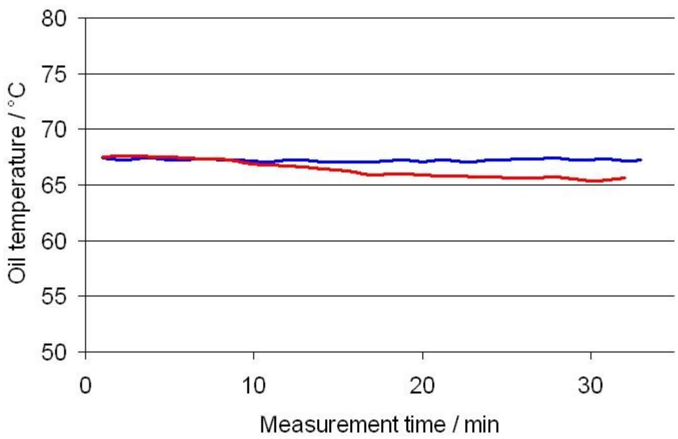

Figure 9 - Comparison of the oil temperature for different sizes of free cross section areas for the current hood (blue line - standard configuration, red line – modified configuration with reduced cross section

The resulting difference between the oil temperature for the common sized steel hood and the steel hood with a reduced outlet is negligible. So the theoretical approach was verified and for future hoods the free cross section could be reduced. This fact allows e.g. to insert additional absorption elements in order to improve the sound insulation capability of the hood without adapting or changing the outer size.

3.2 Mass reduction of the hood

The heavy weight of the current steel made hood leads to problems while maintaining or transporting the construction machine. In both cases the hoods have to be removed. Because of the huge weight additional carrying equipment like fork lifters is required. The new GFRP made hood should be manageable without additional technical equipment. For the reconstructed geometry just a one piece solution is feasible. Based on the carrying limit concerning to standard [3] an overall weight limit of the hood is given by approx. 100 kg to enable the possibility of handling it by two male persons. To meet this target, a lightweight design made of thin ribs and cover layers was chosen (Figure 10).

Using finite element (FE) pre-designing tools, the thickness and geometry were optimised considering the geometric limits of the common steel hood. In the End an overall weight of just 81 kg was realised. This means a reduction of the mass of more than 75 % compared with the original steel hood. Taking into account the higher material damping of GFRP compared to steel, the additional absorption panels at the steel hood could be leaved out for the GFRP hood. Assuming this, the weight of the new GFRP hood is just 15 % of the original steel hood.

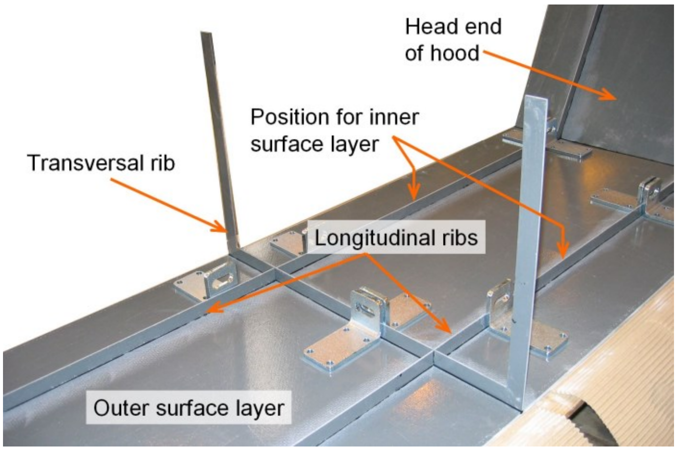

Figure 10 - Lightweight structure of the GFRP-hood

3.3 Standard conformity and quality impression

The stiffness and the deformation of the redesigned GFRP-hood have to meet the limits given by the standard for stand and walk areas at construction machines [4], where a spotty force of 2000 N (respective 200 kg) and an areal pressure load of 4500 N/m² have to be suffered without permanent deformation (Figure 11).

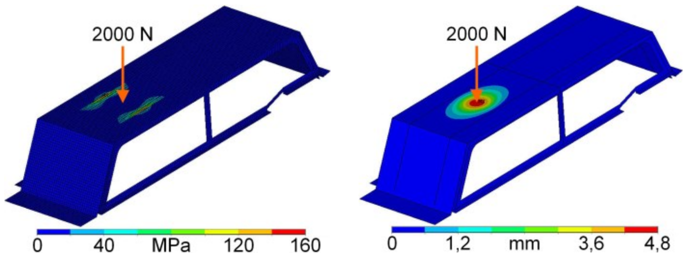

Figure 11 - Resulting equivalent stress (left) and deformation (right) for the spotty load case at the redesigned GFRP-hood

Considering the two standardised load cases, the spotty load leads to higher local deformations and tensions. A maximum tension of approx. 165 MPa is calculated for this load case what is less then the half of the tolerable tension for GFRP. So, no permanent deformation of the hood has to be expected for both load cases and the legal restrictions for using GFRP could be met. Furthermore, the resulting deformations for both load cases are less than 5 mm, what is acceptable to guaranty a high quality impression of that hood.

4. MANUFACTURING AND MEASSUREMENTS

4.1 Manufacturing

Considering the predefined results a GFRP hood was manufactured using water jet cut plates and rips. At first the main rips and the outer cover layers were joined (Figure 10). In the next step the inner cover and the side frames were mounted. Finally, the air baffle and a thin sponge rubber strip were added (Figure 12).

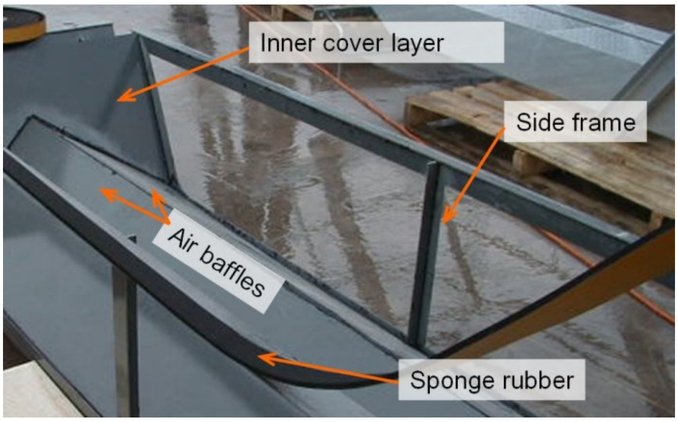

Figure 12 - Final assembly of the GFRP hood

4.2 Measurement

After manufacturing the GFRP-hood and the reference steel hood were mounted consecutively at the same machine and different standardised machine load cases were run. The used measurement setup and the positions of the microphones according to [4] are shown in Figure 13.

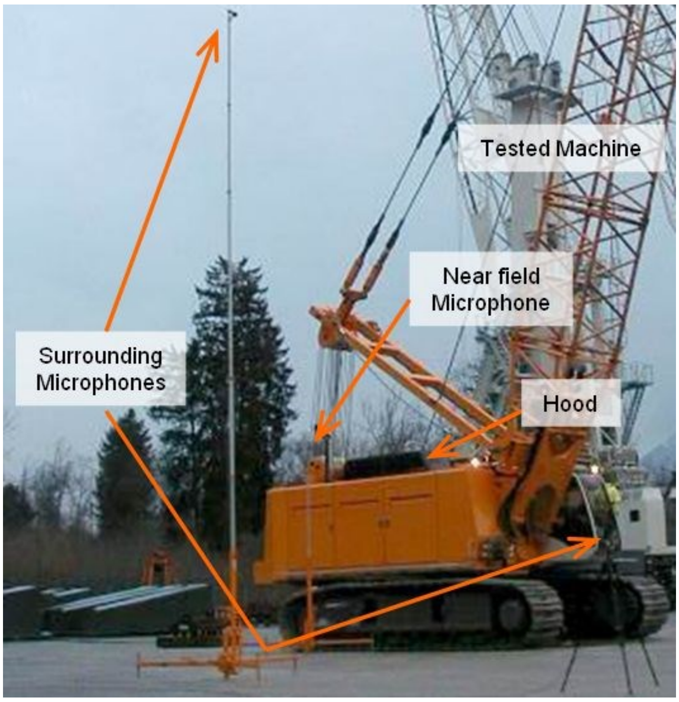

Figure 13 - Microphone positions according to european directive 2000/14/EG [4]

For all load cases, the sound emission at the marked microphone positions are measured and evaluated. In Figure 14 the measured sound pressure spectra at full machine load for the steel hood with mounted absorption elements and the new GFRP-hood without additional absorbers are shown.

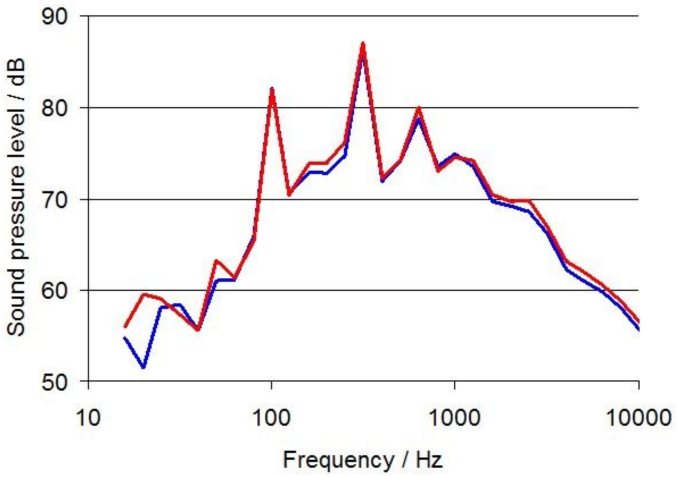

Figure 14 - Resulting sound pressure level for the steel (blue) and the GFRP (red) hood during the same load case

The resulting differences between both curves are less than 2 dB within the range of 80 Hz to 10.000 Hz. So, in this frequency range the efficiency for both hoods is nearly the same. Taking into account the missing absorbing elements inside the GFRP-hood, the high vibro-acoustical potential of GFRP is clearly shown.

Parallel to the acoustical measurements, also the temperature of the machines oil was measured to investigate the influence of the hood material to the heat dissipation of the machine. In Figure 15 the resulting temperature charts over the time are shown.

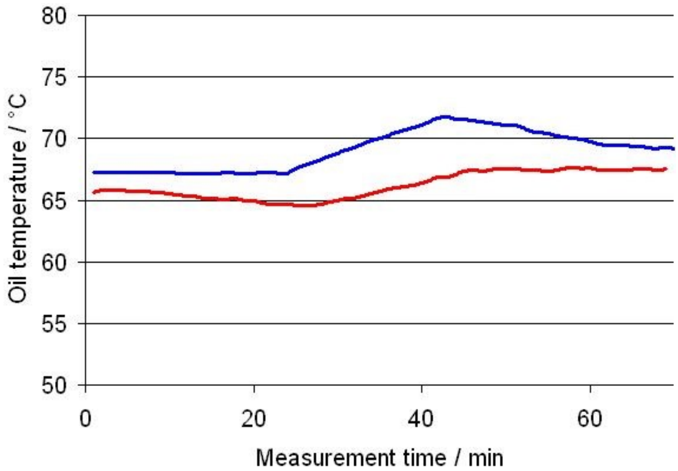

Figure 15 - Resulting oil temperature for the steel (blue) and the GFRP (red) hood during the same load case

The machine with mounted GFRP hood shows lower oil temperatures during the whole working time. This effect is caused by the interior surface of the two hoods. Inside the steel hood additional sound absorber elements were mounted, which leads to a higher flow resistance. At the GFRP-Hood these panels weren't mounted, but the surface geometry of these elements was redesigned using GFRP-plates. So the effective flow resistance of the GFRP-hood is lower than for the steel hood causing a higher air mass flow and consequently a lower oil temperature.

5. DERIVATION OF NEW GFPR-CONFORMING HOOD CONCEPTS

The design and the behaviour of the chosen steel hood are not fixed by any special design or construction guidelines. Thus, the shape for the investigated GFRP-hood was taken from the current solution. To ensure a GFRP-conforming design of new hoods, the geometry should be changed. So, different new hood designs were developed and virtually investigated using FE-tools.

To ensure a high quality impression for the GFRP-hoods, the standardised efforts according to [2] were complemented by own defined deformation limits. So the accepted deformation of the new hood is limited by approx. the same than of the current steel hood. As load cases for these limits two internal standardised cases were used: deformation under dead load and resulting by an 80 kg person standing at the weakest point on top of the hood.

Based on the results of the pre-investigations new hood concepts were designed with a reduced flow cross section and a GFRP-conform design. So, additional rips and air baffles were inserted to create a light and stiff structure on the one hand and longer flow paths for the used air on the other hand. As a result, a very light air hood with just 54 kg over all weight while retaining the dimensions was designed (Figure 16).

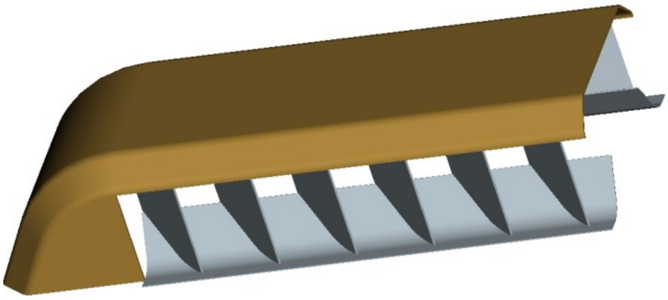

Figure 16 - Concept of a future lightweight GFRP hood

To exploit and improve the high vibro-acoustic potential of GFRP, a concept for a further extension of the air path and an application of additional absorption materials was designed. This advanced concept is based on some smaller complex shaped modules which are easier to handle compared to the full hood and could be plugged together to adjust the hood length for different machine types (Figure 17).

This modular hood design offers a great variability for the length and tolerable costs because of the usage of equal parts for one hood as well as for different machine types. Furthermore, the effective flow cross section could be adjusted by the chosen absorption mats and, especially for smaller machines with less used air the resulting sound emission can be significantly reduced. The designed hood with the same dimensions than the reference hood has an overall weight of 165 kg including the absorption materials, which means a mass reduction of approx. 72 %. Because of the modular construction with modules of 35 kg each, it is possible to handle even for one person.

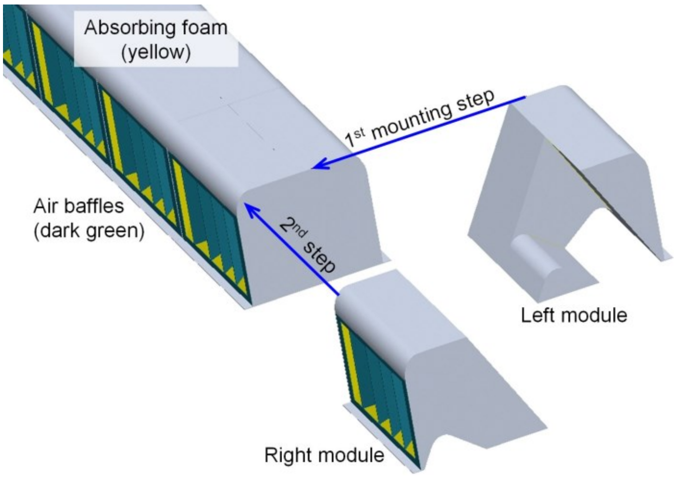

Figure 17 - Modular concept of a GFRP hood with extended flow path and additional absorbing materials inside

6. CONCLUSION

The increasing need of powerful mid-size and big construction machines with a reduced noise emission lead to the installation of additional sound insulating elements and structures. Here, especially additional housing and hood structures are mounted or existing systems be adapted. For transportation and maintenance, this hood and housing elements have to be removed. Caused by the huge weight of these structures, usually additional carrying equipment like a forklift is required. In order to enable the possibility of moving and lifting future hoods without additional equipment, the mass has to be reduced significantly.

This paper presents a possible approach to reduce the overall weight by replacing a common steel hood with a glass fibre-reinforced lightweight hood. For this purpose, the mechanical and acoustical behaviours of a selected steel hood were analysed and an equivalent glass fibre-reinforced composite hood was designed. By testing the new hood under real operation conditions, the high lightweight and vibro-acoustic potential of GFRP as structural material for housing elements and hoods of construction machines could be shown. The main results within this paper are:

1. A significant reduction of the weight, starting from approx. 600 kg for the common steel hood, up to 85 % for a lightweight GFRP hood was attested, without loss of performance.
2. A reduction of the effective flow cross section inside the hood of approx. 50 % is possible without any loss of cooling performance.
3. A modularized concept with huge lightweight and acoustic potential was developed.
4. A cost reduction potential for GFRP hoods compared to common steel hoods could be expected also for small-scale series.

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