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Engineering against failure (5th ICEAF)

Preventing failure of engineering structures necessitates involvement of a large amount of expertise and, depending on the application, might require for a multi-disciplinary and multi-scale approach in terms of both size and time. The series of the International Conference of Engineering Against Failure (ICEAF) aims to provide on a biennial basis a forum to present relevant scientific and technological achievements and discuss with an audience of international experts.

This special issue contains selected papers from works presented at the ICEAF V, which was held in Chios, Greece, on June 20–22, 2018. More than 150 participants contributed to a high-level scientific gathering providing some of the latest research results on the topic as well as some of the latest relevant technological advancements. Particular attention has been given at offering the opportunity to young researchers, from Greece and all over the world, to present their work.

The topics of the Conference were addressed by seven key-note lectures given by world-class scientists and 144 scientific papers structured in a total of 27 sessions. The scientific responsibility of the ICEAF V Conference was shared between the Laboratory of Technology and Strength of Materials and the Applied Electronics Laboratory, both University of Patras. Furthermore, the ICEAF V Conference was held under the auspices of the Federation of European Materials Societies, the Hellenic Metallurgical Society, the European Aeronautics Science Network and the Industrial Systems Institute.

We are confident that the 6th ICEAF Conference in Spetses Island will build on the previous successful ICEAF Conferences and continue the already established tradition of a "customized," low-cost conference acting as a forum for experienced international scientists but also young, motivated researchers to present new scientific knowledge and technological innovation, exchange views and establish new cooperation perspectives.

Looking forward to welcoming you to ICEAF VI Conference, in Spetses, Greece 2020!

The Conference Co-Chairmen

Professor Spiros Pantelakis
Professor Stavros Koubias

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Laboratory of Technology & Strength of Materials,
University of Patras, Patras, Greece, and

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Ballistic impact of steel fiber-metal laminates and plates

George Bikakis, Nikolaos Tsigkros and Emilios Sideridis
National Technical University of Athens, Zografos, Greece, and Alexander Savaidis
Department of Mechanical Engineering Educators, School of Pedagogical and Technological Education, N. Heraklion, Greece

Abstract

Purpose – The purpose of this paper is to investigate the ballistic impact response of square clamped fiber-metal laminates and monolithic plates consisting of different metal alloys using the ANSYS LS-DYNA explicit nonlinear analysis software. The panels are subjected to central normal high velocity ballistic impact by a cylindrical projectile.

Design/methodology/approach – Using validated finite element models, the influence of the constituent metal alloy on the ballistic resistance of the fiber-metal laminates and the monolithic plates is studied. Six steel alloys are examined, namely, 304 stainless steel, 1010, 1080, 4340, A36 steel and DP 590 dual phase steel. A comparison with the response of GLAss REinforced plates is also implemented.

Findings – It is found that the ballistic limits of the panels can be substantially affected by the constituent alloy. The stainless steel based panels offer the highest ballistic resistance followed by the A36 steel based panels which in turn have higher ballistic resistance than the 2024-T3 aluminum based panels. The A36 steel based panels have higher ballistic limit than the 1010 steel based panels which in turn have higher ballistic limit than the 1080 steel based panels. The behavior of characteristic impact variables such as the impact load, the absorbed impact energy and the projectile's displacement during the ballistic impact phenomenon is analyzed.

Originality/value – The ballistic resistance of the aforementioned steel fiber-metal laminates has not been studied previously. This study contributes to the scientific knowledge concerning the impact response of steel-based fiber-metal laminates and to the construction of impact resistant structures.

Keywords Ballistic impact, Ballistic limit, Explicit finite element analysis, Steel alloy, Steel fiber-metal laminate, Steel plate

Paper type Research paper

1. Introduction

Research concerning the mechanical evaluation of composite materials started several decades ago and is currently being carried out in specialized aspects of their mechanical behavior (Kadlec et al., 2016; Koumoulos et al., 2016; Nezbedova et al., 2016; Raimondo et al., 2016). Fiber-metal laminates are hybrid composite materials, consisting of alternating metal layers bonded to fiber-reinforced prepreg layers. Aramid Reinforced ALuminum Laminate (ARALL), CArbon Reinforced ALuminum Laminate (CARALL) and GLAss REinforced (GLARE) belong to this new family of materials. Apart from aluminum, other metal constituents such as steel, magnesium and titanium have been employed in order to manufacture fiber-metal laminates. The possible variations of hybrid laminated material systems are numerous and their mechanical evaluation is a challenging intensive field of scientific research.

The response of steel-based fiber-metal laminates under various loading conditions is very interesting since steel is a classical construction material and their applications can be numerous. Recently, scientific articles concerning fiber-metal laminates with steel metal layers have been published. In these articles, various applications are considered: thick walled tubes subjected to axisymmetric internal pressure (Nowak and Schmidt, 2015), shear walls for the construction of buildings (Petkune et al., 2016), panels for underwater applications (Poodts et al., 2015) and industrial applications (Parnanen et al., 2015). The mechanical evaluation of steel-based fiber-metal laminates includes their impact response, fatigue characteristics, creep behavior, hygrothermal conditioning and earthquake loading response.
The design of impact resistant laminated structures is very important for industries such as aerospace, automotive and shipping since during the life of a structure, impact by foreign objects can be expected to occur during manufacturing, service and maintenance operations (Perogamvros et al., 2016; Sierakowski, 1997). Furthermore, the design of laminated armor to defeat small-caliber projectiles is of primary interest in military applications.

In this paper, extensive research has been implemented in order to investigate and compare the influence of the mechanical properties of different steel alloys on the ballistic resistance of glass-reinforced fiber-metal laminates and monolithic metal plates consisting of these alloys. ANSYS LS-DYNA finite element software is employed for this purpose. A total of 30 different ballistic impact cases have been simulated. Furthermore, the behavior of characteristic impact variables during the impact event is studied.

2. Finite element modeling

For the simulation of the normal ballistic impacts of free-flying projectiles on the targets, we use the structural arrangement of the ballistic impact experiments of Hoo Fatt et al. (2003). Specifically, in these experiments the clamped target is square with dimensions 152.4 mm $\times$ 152.4 mm whereas the projectile is flat-faced Ti-6Al-4V, AMS 4,928 cylinder with a diameter equal to 12.7 mm. The length of the projectile is equal to 25.4 mm and its impacting face has a small radius equal to 0.8 mm. The projectile’s mass is equal to 14.125 g. The impact tests were conducted at room temperature using a gas gun. The velocity of the projectile prior to impact was measured by the interruption of two laser beams a known distance apart.

Fiber-metal laminated targets with coded names 2/1-0.5, 3/2-0.3 and 3/2-0.5 are considered in this paper. As an example, the coded name 3/2-0.5 means that the fiber-metal laminate consists of the following layup: (metal/0° glass/90° glass/90° glass/0° glass/metal/0° glass/90° glass/90° glass/0° glass/metal). The first number of the coded name yields the number of metal layers in the laminate. The second number of the coded name yields the number of composite S2-glass UD fiber-reinforced epoxy layers. Each glass-epoxy layer has a layup (0°/90°)$_s$. Each prepreg ply has a thickness of 0.127 mm. The third number of the coded name is approximately equal to the thickness of each metal layer in mm. The exact thickness of each metal layer is equal to 0.3048 mm in 3/2-0.3 fiber-metal laminates and 0.508 mm in 2/1-0.5 and 3/2-0.5 fiber-metal laminates. These definitions concerning the coded name of fiber-metal laminates are valid for all considered targets of this paper. With reference to the total thickness of fiber-metal laminates, as an example, the total thickness of a 2/1-0.5 fiber-metal laminate is calculated as follows: $2 \times 0.508 \text{ mm} + 4 \times 0.127 \text{ mm} = 1.524 \text{ mm}$.

In our finite element modeling procedure we employ SOLID164 elements to mesh the projectile and the targets. In order to simulate the contact between the projectile and the panel, we use the surface-to-surface contact type with the eroding contact option. For the simulation of the contact between adjacent layers of the fiber-metal laminate we use the surface-to-surface contact type with the tiebreak contact option. The tiebreak contact option is needed in order to allow delamination/debonding between adjacent layers. With this option, the delamination/debonding is governed by the following failure criterion (LS-DYNA, 2014):

$$\left(\frac{\sigma}{\sigma_c}\right)^2 + \left(\frac{\tau}{\tau_c}\right)^2 \geq 1,$$

where $\sigma$, $\tau$ are the normal and shear stresses acting in the interface of two adjacent layers and $\sigma_c$, $\tau_c$ are the normal and shear strength of the interface, respectively. It is assumed that delamination/debonding is taking place due to the interfacial shear stresses between the
contacting layers. For this reason, an increased value of the normal interfacial strength is employed, yielding a shear-driven delamination/debonding failure between glass-epoxy layers, as in references (Yaghoubi and Liaw, 2013; Yaghoubi and Liaw, 2014). Specifically, $\sigma_c$ was set equal to $2 \times 10^5$ GPa whereas $\tau_c$ was set equal to 20 MPa (Hoo Fatt et al., 2003).

In order to reduce the computational cost, the projectile is considered rigid. For this reason, a rigid material model is used for the projectile. The simplified Johnson–Cook plasticity material model (LS-DYNA, 2014) is employed for modeling the material behavior of metals:

$$\sigma_y = (A + B\varepsilon^n)(1 + C \ln \dot{\varepsilon}^*)$$

where $\sigma_y$ is the flow stress, $A$, $B$, $C$, $n$ are material constants, $\varepsilon$ is the equivalent plastic strain and $\dot{\varepsilon}^*$ is the dimensionless plastic strain rate.

We idealize the material behavior of the unidirectional glass-epoxy layers in the fiber-metal laminates by employing the Chang–Chang composite failure model (LS-DYNA, 2014). With the Chang–Chang material model, failure can take place due to matrix cracking, compression or fiber breakage. Specifically, the fiber breakage failure criterion is determined from the following inequality:

$$\left( \frac{\sigma_1}{S_1} \right)^2 + \frac{\tau}{4} > 1,$$

where $\sigma_1$ and $S_1$ are the longitudinal tensile stress and strength, respectively, and $\tau$ is a shearing term which augments each damage mode:

$$\tau = \left( \frac{\tau_{12}^2}{2G_{12}} + \frac{3}{4} \frac{\tau_{12}^4}{S_{12}^2} \right) / \left( \frac{S_{12}^2}{2G_{12}} + \frac{3}{4} \frac{S_{12}^4}{4} \right) ,$$

where $G_{12}$ is the in-plane shear modulus, $\alpha$ is a nonlinear shear stress parameter, $S_{12}$ is the shear strength and $\tau_{12}$ is the shear stress. When inequality (3) is satisfied, the longitudinal and transverse Young’s modulus, the in-plane Poisson’s ratios and $G_{12}$ are set equal to zero.

The matrix cracking failure criterion is determined from the following inequality:

$$\left( \frac{\sigma_2}{S_2} \right)^2 + \frac{\tau}{4} > 1,$$

where $\sigma_2$ and $S_2$ are the transverse tensile stress and strength, respectively. When inequality (5) is satisfied, the transverse Young’s modulus, the in-plane Poisson’s ratios and $G_{12}$ are set equal to zero.

The compression failure criterion is determined from the following inequality:

$$\left( \frac{\sigma_2}{2S_{12}} \right)^2 + \left[ \left( \frac{C_2}{2S_{12}} \right)^2 - 1 \right] \frac{\sigma_2}{C_2} + \frac{\tau}{4} > 1,$$

where $C_2$ is the transverse compressive strength. When inequality (6) is satisfied, the transverse Young’s modulus and the in-plane Poisson’s ratios are set equal to zero.

For the simulation of the perforation of the panels, the panels’ finite elements in way of the projectile must be eroded. In order to achieve this goal, a strain-based erosion criterion is added to the metal and glass-epoxy material models. The employed material properties of glass-epoxy are given in Table I (Hoo Fatt et al., 2003; Yaghoubi and Liaw, 2013).
In order to further reduce the high computational cost of the considered problem, we take advantage of the symmetry of the problem by modeling a quarter of the structure and by applying suitable symmetry boundary conditions to all nodes of the symmetry planes.

Given that the projectile is considered rigid, instead of meshing a solid cylinder, only the exterior surface with 0.8 mm thickness was meshed. The density of the projectile’s material was modified so that the full projectile’s mass remains constant. A fine mesh is used for the projectile in order to represent its geometry accurately, including the radius of its impacting face. With this meshing approach the number of the finite elements needed to model the projectile is minimized and the computational cost is further reduced. In order to verify the accuracy of the hollow projectile model FEM results, complete projectile models were also implemented and solved yielding practically identical results.

The one-point integration has been selected for the SOLID164 elements. For each ballistic impact case we analyze, we verify that the hourglass energy and the sliding energy are small relative to the internal energy, as recommended for an explicit analysis with LS-DYNA (LS-DYNA Support, 2017).

An explicit transient dynamic analysis is employed with geometric and material nonlinearities. The initial velocity of the projectile is predetermined. The duration of our analysis is controlled with a predetermined termination time which allows for complete panel perforation, provided that the initial velocity of the projectile is greater than or equal to the ballistic limit. It is noted that in order to determine the unknown ballistic limit of a specific panel, several trial analyses have to be executed with different initial projectile velocity.

In order to verify the convergence of FEM results, the ballistic limit of the panel, we built two models with increasing in-plane mesh density for each specific case of square panel we analyze. The projectile’s mesh density remains the same for all models. For each model, we obtain the ballistic limit and compare them in order to verify that satisfactory convergence has been achieved. A full structure fine mesh model of a steel-based 3/2-0.5 fiber-metal laminate along with the projectile is depicted in Figure 1.

The fine mesh FEM model of a steel-based 2/1-0.5 fiber-metal laminate and the projectile consists of 45,060 elements and has 728,613 degrees of freedom. These elements correspond to a quarter of the structure. The user defined time step of the simulation is equal to $5 \times 10^{-7}$ s. The duration of the FEM solution with the employed computer is 50 min for complete panel perforation. The computer has four processors at 3.6 GHz and 16 GB RAM.

It is noted that the finite element models of this paper have been validated and used in reference (Bikakis et al., 2017) in order to predict the ballistic impact response of fiber-metal laminates and monolithic metal plates consisting of different aluminum alloys. Figure 2 summarizes the excellent agreement of the numerical calculations and the experimental data from Hoo Fatt et al. (2003) and demonstrates the validity of the implemented finite element modeling procedure.

### 3. Results and discussion

Using the five aforementioned validated finite element models, we study the effect of the mechanical properties of different commercially available steel alloys on the ballistic

<table>
<thead>
<tr>
<th>Table I. Material properties of S2-glass fiber-reinforced epoxy layers</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_{11}$ = 52 GPa (longitudinal Young's modulus)</td>
</tr>
<tr>
<td>$E_{22}$ = 17 GPa (transverse Young's modulus)</td>
</tr>
<tr>
<td>$E_{33}$ = 17 GPa (through-thickness Young's modulus)</td>
</tr>
<tr>
<td>$G_{12}$ = 7 GPa (in-plane shear modulus)</td>
</tr>
<tr>
<td>$G_{23}$ = 7 GPa (out-of-plane shear modulus)</td>
</tr>
<tr>
<td>$G_{13}$ = 7 GPa (out-of-plane shear modulus)</td>
</tr>
<tr>
<td>$\varepsilon_{errosion}$ = 70% (strain erosion limit)</td>
</tr>
</tbody>
</table>
resistance of steel-based fiber-metal laminates and monolithic steel plates. Fiber-metal laminates and monolithic metal plates consisting of 304 stainless steel, 1010, 1080, 4340, A36 steel alloys and DP 590 dual phase steel are examined. These alloys are used in a wide variety of applications such as automotive, ship, aerospace and military structures, buildings, oil rigs, bridges and industrial applications. For this purpose, the material properties of the initial 2024-T3 aluminum alloy of the panels were updated in each validated model in order to represent the analyzed steel alloys. The employed material properties of these alloys are given in Table II (Bikakis et al., 2018). This approach was also implemented by Vo et al. (2013) in order to investigate the influence of aluminum alloys with very different material properties on the low-impulse blast behavior of fiber-metal laminates. It is noted that the ballistic limit convergence along with the desired low levels of hourglass and sliding energy were verified in all of the 30 modeled ballistic impact cases.
In Figure 3, the calculated ballistic limits of the two monolithic plates with 1.6 mm and 3.2 mm thickness are compared. As the thickness of the two examined plates increases, the corresponding ballistic limit is increased for all alloys. It can be observed from Figure 3 that the trend of the ballistic limits vs the panel thickness is similar for the considered alloys. Another observation from this figure is that the 304 stainless steel has greater ballistic resistance than the A36 mild steel and the 2024-T3 aluminum has lower ballistic resistance than the A36 mild steel. Similar pertinent experimental results have been reported by Corran et al. (1983). In that work mild steel, stainless steel and aluminum plates have been tested at several plate thicknesses. They were impacted by blunt cylindrical projectiles in order to determine their ballistic limit. It can be verified for plate thicknesses equal to 1.6 and 3.2 mm that the stainless steel offers the highest whereas the aluminum offers the lowest ballistic resistance among the three materials according to the experiments of Corran et al. as well. It can be observed from Corran et al. (1983) that within this range of plate thickness, the experimental ballistic limit of the three materials varies approximately linearly vs the plate thickness. Consequently, it is very useful to present linear equation trendlines of ballistic limits as a function of the plate thickness for the materials examined here. These approximation equations are given in Figure 3 and are applicable for plate thickness values between 1.6 and 3.2 mm. They can be used when experimental or theoretically calculated ballistic limits are not available for a specific thickness value.

It can be seen from Figure 3 that A36 steel offers higher ballistic limit than 1010 steel. Referring to the 1.6 mm thick plates, the calculated ballistic limits are 165 and 150 m/s, respectively. There are several empirical formulae to predict the critical impact energy required to perforate steel plates subjected to impact loading by projectiles (Corbett et al., 1996). In order to compare the aforementioned ballistic limits, we have applied the Jowett’s formulae (Corbett et al., 1996), which are based on experimental data. These relationships

<table>
<thead>
<tr>
<th>Steel alloy</th>
<th>304 stainless</th>
<th>1010</th>
<th>1080</th>
<th>4340</th>
<th>A36</th>
<th>DP 590</th>
</tr>
</thead>
<tbody>
<tr>
<td>ρ (kg/m³)</td>
<td>7,800</td>
<td>7,870</td>
<td>7,700</td>
<td>7,830</td>
<td>7,850</td>
<td>7,870</td>
</tr>
<tr>
<td>ν</td>
<td>0.265</td>
<td>0.300</td>
<td>0.270</td>
<td>0.290</td>
<td>0.260</td>
<td>0.300</td>
</tr>
<tr>
<td>E (GPa)</td>
<td>200.0</td>
<td>200.0</td>
<td>202.8</td>
<td>200.0</td>
<td>200.0</td>
<td>214.0</td>
</tr>
<tr>
<td>A (MPa)</td>
<td>310.00</td>
<td>367.00</td>
<td>525.00</td>
<td>792.00</td>
<td>286.13</td>
<td>430.00</td>
</tr>
<tr>
<td>B (MPa)</td>
<td>1,000.00</td>
<td>700.00</td>
<td>3,590.00</td>
<td>510.00</td>
<td>500.14</td>
<td>823.60</td>
</tr>
<tr>
<td>n</td>
<td>0.6500</td>
<td>0.9350</td>
<td>0.6677</td>
<td>0.2600</td>
<td>0.2280</td>
<td>0.5071</td>
</tr>
<tr>
<td>C</td>
<td>0.0700</td>
<td>0.0450</td>
<td>0.0290</td>
<td>0.0140</td>
<td>0.0170</td>
<td>0.0171</td>
</tr>
</tbody>
</table>

**Table II.** Material properties and Johnson–Cook constants of steel alloys

**Figure 3.** Ballistic limits and trendlines of monolithic plates consisting of different metal alloys vs the plate thickness.
are suitable for impact on steel plates by short cylindrical projectiles. The minimum energy required for perforation is expressed as a function of the ultimate stress of steel, the diameter of the projectile, the thickness of the target and the span of the target. Jowett's formulae have a specific validity range. Using Jowett's formulae, we have calculated the ballistic energy and the corresponding ballistic velocity for the A36 steel and 1010 steel plates with 1.6 mm thickness. Only these two cases of our study satisfy all restrictions concerning the validity range of Jowett's formulae. Considering UTS values equal to 483 MPa for the A36 steel (AZO MATERIALS, 2017) and 365 MPa for the 1010 steel (AZO MATERIALS, 2017), the empirical ballistic velocities are found to be 151.3 and 131.5 m/s, respectively. Consequently, A36 steel offers higher ballistic limit than 1010 steel according to the empirical formulae as well. Furthermore, the empirical ballistic limits deviate 8.3 and 12.3 percent from their numerical values, respectively. This comparison further demonstrates the validity of the implemented finite element modeling procedure.

In Figure 4, the calculated ballistic limits of the fiber-metal laminates are compared. It is seen from this figure that the ballistic limit of the examined 2/1.0-5, 3/2.0-3 and 3/2.0-5 fiber-metal laminates increases vs the total panel thickness following a similar trend regardless of the monolithic metal constituent. Second-order polynomial trendlines interpolate the experimental and numerical data of the three GLARE 5 plates in Figure 2 very smoothly. Such trendlines are employed in Figure 4 as well. A smooth interpolation of the calculated ballistic limits can be observed in Figure 4. Consequently, it is very useful to present the second-order polynomial equation trendlines of ballistic limits as a function of the panel thickness for the materials examined here. These approximation equations are given in Figure 4 and are applicable for panel thickness values between 1.52 and 2.54 mm. They can be used when experimental or theoretically calculated ballistic limits are not available for a specific thickness value.

It is seen from Figure 4 that the stainless steel based fiber-metal laminates offer higher ballistic resistance than the A36 steel based fiber-metal laminates which in turn have higher ballistic resistance than the GLARE panels. It is also seen that A36 steel based fiber-metal laminates have higher ballistic limit than 1010 steel based fiber-metal laminates. Furthermore, from Figures 3 and 4 it can be verified that the 1010 steel based panels offer higher ballistic limit than the 1080 steel based panels. Consequently, taking also into account the discussed behavior of monolithic plates consisting of 304 stainless steel,

\[
\begin{align*}
\text{A36} & : y = -27.50x^2 + 144.2x + 5.012 \\
\text{1010} & : y = -39.62x^2 + 193.5x - 52.87 \\
\text{4340} & : y = -11.35x^2 + 73.68x + 52.06 \\
\text{DP590} & : y = -16.99x^2 + 90.71x + 37.21 \\
\text{1080} & : y = -16.98x^2 + 85.76x + 44.74 \\
\text{2024-T3} & : y = -21.83x^2 + 107.4x + 18.98
\end{align*}
\]
A36 and 1010 steel, and 2024-T3 aluminum, it is found that the behavior of fiber-metal laminates is substantially affected by the behavior of their metal constituent. This finding means that the impact resistance of fiber-metal laminates can be improved by selection of an impact tolerant metal alloy. Another observation from Figures 3 and 4 is that the 2024-T3 aluminum, the 1080 steel and the DP 590 dual phase steel based panels have comparable relatively low ballistic resistance. From Figures 3 and 4, it is also concluded that the panel thickness is another important design parameter which can be used in order to increase the ballistic limit of fiber-metal laminates and monolithic plates.

Table III summarizes the numerically calculated ballistic limits of the 30 simulated ballistic impact cases on steel-based targets along with the five ballistic limits concerning the impact cases on aluminum based targets. For the cases of aluminum based targets, the experimental ballistic limits are also included in Table III (Hoo Fatt et al., 2003). Furthermore, the aforementioned empirically calculated ballistic limits of A36 steel and 1010 steel plates with 1.6 mm thickness are also included in Table III.

In Figure 5, a representative diagram of the variation of characteristic impact variables vs time is shown. This diagram corresponds to a 1080 steel based fiber-metal laminate with 3/2-0.5 layup subjected to ballistic impact. Specifically, the time histories of the projectile’s displacement, velocity and kinetic energy, symbolized with \( w, u \) and \( E_k \), respectively, are plotted along with the time history of the impact load \( P \) and the energy absorbed by the panel, \( E_{ab} \). The \((w, t)\) and \((u, t)\) curves are directly obtained from the FEM analysis results. The \((E_k, t)\) curve is calculated using the relation \( E_k = 0.5mu^2 \). The \((P, t)\) curve is calculated

<table>
<thead>
<tr>
<th>Layup, thickness</th>
<th>Alloy 304</th>
<th>Alloy A36</th>
<th>Alloy 1010</th>
<th>Alloy 4340</th>
<th>Alloy DP 590</th>
<th>Alloy 1080</th>
<th>Alloy 2024-T3</th>
</tr>
</thead>
<tbody>
<tr>
<td>3/2-0.5</td>
<td>214</td>
<td>194</td>
<td>183</td>
<td>166</td>
<td>158</td>
<td>153</td>
<td>151</td>
</tr>
<tr>
<td>2.54 mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>156</td>
</tr>
<tr>
<td>3/2-0.3</td>
<td>205</td>
<td>181</td>
<td>173</td>
<td>152</td>
<td>149</td>
<td>147</td>
<td>145</td>
</tr>
<tr>
<td>1.93 mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>151</td>
</tr>
<tr>
<td>2/1-0.5</td>
<td>189</td>
<td>161</td>
<td>150</td>
<td>138</td>
<td>136</td>
<td>136</td>
<td>132</td>
</tr>
<tr>
<td>1.52 mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>136</td>
</tr>
<tr>
<td>Plate</td>
<td>179</td>
<td>165</td>
<td>150</td>
<td>142</td>
<td>132</td>
<td>132</td>
<td>131</td>
</tr>
<tr>
<td>1.6 mm</td>
<td>151.3</td>
<td>131.5</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>131</td>
</tr>
<tr>
<td>Plate</td>
<td>251</td>
<td>229</td>
<td>212</td>
<td>213</td>
<td>186</td>
<td>198</td>
<td>191</td>
</tr>
<tr>
<td>3.2 mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td>196</td>
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Table III.
Numerical ballistic limits along with experimental (bold numbers) and empirical (italic numbers) ballistic limits

Figure 5.
Striker-target time history curves of a 1080 steel based 3/2-0.5 fiber-metal laminate subjected to ballistic impact by a cylindrical projectile
with the relation $P = m\gamma$ using the FEM calculated time history of the projectile’s acceleration. The sum of the projectile’s kinetic energy and the panel’s absorbed energy is equal to the total energy of the system which remains constant at the level of the initial kinetic energy of the projectile.

As shown in Figure 5, the impact load is initially equal to zero and increases abruptly as soon as the contact between the colliding bodies begins. As explained later in this section, this abrupt increment of the impact load is due to the contact force caused by the local indentation of the panel under the projectile during the beginning of impact. As soon as the local indentation stage is completed, the impact load drops suddenly, at time $t \approx 0.007$ ms, and the extended panel deformation stage, beyond the projectile’s impacting face area, follows. During this stage, the impact load time history demonstrates noisy variations until the full perforation of the panel. Significant sudden reductions of the impact load at this stage indicate progressive internal damages in the laminate caused by the striker. It is noted that peak sudden contact forces at the beginning of ballistic impact events, concerning target plates perforated by free-flying projectiles, have also been found in the numerical simulations of Borvik et al. (2002), Gama and Gillespie (2011) and Yaghoubi and Liaw (2014); however, none of these articles investigates the impact response of steel fiber-metal laminates.

By observation of Figure 5, it can be seen that the displacement of the projectile is initially equal to zero (for $t = 0$) and starts increasing with decreasing rate due to the panel’s resistance. When the panel is perforated, about 0.37 ms after the beginning of the striker’s motion, its resisting force acting on the projectile becomes equal to zero and the projectile’s displacement increases with constant rate since it now moves with constant velocity according to Newton’s first law.

It is seen from Figure 5 that the striker’s ballistic velocity is initially equal to 153 m/s and decreases abruptly because of the discussed peak impact load applied on the projectile during its contact interaction with the clamped panel. After the local indentation stage, the drop rate of the projectile’s velocity becomes gradually lower as the striker penetrates the target and its stiffness is reduced. The onset and development of plasticity in the metal layers, the delamination between adjacent layers, the failure of composite layers and the erosion of each layer contribute to the reduction of the panel’s stiffness. Eventually the panel is perforated and the projectile then continues to move but with constant low velocity as discussed previously.

It can be observed from Figure 5 that the ballistic kinetic energy of the projectile is initially equal to 165.3 J and is decreased abruptly since it is consumed to deform, cause damage and perforate the panel. After complete perforation the striker’s kinetic energy becomes practically equal to zero. This means that the initial impact energy is consumed during the ballistic impact of the panel. The energy absorbed by the panel follows the opposite trend.

In Figure 6, the $(u, w)$, $(E_k, w)$, $(P, w)$ and $(E_{abs} , w)$ curves corresponding to the time histories of Figure 5 are depicted. Since the projectile is initially (for $t = 0$) located at a distance of 1 mm from the panel, it can be observed from Figure 6 that complete perforation occurs when the projectile is displaced about 14 mm from the position of its initial contact with the panel. Given that the total thickness of the clamped panel is 2.54 mm, it is found that the panel has experienced large lateral deflections and significant membrane stretching before perforation occurs.

Using numerous displacement contours of the projectile and the panel during the beginning of the ballistic impact phenomenon, it has been found that the variation of the peak contact forces at this stage, such as the peak contact force of about 140 kN depicted in Figure 5, coincides with the variation of the local indentation of the panel under the projectile. Specifically, the beginning of the local indentation initiates the sharp increase of
the impact load which is maximized when the local indentation becomes maximum. The beginning of the unloading phase of local indentation initiates the sharp decrease of the impact load which is minimized when the local indentation becomes minimum.

The loading and unloading phases of the local indentation of the panel result in different displacements of the projectile and the target under the projectile throughout their duration. The actual positions of the projectile and the target cannot be determined using a spring-mass model with a single degree of freedom where the projectile’s mass and the target’s effective mass are added to yield the total mass of the mechanical system. At least 2 degrees of freedom, one corresponding to the projectile’s mass and one corresponding to the effective mass of the plate, are needed in order to capture the local indentation phases and the associated peak contact force along with other characteristic impact variables satisfactorily. A mechanical system with a single degree of freedom may be used in order to simulate the impact event satisfactorily after the beginning, when the local indentation phases of the panel under the projectile are completed, until the perforation of the panel.

In Figures 7 and 8, representative displacement contours of the projectile and the panel are shown at characteristic stages of the ballistic impact phenomenon. For these figures, we have used LS-DYNA reflect model command. The figures correspond to a GLARE 5-2/1-0.5 panel subjected to normal impact by the cylindrical projectile with the ballistic limit velocity. In these figures, the displacement contours show the resultant displacement vectors in m. Figure 7 illustrates a 3-D view of the relatively large displacements near the center of the panel where the striker is acting, after the beginning of the impact phenomenon and before the perforation of the panel. Figure 8 illustrates a 3-D view of the displacements of the panel during its perforation by the projectile.

4. Conclusions
This paper deals with the transient response of square clamped fiber-metal laminates and monolithic metal plates subjected to central normal ballistic impact by a rigid flat-faced cylindrical projectile. ANSYS LS-DYNA explicit nonlinear finite element software is employed in order to simulate the ballistic impact phenomenon.

Using the five validated finite element models of GLARE 5-2/1-0.5, GLARE 5-3/2-0.3 and GLARE 5-3/2-0.5 fiber-metal laminates and 2024-T3 monolithic aluminum plates with 1.6 and 3.2 mm thickness, the ballistic impact response of steel-based fiber-metal laminates and monolithic plates is studied. Six steel alloys are examined, namely 304 stainless steel, 1010, 1080, 4340, A36 steel and DP 590 dual phase steel.
Steel fiber-metal laminates and plates

Figure 7. Resultant displacement vectors in m of a GLARE 5-2/1-0.5 fiber-metal laminate subjected to ballistic impact by a cylindrical projectile before the perforation of the target.

Figure 8. Resultant displacement vectors in m of a GLARE 5-2/1-0.5 fiber-metal laminate subjected to ballistic impact by a cylindrical projectile during the perforation of the target.
It is found that the ballistic limits of the panels can be substantially affected by the constituent metal alloy. The stainless steel based panels offer the highest ballistic resistance followed by the A36 steel based panels which in turn have higher ballistic resistance than the 2024-T3 aluminum based panels. The A36 steel based panels have higher ballistic limit than the 1010 steel based panels which in turn have higher ballistic limit than the 1080 steel based panels. The 2024-T3 aluminum, the 1080 steel and the DP 590 dual phase steel based panels have comparable relatively low ballistic resistance.

It is revealed that during the beginning of the considered impact phenomenon the variations of characteristic impact variables cannot be captured using a spring-mass model with a single degree of freedom. At least 2 degrees of freedom are needed for the mechanical system.

References


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Piston-ring performance: limitations from cavitation and friction

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Abstract

Purpose – Cavitation in piston-ring lubrication is studied as part of the performance of piston-ring assemblies. Cavitation degrades performance in engineering applications and its effect is that it alters the oil film pressure, generated at the converging-diverging wedge of the interface. Studies tried to shed light to the phenomenon of cavitation and compare it with cavities that have been identified in bearings. The paper aims to discuss this issue.

Design/methodology/approach – Lubricant formulations were used for parametric study of oil film thickness (OFT) and friction providing the OFT throughout the stroke and LIF for OFT point measurements. Lubricant formulation affects cavitation appearance and behaviour when fully developed.

Findings – Cavitation affects the ring load carrying capacity. Different forms of cavitation were identified and their shape and size (length and width) is dictated from reciprocating speed and viscosity of the lubricant. A clear picture is given from both techniques and friction results give quantifiable data in terms of the effect in wear and cavitation, depending on the lubricant properties.

Research limitations/implications – Engine results are limited due to manufacturing difficulties of visualisation windows and oil starvation. Therefore, full stroke length sized windows were not an option and motoring tests were implemented due to materials limitations (adhesive and quartz windows). Lubricant manufacturer has to give data regarding the chemistry of the lubricants.

Originality/value – The contribution of cavitation in piston-ring lubrication OFT, friction measurements and lubricant parameters that try to shed light to the different forms of cavitation. A link between viscosity, cavitation, shear thinning properties, OFT and friction is given.

Keywords Tribology, Cavitation, Visualization, Friction and oil film thickness measurements, Laser induced fluorescence measurements, Piston-ring lubrication

Paper type Research paper

1. Introduction

The formation of cavities and their subsequent disposition affects the pressure generated in the continuous thin lubricant film, and hence, any integrated quantities such as the load capacity of bearings. Understanding of the flow in a simplified test rig rather than in an ordinary ring pack provides information of practical importance and leads to characterisation of the lubricant transport phenomena in the piston-ring pack of production engines (Dellis, 2005).

Cavitation has long been recognised to degrade performances in most engineering applications and its effect in piston-ring lubrication is that it alters the oil film pressure profile, generated at the converging-diverging wedge of the piston-ring and liner interface. An area of the piston-ring surface is void, corresponds to subatmospheric pressures and thus, the piston ring load capacity is altered. The load carrying capacity of the interface is the integral of the pressure distribution, the lubricant film thickness, the friction force and the lubricant flow rate (Prist et al., 2000).

The viscosity of mineral lubricants (non-shear thinning) would result in greater friction power loss when compared with the shear thinning ones. The viscosity of the lubricant at the piston-ring cylinder liner interface varies during the cycle as the shear rate experienced by it changes (Rastogi and Gupta, 1992).

The importance of studying cavitation lies in the fact that it can cause tremendous damage to fluid handling machinery. Cavitation is also related to the eroding of metals and
to the wear of components which can dramatically shorten the component’s life. Cavitation wear, which is a fluid to surface type of wear, appears under the formation of vaporous cavitation. The lubricant is exposed to tensile stresses that cause the lubricant to boil and then exposed to compressive stresses that cause the vapour bubbles to collapse, producing a mechanical shock and microjets that simultaneously attack the nearby surfaces. Shockwaves can also erode the piston-ring/cylinder liner interface. Cavitation wear is also known as cavitation erosion, vaporous cavitation, cavitation pitting, cavitation fatigue or liquid impact erosion. The small volumes of liquid, when the cavitation bubble collapses force small volumes of liquid towards the interface, which, in turn, results into many small high pressure and high temperature spots. If the force applied to these spots exceeds the strength of the material, the component will fail (Vasilakos, 2017). Bubble formation is caused by the release of dissolved gas from the liquid where it sustains a near zero or negative pressure. When a bubble collapses against a solid surface, very high stresses reaching 0.5 GPa in some cases are generated and this will cause wear (Stachowiak and Batchelor, 2000).

Cavitation affects squeeze film forces by the formation of compressible bubbles in an otherwise incompressible lubricant. The appearance of bubbles is due to the much slower rate of bubble dissolution as compared to the rate of bubble formation. Eventually load capacity is reduced when compared to the values calculated by hydrodynamic theory assuming no cavitation effects (Haber and Etsion, 1985; Parkins and Stanley, 1982), as cavitation affects the load capacity in an oscillating squeeze film. A significant amount of bubble formation in the lubricant film, drops load capacity accordingly (Stachowiak and Batchelor, 2000).

During the reciprocating motion of the piston-ring/cylinder liner interface, the contact area is much larger and the friction force is much higher at the reverse points. The stress field is changed via the load variation throughout the stroke, which inevitably results into fatigue wear in the contact area. In tested areas, cracking that leads to spalling and particle pullout was observed in the sliding contacts, where grooving, polishing, plastic deformation, micro-chipping and fatigue was observed. It was concluded that the wear rate strongly depends on material properties and the combination of the ring-liner pair (Ye and Cheng, 1996).

Under certain circumstances the risk of vaporous cavitation occurring (which is the case in the converging-diverging wedge of the piston-ring/cylinder liner interface (Dellis and Arcoumanis, 2004) will effectively contribute to increased wear. Higher viscosity oils provide adequate wear protection while lower viscosity oils reduce frictional losses and increase efficiency (Stachowiak and Batchelor, 2000). The cavity length data are linked to lubricant properties (Dellis P.S., 2018). The addition of viscoelastic additives to lubricants raises the load capacity and reduces any wear that might occur (Stachowiak and Batchelor, 2000).

Since the load carrying capacity of the piston ring is affected due to cavitation (Dellis and Arcoumanis, 2004, 2013; Dellis, 2017), the lubricant behaviour at the interface between the piston ring and liner should be optimised for addressing the cavitation issues without compromising the load capacity of the piston ring (Dellis P.S., 2019).

Vladescu et al. (2015) pointed out that with the LIF (laser induced fluorescence) system, zones of cavitation were observed. LIF measurements (Pyke, 2000; Dellis, 2005) identified cavitation under the piston ring and in Dellis P.S. (2019) the change of cavitation pattern is stressed out for the different applied loads and the chemical formulation of the lubricants. The researchers concluded that the results highlight the possibility of using the set-up of the simulating rig to study the effect of oil properties on cavitation behaviour.

The viscosity of different oils varies at different rates with temperature. More viscous oils would theoretically give better performance but more power is required for these oils to be sheared. Higher temperatures lead to lower viscosity. Therefore, lubricant film
thickness and load carrying capacity of the oil film decreases significantly with increasing
temperature (Akalin and Newaz, 2001).

The power losses are higher and more heat is generated resulting in a substantial
increase in the temperature of the surfaces in contact, which may lead to component failure
(Stachowiak and Batchelor, 2000). Liquids can be heated by intense viscous shearing. Anti-wear additives are introduced to protect the contacting surfaces at high temperatures above the range of the effectiveness of absorption of boundary agents at the surfaces of the contacting interface (Stachowiak and Batchelor, 2000). The loss of viscosity due to heating causes a significant loss of load capacity and that fact needs to be controlled. In addition to heating, elastic distortion (Hertzian contact) also diminishes load capacity.

The prevailing trends towards higher speeds and loads have heightened the need for
accurate predictions of load capacity. In piston-ring modelling, the proper determination
of the reformation boundary is critical to accurately determining the load capacity of the piston ring (Yang and Keith, 1995), while the pressure contribution from the reformed lubricant film and the trailing edge pressure may contribute substantially to the overall load support (Ting, 1980). Hydrodynamic action in the converging section of the ring is responsible for generating the load supporting pressure in the lubricant film. The increased surface separation in the case of hydrodynamic lubrication decreases the amount of asperity-asperity contact and this is evident as a lowering of the friction spike close to the ends of the stroke (Bolander et al., 2005). Under conditions of lubricant starvation, hydrodynamic lubrication remains ineffective and if the limits are exceeded by excessive load or insufficient speed, there is the risk of a large rise in friction. Lubrication affects friction reduction by the formation of low shear strength interface between the opposing surfaces (Stachowiak and Batchelor, 2000).

When a transition temperature is exceeded, damage to the adsorbate film is more rapid
than film repair, so that the adsorption film is progressively removed. High friction and wear is then inevitable. A thin layer formed by adsorption of the lubricant additives on the worn surface makes the contact of the interface identical to that of dry surfaces. The small contact area between asperities (smaller than the apparent) is followed by a large temperature rise, as the frictional energy and the resulting heat become highly concentrated. This concentration of frictional energy controls friction and wear, where local temperatures can rise to very high values with the minimal input of frictional energy. Temperatures, in turn, affect oil viscosity, load capacity and the formation of cavities (Stachowiak and Batchelor, 2000).

Viscosity has a damping effect on cavity growth and collapse and as a consequence, the reduced viscosity of shear thinning lubricants should lead to large sized cavities. The size of the cavitation zone in shear thinning lubricants is significantly larger than for Newtonian lubricants and this is derived as a result of the lower tensile strength of shear thinning lubricants (Rastogi and Gupta, 1992). Shear stress has an effect on viscosity. The higher the shear stress, the lower the viscosity. Viscosity has a major impact on flow separation points (effect on flow structure), which, in turn, has an impact on the pressure distribution and as a consequence, cavitation is also affected (Joseph, 1998). The lubricant properties play the primary role in the formation and collapse of cavities and influence cavitation behaviour indirectly by introducing changes in the film thickness. Viscosity inhibits cavity growth and collapse and eventually the effect of oil film thickness (OFT) on cavitation behaviour must also be taken into account while assessing the influence of viscosity on cavitation damage (Dellis P.S., 2019).

Viscosity can be directly linked to OFT and the lubricant with the lowest viscosity would have the smallest OFT. That applies for the width and the cavity length as they appear at the oil film visualisation results. The lower the kinematic viscosity, the higher the cavitation area, the length and the width of the cavities (Vasilakos, 2017), affecting in parallel the “effective” ring width (Dellis and Arcoumanis, 2013; Dellis, 2005).
The percentage of mixed lubrication is reduced at lower engine speeds when a large curvature piston ring is used, as hydrodynamic lubrication is promoted and load carrying capacity is increased. Lower power losses were recorded for the larger ring width and the results were verified in Dellis (2010) and that applies for total friction power losses (over the whole cycle) as well as for the region where mixed and boundary lubrication takes place. However, larger ring width also provides larger shearing area.

Mixed lubrication conditions occurring in the vicinity of either the top or bottom dead centre, qualitatively agrees with the wear pattern observed on cylinder liners (higher wear at the top and bottom of the stroke) (Jeng, 1992). Very high values of the boundary friction force could lead to wear. Boundary film is not effective to resist scuffing as the change of friction coefficient will be directly related to the load carrying capacity, when the working temperature is kept constant. The scuffing is dependent on the competition between heat generation of plastic deformation and heat dissipation at the locally deformed area (Wan et al., 2017).

Therefore the selection of appropriate exit boundary conditions which are applicable to the contact in question is paramount for the prediction of load carrying capacity and friction (Tang et al., 2015).

In previous experiments it was noted that for a ring-liner conjunction a limited cavitation region does survive through the dead centre inlet reversal (Arcoumanis et al., 1995; Dellis and Arcoumanis, 2004). The pre-reversal cavitation region is sealed off by the lubricant film and forms a bubble. However, while located at the inlet it depletes the available lubricant and leads to starvation. In an engine even if the first piston ring could be fully flooded, the consequent piston ring is operating under conditions of lubricant starvation and the pressures at the leading and trailing edges of the contact are different (Chong et al., 2011). Due to lower contact pressure, the pre-reversal cavitation is larger and as a consequence the resulting bubble takes much longer to be fully absorbed into the lubricant film.

The lubricant reformation is dependent on squeeze film effect. Therefore, when this is significant, the lubricant reforms earlier. There is a direct correlation between film thickness and friction force magnitude (Chong et al., 2011).

Rahnejat et al. (2006) showed that for a laser textured liner, improvements by up to 4.5 per cent could be observed in overall friction measurements, compared to the standard liner, while Costa and Hutchings (2007) argued that surface texture can increase the minimum OFT in the reciprocating contact. Vladescu et al. (2015) presented results that suggest the increase in full film friction close to surface texture may be due to the interaction of pockets with the cavitated region. Visualisation results confirm that the number of cavitation streamers in full hydrodynamic lubrication increases as the film thickness decreases. Similar findings were made by Dowson and Taylor (1979) and Taylor (1974) and it was suggested that the effect of oil properties on cavitation behaviour should be studied. With microtextured patterns greater load support and thicker films were achieved. It was also found that if grooves are longer than the contact zone in the sliding direction, high pressure oil will leave the pockets resulting in the collapse of the oil film. Eventually, after reversal this will prevent the building of oil film and hence, reduce the OFT throughout the stroke causing a higher shear rate in the film and increased asperity contact. Friction losses are increased, load support capability is reduced and higher wear rates are evident. In total, pockets close to the reversal are beneficial in reducing friction by helping to build the oil film shortly before and after the reversal points. The effect of surface texture is to boost lubricant entrainment. In mixed and boundary lubrication regimes, the cavitated region may reduce the lubricant entrainment after reversal and friction will increase. The presence of pockets machined on the liner surface or custom made topography applied on the ring or liner, bring oil into the cavitated region and this effect can decrease boundary friction without
increasing oil consumption (Ryk et al., 2002; Rahnejat et al., 2006). At the same time the elastic deformation of the component in contact region should be considered because it is possible that lubricant is being squeezed out of the pockets.

Cavitation effects in OFT have been presented in previous work (Dellis P., 2018). In this paper a link is trying to be established between friction and cavitation after characterizing the OFT. OFT is measured and the effect of different oil properties and temperature are considered for friction and OFT results. A link between viscosity, cavitation, shear thinning properties, OFT and friction is given.

It is therefore, a matter of great significance to study the effect of oil properties and their subsequent influence on cavitation, whether it is a large cavitation area at the interface of the ring-liner assembly or cavity-shape details regarding the size (width and length) of the cavitation forms at oil film rupture.

2. Experimental set-up
The study was conducted in two experimental test rigs: a single-ring test rig that simulates the piston-liner movement but the motion is inversed, i.e. it consists of a steady piston-ring section of 5 mm width that is fixed under a flat liner section and a single-cylinder Lister–Petter PHW1 diesel engine that was used for motoring experiments. The liner as well as the cylinder block was modified to accommodate glass window sections for the visualisation experiments.

2.1 Single-ring test rig
The single-ring test rig itself is capable of providing valuable information about piston-ring lubrication under simplified conditions, by eliminating the in-cylinder complications resulting from ring dynamics, thermal and elastic deformation of the rings, starved lubrication, circumferential variations, lubricant degradation and blow-by. Easy access to the ring-liner interface is its basic characteristic and thus, a better understanding of the lubrication and the lubricant characteristics can be achieved through a series of measurements from sensors that are mounted in this experimental set-up. In this particular simplified experimental set-up, the liner and the piston-ring movements are inversed, i.e. it consists of a steady piston-ring section and a reciprocating liner.

OFT is measured both by a capacitance sensor (electrical method) and an LIF sensor that is mounted within the liner body and reciprocates according to the testing conditions set. Friction is measured via a force measurement sensor mounted outside the oil bath so that total axial friction can be recorded. It is very sensitive to small displacement of the ring assembly caused by the axial friction as the liner reciprocates. This movement results in tension or compression of the quartz transducer. Friction close to the dead centres where peaks were recorded have been validated by the capacitance results (thinnest oil film corresponds to boundary lubrication conditions).

The capacitance method is a very popular technique for measuring OFT. Hamilton and Moore (1974) measured successfully the lubricant film thickness by using capacitance transducers mounted in the cylinder liner. Films up to 10 μm could be measured, above this changes in capacitance were too small to be determined accurately. Grice and Sherrington (1993) investigated cavitation by comparing the output of several other probes implanted at various points in the liner with the measured ring profiles. Takiguchi et al. (2000) used the capacitance method to measure the OFT in the 4th cylinder of a six cylinder engine and it was found that the oil film in the top ring is thinner than other subsequent films. Pyke (2000) and Dellis (2005) verified theoretical results with extensive experimental parametric OFT measurements using the capacitance technique described below, in the single-ring test rig. The experiments consisted of measurements of the OFT for different lubricants with varying physical and chemical properties under variable conditions of speed, load and temperature.
In this parallel plate capacitor, the capacitance is directly proportional to the surface area of the parallel plates and inversely proportional to their separation, which is around 30 microns. As the liner accelerates away from the top dead centre, the lubricant film begins to develop. At this point and close to the dead centre, asperity interaction between the surfaces remains significant but the squeeze film effect also takes place resulting in beneficial load support, as it is supported partly by the lubricant present in the contact. This region corresponds to the mixed lubrication regime and as velocity increases, the surfaces separate, asperity interaction decreases and the regime is full film-hydrodynamic. From near mid-stroke to bottom dead centre, the opposite happens and the film from hydrodynamic turns to small thickness as the velocity of the liner decreases. Then due to low velocity the surfaces cannot separate and the lubrication returns to mixed lubrication regime, the developed forms of cavities diminish into bubbles and the effect of the squeeze film takes place at the other end of the stroke (Bolander et al., 2005). The minimum OFT as measured by the capacitance sensor is not symmetric, but the point of absolute minimum is shifted a few degrees from TDC and BDC due to the squeeze film effect. The transition of load support from asperity contact to hydrodynamic lubrication is evidenced by the peak oil film pressures (Dellis P., 2018) and peak friction appear. At these lower velocities, the hydrodynamic action is not sufficient to sustain the thick film and the lubricant begins to squeeze out of the contact as the surfaces move closer. The pressure generated through this squeezing motion shifts the profile towards the centre line. The squeeze pressure is independent of the asperity contact pressure that begins to increase as the film continues to drop (Stachowiak and Batchelor, 2000). The load supporting pressure is due to hydrodynamic lubrication at the converging wedge of the piston-ring/liner interface. Increased surface separation decreases asperity-asperity contact, in boundary lubrication conditions, which is evident as a lowering of the friction spike in the friction signal as it is acquired throughout the stroke.

A second liner section was manufactured that incorporates an ultra-miniature pressure transducer providing valuable oil film pressure data that can be accompanied by visualisation experiments that were carried out with another specially manufactured liner. This liner, which consists of a glass section, enabled visualisation of the oil film at the ring-liner interface that helped in studying further cavitation and extracting useful results, after the initial evidence of cavitation from the LIF measurements. In Figure 1(a) the glass liner section is shown and in Figure 1(b) the piston ring section set-up for the experiments, the oil jets, the friction and the capacitance sensor. Figure 1(c) shows a schematic of the experimental set-up that speed, load, temperature and stroke length can all be adjusted (Dellis, 2005; Dellis and Arcoumanis, 2004).

2.1.1 Single-ring test rig results. The evaluation of pressure measurements and visualisation tests showed clear forms of cavitation at the interface between the piston ring and the liner surface. The following schematic shows different forms of cavities that start appearing at the surface of the piston-ring specimen and after developing in size as fern-shaped cavities after the dead centres of the stroke, eventually they turn into fissure shaped cavities and then string cavities that do not reach the trailing edge of the piston ring. Close to mid-stroke where the liner reaches its maximum speed due to its sinusoidal movement, the string cavities reach the trailing edge of the piston ring and the cavitation becomes open form instead of closed that was before. According to Figure 2 the stages of cavitation initiation and development at the beginning of the stroke can be seen.

From the capacitance signal measurements shown in Figure 3, it was evidenced that there is a hump in the minimum OFT measurement, which is happening close to mid-stroke and according to the accompanying figures from visualisation experiments, it was verified that this hump is caused by cavitation stages transition (Dellis and Arcoumanis, 2004).

When the cavitation strings do not reach the trailing edge of the ring, the pressure in the closed form cavity is subatmospheric (Figure 4(a)–(c)). When the transition from this cavitating
condition to the consequent open form happens, there is this change-hump in the minimum OFT measurement as seen in the red highlighted areas of the capacitance signal (Figure 3) and eventually the negative oil film pressure transforms into atmospheric (Figure 4(d)–(f)).

What should be discussed further on is the eventual effect of cavitation on friction measurements. First, a friction signal is presented, so that a complete picture of the types of lubrication and its effects can be analysed. A typical friction signal is in Figure 5.

Different lubrication regimes are identified for the downstroke friction signal at high load for three different speeds. The friction force at contact (BDC, TDC, top-bottom dead centre) is due to asperity interaction and the lubrication regime is boundary as already discussed in Section 2.1. Next to the boundary lubrication region, there is a mixed region where the frictional force due to asperity interaction is much larger than the viscous losses. This results in a point of minimum $C_f$ (friction coefficient) as the ring makes a transition from mixed to full film lubrication (Bolander et al., 2005) and the hydrodynamic region where all frictional losses are due to viscous drag. According to Bolander et al. (2005) and Dellis (2010), it is known that the piston ring encounters the entire range of lubrication

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**Figure 1.** The simplified single-ring test rig

Notes: (a) The glass liner section; (b) close-up of the piston-ring specimen; (c) schematic of the experimental set-up

Source: (b) Dellis (2005)
regimes through each stroke (i.e. boundary lubrication, mixed lubrication and elastohydrodynamic lubrication). Thus, an investigation into frictional losses at the piston-ring cylinder liner contact must take into account the transitions from a state of full film lubrication to boundary lubrication.

Extensive parametric study showed that the friction spikes are dependent upon piston-ring curvature, speed, load and temperature conditions, and high temperature high shear (HTHS) viscosity of the lubricant (Dellis, 2010). Power losses were calculated for the different test cases, for the whole stroke and for the boundary lubrication region so that data could be compared and useful results derived from the parametric study to be evaluated. Figure 6 shows the effect of HTHS viscosity on the friction peaks for different lubricants tested at BDC (bottom dead centre).

Each group of oils (oil properties in Table I) in Figure 6 has similar HTHS viscosity. Furthermore, in Figure 6 one can see the effect of viscosity on friction signals at the dead centre of the stroke. This parametric study showed that at 33°C, the oil with the lowest V₄₀
oil 4C, had the highest friction peak and the oil with the highest viscosity, 1A, had the lowest friction peak (at boundary lubrication conditions) (Dellis, 2010).

For the same oil, high temperature testing showed high friction results for the boundary and mixed lubrication region, whereas in the hydrodynamic region, the results showed that there is no significant change in the friction force value.

2.2 Single-cylinder diesel Lister Petter engine

A single-cylinder Lister–Petter PHW1 engine that was used for visualisation imaging in motoring tests was modified in the single-cylinder engine block and liner to accommodate
the experiments. The main purpose of these modifications was to capture the same type of images, as in the case of the single-ring test rig, and results concerning the initiation and development of cavitation to be extracted for the respective engine experiments. The visualised area, though, was not covering the whole of the stroke length and visualisation with a CCD camera was depending on the lighting provided by a flash lamp with fibres to guide the light, depending on the proximity of the camera to the engine block and the position of the camera along the stroke length (Dellis, 2005, 2017) (Figure 7).

### Table I. Tested oil properties

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<th>Blend code</th>
<th>Grade (SAE)</th>
<th>HTHS (mPas)</th>
<th>V₄₀ (cSt)</th>
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</tr>
<tr>
<td>7B</td>
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<td>3.64</td>
<td>66.82</td>
<td>187</td>
<td>1st</td>
</tr>
<tr>
<td>3B</td>
<td>0W30</td>
<td>3.30</td>
<td>68.93</td>
<td>182</td>
<td>1st</td>
</tr>
<tr>
<td>17C</td>
<td>0W30</td>
<td>3.39</td>
<td>67.78</td>
<td>183</td>
<td>1st</td>
</tr>
<tr>
<td>4C</td>
<td>0W30</td>
<td>2.95</td>
<td>52.90</td>
<td>174</td>
<td>2nd</td>
</tr>
<tr>
<td>32A</td>
<td>0W30</td>
<td>3.35</td>
<td>65.97</td>
<td>187</td>
<td>1st</td>
</tr>
</tbody>
</table>
2.2.1 Single-cylinder diesel engine test results. Images acquisition were not as clear as in the case of the single-ring test rig, but similar forms of cavities were visualised and as a result, a hypothesis was made for the possible cavitation stages of the engine case, for the motoring tests. Figure 8(a) shows the possible cavitation stages, combined with possible

![Possible Cavitation Stages](image)

**Notes:** (a) Possible forms of cavities on the surface of the top compression ring for the engine experiments; (b) visualisation experiments with high magnification of the top ring

**Source:** Dellis (2017)
respective pressure measurements. Figure 8(b) shows an image of the cavitation (string form) on the surface of the top compression piston ring. Proper magnification enabled the visualisation of these cavities (Dellis, 2005, 2017).

Images taken at 45 mm from TDC at a recording rate of 15,000 frames per second showed oil accumulated on the second land travelling axially in the opposite piston direction and entering the converging wedge of the second compression ring. Additionally, oil bubbles and droplets forming a ligament on the first piston land and travelling upwards to the top compression ring were identified. It is assumed that this oil film is driven by inertia forces and depends on the volume of oil available on the piston land (Dhunput, 2009).

To capture the transient cavitation structures the recording rate was set to 30,000 frames per second. The fern-shaped structures or the cavitation fissures and strings were not captured in contrast to the test rig. It is thought that engine cavitation structures, during their development are much smaller and diminish quickly compared to the test rig. Lubricant starvation should also be considered instead of the fully flooded lubrication conditions in the test rig experiments. It was, however, observed, that oil film enters the converging wedge of the piston ring and later on in the stroke it exits the diverging wedge as bubbles. Then bubbles coalesce into an oil puddle growing in size and moving towards the first land. It is suggested that the oil film cavitates and then disintegrates into bubbles. Oil droplets and mist can initiate a larger oil transport towards TDC (Dhunput, 2009).

3. Discussion

Quickly after the dead centre, a new cavitating region forms at the trailing edge of the contact. Although these two cavitating regions only coexist for a brief period, together with a very low entrainment motion and high contact loads, lead to thinner films and higher friction forces as shown in the respective figures.

This region which is unwetted by the lubricant and is exposed to outlet gas pressure reduces the required hydrodynamic load capacity from the lubricant film. At this stage, we see in the following schematic (Figure 9) the reduction or increase of the “effective” hydrodynamic film area for different operating conditions of speed and load. The higher the speed, the less support from the hydrodynamic film is provided. The higher the load, the greater hydrodynamic support exists at the piston-ring liner interface (Dellis, 2005; Dellis P.S., 2019).

Thus, there are extensive cavitating regions which contribute no radial support but generate frictional losses. The load capacity of the piston ring is affected by the cavitating region, which is different in size because of the geometry of the piston rings, but it is not

![Figure 9. Speed and load effect on the “effective” piston-ring width](source: Dellis (2005))
clear whether cavitation development at the beginning of the stroke plays a significant role in the measured friction peaks that correspond to boundary lubrication conditions and cavitation initiation with the fern-shaped or fissure shaped cavities (Dellis, 2005).

The “effective” piston-ring width, which is the corresponding piston-ring width for the oil film pressure curve obtained, is changing for each measurement. Figure 9 is providing an insight of the “effective” piston-ring width and how does it change with varying speed and load (Dellis, 2005; Dellis P.S., 2019).

The “effective” ring width is altered as it gets wider by a small margin while load increases. Initially, in the pressure testing results for different piston rings, the slight variation of the mid-stroke setting of the liner relative to the ring was blamed for the pressure peaks shift that was noticed for the different specimens. Without this reason being excluded as it was possible to set the stroke length only relative to marks and a reference stroke measurement point, Figure 9 provides an additional explanation (Dellis, 2005; Dellis and Arcoumanis, 2013). It is important to note that the “effective” ring width is varied due to the different ring curvature and it is also necessary to stress the fact that the point where the oil film attaches to the piston ring changes as well due to the operating conditions. So, for the same speed and load, different oil film pressure peaks crank angles are obtained for ring specimens with different curvature (Dellis, 2005).

The lubricant film thickness has a major effect on cavitation behaviour. Fluid properties play the primary role in the formation and collapse of cavities. Shear thinning: results in a larger cavitation zone while it simultaneously reduces bubble collapse; and influences cavity growth and collapse by introducing differences in the variation of OFT with time. Oil viscosity and OFT are characteristics that affect friction losses (Rastogi and Gupta, 1992). Higher viscosity oils provide adequate wear protection but movement creates more resistance and high frictional losses when measured for the total stroke length (Dellis, 2010). Lower viscosity oils reduce frictional losses and increase efficiency.

Figure 10 shows the effect of temperature in OFT for a specific lubricant (oil 3B). As temperature rises, viscosity is getting lower, depending on the shear thinning properties of the multigrade lubricant, and OFT is getting low, respectively. The effect of lubricant chemistry and in this case of viscous modifiers can alter the OFT with changes in viscosity as temperature rises, or change the overall lubricant properties when compared to another similar SAE grade lubricant but without the addition of the viscous modifier.

As one can notice in Figure 11, the viscosity changes dramatically from 35 to 50°C and afterwards it decreases again but in a much smoother way. OFT decreases 46.58 per cent whereas viscosity decreases 43.35 per cent from 35 to 50°C. It is concluded that OFT is strongly viscosity dependent. In terms of cavitation behaviour, recent experiments showed that the lower area of cavitation is noticed in the lubricant with the higher kinematic viscosity (Vasilakos, 2017) and it is a good indicator to whether lubricants of the same base

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**Figure 10.** Effect of temperature in oil film thickness for oil 3B
group would be more prone to cavitation. The addition of viscous modifier in the lubricants
tested, showed in images less cavities but of a greater length that cover a greater area of the
piston ring as they develop much sooner to finger cavities (Vasilakos, 2017).

The experimental results of the temperature effect in MOFT measurements are affected
by the temperature drift of the liner itself as it is heated up by the hot oil flow. Thermal
expansion of the plate on top of the capacitance transducer is expected. Figure 12 shows the
temperature effect on the offset of the error curve which is acquired at very low speed.
The error variation due to temperature drift does not allow for determining the minimum
of the MOFT curve, i.e. what is the MOFT when the flow reverses but it is assumed that it is
around 0.3 microns in average.

This set of experiments was made so that the absolute value of the minimum OFT could
be directly compared with that of the different oils used. The effect of high temperature is
taken into account for these comparisons and graphs are presented in Figures 13 and 14 for
all the oils tested at high temperatures (50°C – 70°C) (Dellis, 2005).

In Figures 13 and 14 testing showed that oil 2A had the thicker film with temperature
variation. Oil 2A has the highest viscosity at all temperatures. The fact that oil 2A produces
a thicker OFT is more pronounced at 50°C tests. The properties of the tested oils at different
temperatures are in Table II (Dellis, 2005).

The viscosity index (VI) effect in OFT is shown in Figure 15. The following graph shows
the increase in OFT as the VI increases. The results are for the same speed, 400 rpm, no
external load (1,159 N/m), 40°C and similar grade oils 0W–30 but with different additives
that affect the VI but all of them within a slight margin.

Friction peaks data are acquired for high temperature testing in Figure 16. The following
test provides evidence to the high friction peaks noticed when temperature rises.
The following set of oils has been tested at maximum load (3,371 N/m).
Table II.

<table>
<thead>
<tr>
<th>VI and Viscosity for the different oils at various temperatures, used for the effect of temperature on MOFT testing</th>
<th>Oil type (castrol code)</th>
<th>002A/02</th>
<th>003B</th>
<th>005A/02</th>
<th>006E/02</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity (cSt)</td>
<td>50°C</td>
<td>64.34</td>
<td>46.61</td>
<td>21.49</td>
<td>47.13</td>
</tr>
<tr>
<td>60°C</td>
<td>45.76</td>
<td>33.65</td>
<td>15.72</td>
<td>34.55</td>
<td></td>
</tr>
<tr>
<td>70°C</td>
<td>33.46</td>
<td>25.09</td>
<td>12.60</td>
<td>26.29</td>
<td></td>
</tr>
<tr>
<td>VI</td>
<td>160</td>
<td>182</td>
<td>146</td>
<td>196</td>
<td></td>
</tr>
</tbody>
</table>

Figure 13.
MOFT variation, 400 rpm, 50°C for oils 3B, 6E, 5A, 2A

Figure 14.
MOFT variation, 400 rpm, 70°C, for oils 3B, 6E, 5A, 2A

Figure 15.
Viscosity index effect in oil film thickness

Source: Dellis (2005)
The above set of experiments (Figures 10 and 16) gives qualitative insight into the effect of viscosity change due to temperature in friction and OFT for the same lubricant. Another set of LIF experiments is showing the OFT and the film breakage due to cavitation under the piston ring at mid-stroke. These experiments show the OFT under the piston ring for the different oil types that were used – the same also applies for the capacitance measurements in Figures 13 and 14. The results show that OFT has a marginal variation. At 300 rpm for all tested oils, high load (2,823 N/m), oil 17C had the lowest thickness. From the results shown in Figure 17, it looks that the MOFT might contribute to the behaviour of the oil film in terms of cavitation development, in a sense that different size of string cavities, width and length, appear in the visualisation tests, depending on the chemistry of the lubricant. The LIF tests at 300 rpm (Figure 17) showed that for oil 7B there is an oil film breakage and for oil 1A a “mixed” lubrication region that consists of oil film streamer and cavities. This is the result of the LIF point measuring technique.

Figure 18(a)–(c) shows the error analysis – standard error of mean for 50 cycles, for oil 3B, at 400 rpm speed and 971 N/m load. In these figures the results are shown for parts of the stroke instead of the whole stroke.
3.1 Cavitation and wear

Cavitation damage results mainly from liquid impact on the interacting surface during cavity collapse. Since viscosity inhibits cavity growth and collapse, it is expected that the collapse velocity will be lower in a Newtonian lubricant than in a shear thinning lubricant. Shear thinning influences cavitation behaviour primarily by modifying the film thickness. The shear thinning feature of multigrade lubricants and use of thinner monograde oils can generally reduce frictional losses but this may result in an increased wear of the cylinder liner (Rastogi and Gupta, 1992).

The hydrodynamic pressure is strongly dependent on the thickness of oil film and the effective width of the ring surface. When the ring-liner tribological conjunction is under the starved lubrication condition, the oil attachment and detachment would be changed. The effect of film thickness on cavitation behaviour must also be taken into account while assessing the influence of viscosity and cavitation damage (Rastogi and Gupta, 1992).

Oil viscosity and OFT are characteristics that affect friction losses. Higher viscosity oils provide adequate wear protection but movement creates more resistance and high frictional losses when measured for the total stroke length (Dellis, 2010). Lower viscosity oils reduce frictional losses and increase efficiency but high volatility leads to hydrocarbon emissions. As Wakuri et al. (1995) pointed out the reduction of oil viscosity makes the lubrication

Notes: (a) Capacitance, 400 rpm – 971 N/m and 50 cycles; (b) friction force, 400 rpm – 971 N/m and 50 cycles; (c) LIF, 300 rpm – 971 N/m and 50 cycles
around the dead centre of the stroke much more severe. It is, of course, a matter of great importance to further investigate lubricant properties and friction as there is always a trade-off between performance and emissions control.

The size of the cavitation zone in shear thinning lubricants is significantly larger than that for Newtonian lubricants. This difference in cavitation behaviour arises as a result of the lower tensile strength of shear thinning fluids (Rastogi and Gupta, 1992).

According to Kim et al. (1995), the striated oil film with fingerlike cavities is observed and verified and friction force in the region of oil film rupture must be taken into account.

A series of additives can be used to reduce friction under boundary lubricating conditions. Surface active agents are closely related to extreme pressure additives in structure and mode of action (boundary lubrication). VI improvers hold the decline in oil viscosity with temperature, a basic characteristic of multigrade oils.

The problem of changing the friction properties of the ring-liner interface was addressed with the introduction of the textured liner. Besides, because texture features can work as lubricant microreservoirs, the oil feeding induced by the texture features should also be considered as the additional source of oil supply. Surface texture changes the friction properties at the boundary-mixed lubrication region. Different groove styles and position contribute to the surface texture (piston ring or liner) and studies showed the existence of microcavitation on these grooves.

### 4. Conclusions
The above experimental research concludes:

- Cavitation causes the effective load area to shrink and the load capacity declines, even if specific hydrodynamic pressures remain high.
- The cavitating region for the different lubricants and piston rings should be considered.
- Friction peaks appear due to the collapse of the squeeze film close to the dead centres of the stroke, they are affected by speed, load, viscosity and HTHS viscosity (pronounced at the dead centres) and provide indications of wear development at the piston assembly interface.
- Liner or ring etching changes the lubrication regime at the areas of interest. Coordinated theoretical and experimental approaches are required to study cavitation initiation, development and the covering area on the piston ring surface.
- The lubricant film thickness has a major influence on the cavitation behaviour, which, in turn, affects the shape of the hydrodynamic pressure profile in the entrainment direction and consequently the ring-pack’s load carrying capacity.
- The shear thinning properties of the fluid seem to be the prevailing factor along with the effect of $V_{40}$ value of the lubricants that produce the LIF results. These factors seem to dictate the appearance, shape and size of cavities at mid-stroke. The chemistry of the lubricant and the addition of modifiers have an effect in the shapes of cavitation. Lubricant additives that can alter these properties have a significant effect not only in the film thickness but also in the form of cavitation at the trailing edge of the piston ring, affecting the shape of cavities (width and length) as the visualisation and LIF results show.
- Capacitance results (together with LIF results) showed that there is a slight change in the OFT measurements for the same testing conditions for different lubricants. This change is attributed to the kinematic viscosity of each lubricant and VI.
- MOFT and friction results are viscosity and VI dependent.
It has been verified that different forms of cavitation appear after the dead centres of the stroke that accompany the squeeze film (which is measured in the capacitance and friction signals).

The cavitation pattern as well as the formation and collapse of cavities at the interface of the piston ring/cylinder liner varies due to operating conditions, the physical properties and the chemical composition of the lubricant. Changes are introduced in the film thickness affecting the overall friction losses throughout the stroke and load carrying capacity of the piston ring through higher shear rates in the oil film and increased asperity contact. Increased friction losses and reduced load support leads to higher wear. The changing stress field as a result of this unique loading situation promotes fatigue wear, cracking, spalling, particle pullout and grooving (Ye and Cheng, 1996). Starvation of the piston rings in a real engine operation affects the lubricant film thickness and the subsequent phenomena.

The transition between cavitation stages is a factor that needs to be further investigated in terms of the effect it has in the load carrying capacity of the piston ring through the size of the cavitation area and also linked to the lubricant properties and additives (e.g. viscosity modifiers).

The well-defined cavitation stages in the test rig were not identified in engine visualisation testing, due to the engine testing conditions and starvation. Engines work in conditions of peak load and starved lubrication. The effect of different lubricant chemistry would provide useful results in terms of lubricant transport and cavitation.

References


**Further reading**


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Failure investigation of Cu-DHP tubes due to ant-nest corrosion

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Abstract
Purpose – The purpose of this paper is to address the main aspects of ant-nest corrosion failure mechanism of a Cu tube in heating ventilation and air-conditioning (HVAC) installations and analyze the possible root causes through various case studies presented.

Design/methodology/approach – Failure investigation process includes mainly stereo-, light optical and scanning electron microscopy coupled with energy-dispersive X-ray spectroscopy for elemental microanalysis, as the main analytical techniques for material characterization and root-cause analysis.

Findings – The investigation findings, obtained from corrosion products' analysis in conjunction to metallographic evaluation in transverse sections, illustrate the principal characteristics ("fingerprints") of ant-nest (formicary) corrosion mechanism.

Originality/value – This paper which deals with the presentation of applied failure analysis/case histories' investigation, summarizing the main aspects of an important and insidious type of Cu corrosion, taken place in HVAC installation systems and, on the other hand, presenting a complementary analysis of the chemical processes involved in the progressive failure mechanism constitutes an integrated approach, aiming to become a concise contribution to this subject.

Keywords Metallography, Ant-nest corrosion, Copper tubes

Paper type Technical paper

1. Introduction
Copper is the first member of Group IB of the Periodic Table having atomic number 29 and electronic configuration \([\text{Ar}]^{3d^{10}}{4s}^{1}\). The loss of the outermost s-electron produces the cuprous ion \(\text{Cu}^{+}\) and a second electron may be lost from the filled d-shell to form the cupric ion \(\text{Cu}^{2+}\). The availability of the d-electrons for coordination allows copper to readily form complexes with species, such as \(\text{NH}_3\) and \(\text{CN}\) as mentioned by Myers and Cohen (1995). As copper is not an inherently reactive element, it is not surprising that the rate of corrosion is usually low. Due to its high thermal conductivity is frequently used in unit installations such as air conditioners and refrigerators. Copper tubes from different applications exhibit various types of corrosion: pitting corrosion investigated by Lytle and Nadagouda (2010), erosion corrosion examined by Kuznicka (2009), microbiological influenced corrosion examined by Olszewski (2007), etc. A valuable review concerning the pitting corrosion mechanisms of Cu was presented by Edwards et al. (1994). A case study summarizing the main morphological aspects of cold water pitting corrosion (Type I) of Cu-DHP water tube is presented in a recent paper by Pantazopoulos et al. (2016), while hot water pitting corrosion (Type II) was addressed in a
relevant paper (Tzevelekou et al., 2013). Another Cu failure mechanism is related to stress corrosion cracking, driven by the presence of tensile stresses and corrosive environment, containing species, such as chlorides, ammonia, amines and moisture; characteristic studies are elaborated by Pantazopoulos et al. (2011) and recently by Metcalfe and Pearce-Boltec (2018). Fatigue fracture due to alternating thermal stresses, imposed as the effect of the continuous operation of heating–cooling cycles, is a frequent failure mechanism met in refrigerant systems (Pantazopoulos et al., 2013).

Temperature, chlorine, natural organic matter and microbial extracellular polymeric substances are among the factors most commonly cited as influencing copper corrosion in soft waters (Fontana, 1987). But, Cu can get corroded even from chlorinated domestic water and cooling water used in industrial applications (Fateh et al., 2017).

Environmental effect of coolant leakages due to Cu failure and the production of copper scrap can be critical to change flora and fauna’s stability and therefore it is quite important to prevent those defects (Zajicek and Coombs, 1982). The deposits that are formed by many types of corrosion can be dissolved or removed by using treatments with solutions of inorganic or organic acids for a proper time and temperature range (Kjellstrom and Lundgren, 2013).

USEPA emphasizes that water pipe corrosion and ageing is one of most serious concerns related to water distribution pipelines and may pose public health problems. Electrochemical corrosion is the main reason for exterior corrosion of cast and ductile pipes, which results in pipe deterioration in the form of corrosion pit on pipe walls. The physical mechanisms that lead to pipe leakage are often complex and not completely understood. The pipes having internal protection by lining and/or coating are less susceptible to corrosion. Modern metallic pipes are mostly manufactured with internal linings to prevent internal corrosion from soft or aggressive water.

Cu-DHP tubes from finned heat exchanger units, suffered from ant-nest or formicary corrosion, were investigated by Peltola and Lindgren (2015). Organic matter was detected on the outer surface and in the interior of the crack. It was believed that formicary corrosion was detected only in pure copper tubes but other copper alloys (CuNi) could also be susceptible to such type of corrosion (Olszewski and Corbett, 2007).

Only few works exist concerning the study of this type of corrosion in heat exchanger units. Typical examples of such case studies are referred by Chandra et al. (2014), Nasrazadani and Nakka (2016) and Zhou et al. (2018). The synergistic effect of ant-nest corrosion and stress corrosion was addressed by Notoya (1991a, b). There is a rather significant value of such industrial research aiming to analyze, understand and prevent failures, which eventually exhibit high environmental impact and material/equipment losses. Ant-nest corrosion constitutes a representative example of a progressive deterioration process, possessing high advancement rate and low detectability (Baba and Kodama, 1995). The name of this corrosion failure is originated from the morphology of the problematic area, which consists of corroded pinhole traces of several microns size. The main contributing factor to this phenomenon is the presence of carboxylic acids, which participate to the corrosion reactions together with moisture and dissolved oxygen. The scope of the present work is to summarize critical issues of ant-nest corrosion in order to suggest ways to prevent such failures. For this reason, the results of selected case histories will be summarized in the frame of this paper. The present paper was based on a recent presentation, illustrating some essential aspects of ant-nest corrosion (Vazdirvanidis et al., 2018).

2. Experimental methods
The present study is a compilation of a variety of characteristic case histories investigated within almost a three-year time period and were related to thin walled Cu-DHP tubes for air-conditioning and refrigeration as well as to thicker tubes for sanitary applications, which
exhibited leakage after a very short operation duration. Visual inspection of the surfaces of samples was performed after longitudinal cross-section, while metallographic specimens were made by transverse cross-sections for observation of the corrosion type. Stereoscopic observation of the tubes was performed with a Nikon SMZ 1500 stereo-microscope, while metallographic examination of mounted specimens (transverse and longitudinal sections) was accomplished by a Nikon Epiphot 300 inverted metallographic microscope. Higher magnification observation was carried out using an FEI XL40 SFEG scanning electron microscope (SEM) under 20 kV accelerating voltage, coupled with an EDAX energy-dispersive x-ray spectroscopy (EDS) Apollo XF silicon drift detector of 60 mm² window area (ZAF corrected and standardless mode).

3. Investigation findings

This type of corrosion as has already mentioned leads to a premature failure of the Cu tube, within few months. The outer surfaces of these tubes were often without serious indications of corrosion or severe mechanical damage, while the inner surfaces were heavily corroded exhibiting complex and branching channels, covered by corrosion products, while microcracks were found on some of them. The corroded tubes and relevant features of the corrosion products for the selected case studies were presented in Figures 1–10. In optical micrographs (Figures 1–3), two different cases were depicted, namely, Case A (Figure 1) and Case B (Figures 2–3), a branched propagation with oxide deposits inside the caverns was exhibited initiating from tube's inner surface, containing copper oxides. The nature of the interior cavern contents was also observed under polarized light illumination. A similar type of corrosion propagation, initiated from the outer surface in this case, and showing the complexity of the interconnecting channels leading to final, through-the-wall, tube perforation, is shown in Figure 3. Information about the exact use of the tubes, environmental conditions, temperature, etc., was confidential and/or could not been retrieved, since the detailed service conditions remain unknown and/or non-controlled.

The chemical composition and the morphology of the corrosion deposits existed inside the pit were thoroughly examined by means of SEM/EDS analyses. The relevant findings are presented in Figures 4–10. A typical spectrum of the corrosion products, corresponding to Case A (see Figure 1), showing their semi-quantitative chemical composition is presented in Figure 4. Similar results, pertaining to the same Case A, showing the compact and coarse morphology of the interior pit contents, together with their chemical composition obtained by EDS analysis is shown in Figure 5. These corrosion products consisted mainly of cuprous oxide (cuprite, Cu₂O). The cubic crystal morphology of the cuprite crystals deposited in the corroded surfaces is highlighted in Figure 6. Malachite nodules constitute the upper layer of the “sandwich” layered structure, covering normally cuprite crystals, are shown in Figures 7 and 8. These carbonate salts have a typical morphology or radiated needle crystals forming spherical aggregates; similar structures were depicted in relevant studies (Pantazopoulos, 2009; Pantazopoulos et al., 2016). The coarse deposits were also contaminated by S, Al, K and C, coming from species contained in the flowing medium (Figure 8), which were found to a similar, but different case, denoted as Case C.

The formation of blocky and high volume oxide deposits inside the corrosion channels provide additional tensile stresses and increase local stress intensity, as the result of the “wedging effect,” leading to crack initiation and propagation, facilitating the ultimate tube rupture (Figures 9 and 10).

Corrosion was propagated through branched and interlinked corrosion channels filled with coarse corrosion products. Chemical analyses exhibited mostly O, S, Al, K but also C which could be attributed to organic matter presence. All the above findings, pertaining to the corrosion patterns, deposits’ chemistry and morphology are consistent to the action of ant-nest corrosion mechanism.
4. Discussion
The form of the corrosion propagation in combination to the nature of the corrosion products and low endurance are signs of ant-nest corrosion process. Typical results of ant-nest corrosion also presented in related literature for refrigerating units and Cu tube/Al plate finned type heat exchangers (Chandra et al., 2014; Nasrazadani and Nakka, 2016; Zhou et al., 2018). Ant-nest corrosion is a rapidly evolved corrosion process which is usually propagating through grain boundaries, resulting in perforation of a thin-wall tube, usually in a very short time period (within weeks or months). Ant-nest corrosion most commonly appears in the copper piping of air-conditioning or refrigeration equipment and is mostly found in DHP-Cu, which is the main material of construction of heat exchangers. Organic compounds used during the fabrication and joining of the copper tubes (e.g. the residues of lubricants, drawing emulsions, degreasing agents and detergents) if not completely removed may be dissociated to carboxylic acids, in presence of air and humidity. The exposure of copper surfaces and interfaces to carboxylic acids, such as formic acid (HCOOH), acetic acid (CH₃COOH), propionic acid (C₃H₆COOH) and butyric acid (C₄H₇COOH), stimulates the reaction leading to the formation of copper carboxylates. The detailed corrosion mechanism is also described in detail by Notoya (1991a, b), Bastidas et al. (2000) and Stepanus Situmorang and Kawai (2018). Ant-nest corrosion attack initiates at some local surface discontinuity, i.e. such as a surface defect or a scratch, a flaw in the protective oxide film. These sites are attacked by the carboxylic acid, which penetrates through the discontinuity of oxide film. In moist conditions, the copper base metal oxidizes and dissolves, according to reaction (1) leading to the formation of small corrosion pits, which are considered characteristic of ant-nest corrosion:

\[ \text{Cu(s)} \rightarrow \text{Cu}^+ (\text{aq}) + \text{e.} \]  

(a) Macrograph of sample examined (Ø12.7×0.35 mm wall thickness for ACR) indicating area of interest; (b) transverse cross-section showing the copper corrosion products (inside-out corrosion); (c) same area observed under polarized light – copper oxides are shown reddish (Case A)

Figure 1. Cu-DHP tubes showing leakage

Notes: (a) Macrograph of sample examined (Ø12.7×0.35 mm wall thickness for ACR) indicating area of interest; (b) transverse cross-section showing the copper corrosion products (inside-out corrosion); (c) same area observed under polarized light – copper oxides are shown reddish (Case A)
Notes: (a)-(b) Optical micrographs, cross-section of another tube showing ant-nest corrosion type failure (Ø15 x 0.70 mm wall thickness for sanitary use). Note the accumulation of the corrosion products close to the inner surface; (c) detail of the oxide contents captured under polarized light illumination (Case B)

Figure 2. Optical microscopy of failed Cu-DHP tubes

Figure 3. Optical micrographs, cross-section of another tube showing ant-nest corrosion type failure on Cu-DHP tube sample (soft annealed tube for air-conditioning) (Case B)
Under the presence of carboxylate anions, the dissolved copper ions react forming unstable Cu(I) carboxylate, which oxidizes further to form Cu(II) carboxylate and copper oxide Cu₂O according to the chemical reactions (2) and (3):

\[ \text{Cu}^+ (\text{aq}) + \text{HCOO}^- \rightarrow \text{Cu}(	ext{HCOO})(\text{aq}), \quad (2) \]
The solid constituents Cu(COOH)₂ and Cu₂O precipitate in the interior of the corrosion pit. Due to the volume expansion associated with the formation of Cu₂O, microcracks may initiate and propagate outward within the corrosion pit wall under the imposed wedging effect. The microcracks liberate an appreciable fraction of pure copper surface area and the process advances within the cracks between Cu (II) carboxylate and pure unaffected Cu. Consequently, the Cu (I) complex metallo-organic compound is re-produced according to the following chemical reaction:

\[
4\text{Cu(HCOO)}(\text{aq}) + \frac{1}{2}\text{O}_2 \rightarrow 2\text{Cu(COOH)}_2(\text{s}) + \text{Cu}_2\text{O}(\text{s}).
\]  (3)

Therefore, the reactions (3) and (4) autocatalytically occurred, forming channels, until final through wall perforation occurs. The standard oxygen cathodic reactions (5) and (6) are

\[
\text{Cu(COOH)}_2(\text{s}) + \text{Cu}(\text{s}) \rightarrow 2\text{Cu(HCOO)}(\text{aq}).
\]  (4)
Figure 7.
SEM micrographs of the inner surface showing malachite\'s formation on a Cu-DHP tube sample (soft annealed tube for air-conditioning (Ø6.35)).

Note: Note the spherical nodules composed of radiating needle crystals.

Figure 8.
A typical energy-dispersive X-ray spectrum of spherical aggregates.

Notes: The elemental analysis shows that they consisted mostly of carbonate crystals. Traces of S, Al, K and C, coming from species contained in the flowing medium are also detected (Case C).
the following:

\[ \text{O}_2 + 2\text{H}_2\text{O} + 4\text{e}^- \rightarrow 4\text{OH}^- , \]  

\[ \text{O}_2 + 4\text{H}^+ + 4\text{e}^- \rightarrow 2\text{H}_2\text{O}. \]

The end-points of the pits or the advancing channels act as anodes for Cu oxidation, and the walls of the channels and the surface of the copper tube act as cathodes for oxygen (cathodic) reduction reactions. The presence of carboxylic acid in the humid atmosphere catalyzes this reaction mechanism. There are three factors that are necessary for ant-nest corrosion to occur, and they should be in effect simultaneously: moisture, air and an organic compound (carboxylic acid). If any of the three components is consumed or removed, ant-nest corrosion is terminated (Lopez-Delgado et al., 2001). The inner and often outer tube wall surfaces is usually covered by copper (hydro)oxide products, while the presence of C, Al, S, K is observed as a result of contaminants, deposited on the surface or infiltrated in to the crack area, see also Chandra et al. (2014), Notoya (1991a, b).

It is assumed that damages have resulted in local through-thickness discontinuities, which contributed to the leakage observed by the user. Summarizing the above findings, based on pit morphology and propagation, as well as corrosion product nature, it could be inferred that leakages were the outcome of a rapid corrosion process.

The light microscopy and SEM/EDS analysis results obtained for the studied cases (Figures 1–3), indicated the occurrence of ant-nest corrosion mechanism. The complex-tortuous shape of the interlinking channels, revealed mainly by light optical microscopy, as well as the
elemental composition of the corrosion products, constitute supporting evidence of the stated corrosion mechanism. EDS analysis, mainly illustrated in Figures 4–5, indicated the presence of cuprous oxide (Cu$_2$O) and carbon, most likely of organic nature, as the formation of these products is predicted by the chemical reactions (2)–(4). The collected experimental evidence together with the short lifetime of the tube is rather suggestive of the occurrence of ant-nest corrosion mechanism.

5. Conclusions
The examined tubes exhibited leakages after short periods of usage (within a few months of service). Since the defected/damaged areas were not visible to the naked eye, tube degradation proceeds to final failure and leakage without any prior warning or detection of evolving discontinuity. The investigation findings advocate the occurrence of ant-nest corrosion as the pertinent failure mechanism, which was rapidly evolved, having caused severe damage to the heat exchanger (and refrigeration) units with serious consequences to environmental and equipment damages. The main findings of the current investigation can be summarized as follows:

(1) corrosion had propagated, following a tortuous and branching path, causing tube wall perforation;

(2) the inner tube wall surfaces were covered by coarse copper (hydro)oxide products, while the presence of C, Al, S, K coming from water condensates and organic matter (C) has also been identified; and

(3) the presence of carbon (C) indicated possible presence of organic substance (e.g. lubricant).
Further research and acquisition of additional knowledge focused on the allocation of the nature of organic residues (e.g. lubricants, cleaning agents), which play an important role on the activation of ant-nest corrosion mechanism is recommended, in order to obtain a valuable insight of the degradation process mechanism (Elliott and Corbett, 2001). Possible prevention of such phenomena might involve the development and realization of the suitable cleaning/dgreasing procedures of heat exchanger tubes in order to remove effectively lubricant and organic substance residues, which constitute significant factors of the imminent corrosion mechanisms. Drastic removal of organic contaminant even if achieved cannot exclude the possibility of contamination during operation related to the operating environment. When ethanol was used as the working fluid with high oxygen content, more corrosion occurs, which was due to the working fluid reaction and should be avoided (Kuroda et al., 2001). Ant-nest corrosion is mainly focusing on the operating conditions of the indoor heat exchanger units. The application of an experimental simulation method of ant-nest corrosion will further facilitate the verification of the effectiveness of the proposed corrective actions and test potential alternatives regarding copper chemical composition to combat such insidious corrosion phenomena.

References


Further reading


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Elastic wave mode conversion phenomenon in glass fiber-reinforced polymers

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Abstract

Purpose – The purpose of this paper is to present the results of experimental analysis of the elastic-guided wave mode conversion phenomenon in glass fiber-reinforced polymers. The results of this research presented in this paper are strictly focused on S0/A0’ mode conversion phenomenon caused by discontinuities in the form of circular Teflon inserts (artificial delaminations) and impact damage. Results of this research could be useful in problems of damage detection and localization.

Design/methodology/approach – In the research, guided waves are excited using a piezoelectric transducer and sensed in a non-contact manner using a scanning laser Doppler vibrometer. Full wavefield measurements are analyzed. Analysis of the influence of investigated discontinuities on S0/A0’ mode conversion is based on the elastic wave mode filtration in frequency-wavenumber domain. Mode filtration process allows us to remove the effects of the propagation of unwanted type of mode in forward or backward direction. Effects of S0/A0’ mode conversion are characterized by a mode conversion indicator (MCI) based on the amplitude of new mode A0’ and the amplitude of incident S0 mode.

Findings – It was noticed that the magnitude of MCI depends on the depth at which the Teflon inserts were located for all analyzed excitation frequencies and diameters of inserts (10 and 20 mm). The magnitude of MCI also increases with increasing impact energies. The S0/A0’ mode conversion phenomenon could be utilized for the detection of surface and internal located discontinuities.

Originality/value – This paper presents the original results of this research related to the influence of discontinuity location with respect to the sample thickness and severity of discontinuity on S0/A0’ mode conversion.

Keywords Discontinuities, Elastic waves

Paper type Research paper

1. Introduction

Nowadays, attention of many researchers is strictly focused on the phenomenon of elastic-guided wave propagation and its application for the structural health monitoring. Damage initiated in the structure creates material discontinuity that is then the potential source of changes of wave propagation. These changes manifest as reflection, scattering or mode conversion. Based on the analysis of elastic wave propagation in the structure, it is possible to detect and localize damage. Research related to the elastic wave propagation method could be divided into numerical and experimental methods.

The aim of numerical modeling is a simulation of elastic wave propagation in different structures very often including damage. The aim is to determine the influence of damage on the elastic wave propagation phenomenon. Different methods for the simulation of guided wave propagation are utilized. The finite element method was utilized for modeling of guided wave propagation in the composite structure in the papers of Murat and Fromme (2016), Hennings and Lammering (2016) and Shoja et al. (2018). Detailed model for analysis of the effect of quasi-continuous mode conversion was presented by
The aim of the study presented by Shoja et al. (2018) was to find an efficient way of FEM modeling of guided wave propagation in composite laminates. Delaminations were modeled by local reduction of the stiffness. Guided waves simulations using four different simulation codes, the commercial FEM modeling packages included in ABAQUS, ANSYS and COMSOL software, and a custom developed code executing the elastodynamic finite integration technique, were performed by Leckey et al. (2018). Another method is a spectral element method (SEM) in the time domain utilized for modeling of elastic wave propagation in long fiber-reinforced composites (Kudela et al., 2018a) and for short randomly oriented reinforcing fibers (Kudela et al., 2018b).

The aim of the experimental research is very often validation of numerical results. A large number of publications containing experimental results for the field of elastic wave propagation could be found. The results of simulation cases based on FEM were compared with experimental results in Leckey et al. (2018). In this case, scanning laser Doppler vibrometry (SLDV) was utilized in experimental research. The SLDV technique was also utilized for the verification of numerical results obtained from SEM (Kudela et al., 2018a, b). A full wavefield method using an SLDV technique is very popular in the research related to elastic wave propagation. This method was utilized by Murat and Fromme (2016) for the study of wave scattering at delamination. There are also many interesting applications of the elastic wave propagation method. The suitability and sensitivity of three circumferential guided wave modes for detecting wall thinning defects have been studied by Howard and Cegla (2017). The guided wave modes were evaluated using two measurement modalities: using the reflected signal and using the through transmitted signal. This method was utilized also for the inspection of level crossing rails (Evans et al., 2018). A study of optimal piezoelectric transducer for the excitation of guided waves in a rail web was proposed in Ramatlo et al. (2018). Local defect resonance method was proposed by Kersemans et al. (2018). A multi-mode reverse-time migration approach for damage imaging using elastic waves was proposed by He et al. (2018). The aim was to develop a multi-mode damage imaging technique that will be utilized for the identification of its size and location based on guided waves. The proposed technique combines a reverse-time migration imaging algorithm with a 3D wave propagation simulator using different wave modes. Quantitative characterization of disbonds in multilayered bonded composites was performed using laser ultrasonic guided waves by Zhang and Zhou (2018). Detailed analysis of guided wave propagation in the frequency-wavenumber domain for the purpose of damage detection was performed by Carrara and Ruzzene (2015). The damage detection approach based on local wavenumber analysis was proposed by Juarez and Leckey (2015). There are still a small number of numerical and experimental works related to the analysis of mode conversion phenomenon. The effect of continuous mode conversion in CFRP plate was analyzed by Hennings and Lammering (2016) and Willberg et al. (2012). A mode conversion phenomenon was utilized for the estimation of rectangular notch parameters in the metallic plate which was utilized by Ghadami et al. (2015). In the phenomenon of Rayleigh wave mode to Lamb wave mode, the conversion was investigated numerically and experimentally in the case of the thick metal plate by Schaal et al. (2015). Investigated discontinuity that initiated mode conversion was crack created in a thick plate. The problem of shear horizontal (SH) mode conversion in the tapered aluminum plate was considered by the paper of Nakamura et al. (2012). Several experiments were conducted to compare the efficiency of mode conversion phenomenon in the pipeline by Sun et al. (2016). It can also be inferred that the S0 wave has a better ability to convert to the T(0, 1) mode, whereas the SH0 wave easily converts to the L(0, 1) mode and L(0, 2) mode. The conversion of evanescent SH guided waves into propagating
waves was studied by Yan and Yuan (2018). The conversion is exemplified by a time-harmonic SH evanescent displacement prescribed on a narrow aperture at an edge of a semi-infinite isotropic plate. The conversion efficiency and converted modes are dependent on the geometric configuration of the aperture as well as the selection of the excitation frequency. 3D scanning laser vibrometry and mode conversion effect utilized for damage detection were studied by Pieczonka et al. (2017). The mode conversion phenomenon in the case of damage simulated by Teflon insert located non-symmetrically in respect to the thickness of composite panels was investigated by Wandelowski et al. (2018). The results presented in this paper showed that $S_0/A_0'$ mode conversion occurs for all non-symmetrical locations of Teflon inserts which is obvious. Moreover, preliminary results indicated that even for Teflon insert symmetrically located in respect to the thickness of composite panel (the same number of plies above and below insert) mode conversion $S_0/A_0'$ was noticed. It proved that perfect symmetry was not achieved in real conditions.

Experimental research presented here is the continuation of the study of the influence of discontinuities on mode conversion presented in Wandowski et al. (2018). The aim of this research presented in the current paper was to investigate the influence of location (in respect to the sample thickness) of circular Teflon insert on $S_0/A_0'$ mode conversion phenomenon. Teflon inserts were located between layers of glass fiber-reinforced polymer (GFRP) composite material at different depths. Moreover, the influence of impact damage with different impact energies on $S_0/A_0'$ mode conversion was investigated. This paper presents the results of original research related to the influence of discontinuity location with respect to the sample thickness and severity of discontinuity on $S_0/A_0'$ mode conversion. It was shown that the proposed mode conversion indicator based on $S_0/A_0'$ mode conversion phenomenon could be utilized as the indicator of internal discontinuity in material and discontinuity located on the surface. The mode conversion phenomenon could be utilized in a similar way like mode reflection and transmission effects for the characterization of discontinuities in the material.

2. Materials

Research reported in this paper is related to GFRP materials. In the purpose of the research, GFRP specimens in the form of panels with dimensions: 500 mm × 500 mm and thickness ~1.5 mm were manufactured. Each investigated specimen consisted of 12 layers of VV192T/202 prepregs with epoxy resin IMP503 with orientations: (0/90/0/90/0/90)_S. First specimen contained discontinuities in the form of four circular Teflon inserts with diameters 20 mm. Second specimen also contained discontinuities in the form of four circular Teflon inserts but its diameter was smaller and equaled 10 mm. Teflon inserts were located between different layers of prepreg at different depths (Figure 1(a) and (b)). Third GFRP specimen contained discontinuities in the form of impact damage with energies 5, 10 and 15 J (Figure 1(c)). Impact damage was created by steel ball dropped from selected heights in order to achieve desired impact energy levels.

3. Experimental set-up

Experimental set-up utilized in presented research consisted of investigated GFRP specimens, piezoelectric transducers bonded on the samples and Polytec SLDV PSV-400. Each specimen was equipped with a piezoelectric transducer utilized for elastic wave excitation. The piezoelectric transducer was placed at the middle of the specimen (see Figure 1(a)–(c)).

Excitation signal was in the form of five cycles of sine modulated by Hann window. Three excitation frequencies were utilized in this study: 100, 150 and 200 kHz. Elastic wave sensing was conducted in a non-contact way using SLDV. Measurements were taken at
the surface of specimen opposite to the surface where the piezoelectric transducer was bonded on (indicated by arrow in Figure 1(a)–(c)). Only one scanning laser head of the laser vibrometer was utilized – measurements were taken along laser beam approximating out-of-plane guided elastic wave velocities of material particle.

4. Wavenumber mode filtering

In the frame of the experimental research presented in this paper, full wavefield measurements of propagating elastic wave were performed. Non-contact full wavefield measurements were conducted based on SLDV equipment for all investigated samples. Based on full wavefield measurements, matrix containing wave signals in time domain is obtained \( s(x, y, t) \). This matrix contains a set of signals registered at points of dense mesh with coordinates \((x, y)\) spanned over the whole surface of investigated specimen. Based on such matrix of signals \( s(x, y, t) \), animation presenting the propagation of elastic waves in specimen could be created taking subsequent frames in time. Results in the form of selected frame for such animation for first specimen (Figure 1(a)) are presented in Figure 2. The frame presented in Figure 2 is related to the propagation of elastic waves with frequency 200 kHz. Time instant for this frame was selected in such a way to present the propagation of S0 and A0 mode without any interaction with Teflon inserts and edges of specimen. Both modes (S0 and A0) could be distinguished by different wavelengths (longer for S0) and velocities of propagation (higher for S0).

Having the full wavefield matrix of signals in time and space domain \( s(x, y, t) \), it is possible to transform this matrix to frequency-wavenumber domain \( S(k_x, k_y, f) \) using 3D Fast Fourier transform based on formula:

\[
S(k_x, k_y, f) = \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} s(x, y, t) e^{-j(2\pi f t - k_x x - k_y y)} dt dx dy,
\]

where \( t \) is the time, \( f \) the frequency, and \( k_x, k_y \) the wavenumbers along \( x \) and \( y \) axis.

In this paper, one-dimensional analysis is performed. It means that only signals gathered along lines not from the whole surface of specimen are considered in further analysis.
In research presented here, four lines running from the point of wave excitation (location of piezoelectric transducer at the middle of the specimen) through discontinuity in the specimen to the edge of specimen are utilized. In Figure 3, such four lines L1–L4 for specimen with Teflon inserts were presented.

These lines include Teflon inserts T1–T4 (see Figure 1(a) and (b)). In the case of one-dimensional analysis reduced matrices of signals: \( s(y, t) \) for lines L1 and L3 and \( s(x, t) \) for lines L2 and L4 and are utilized in a signal processing algorithm. Matrices in time and space domains could be utilized in order to create waterfall plot. In Figure 4, the example of such plot for signals in time-space domain collected along line L2 (including Teflon insert T2) in sample No. 1 is presented. These signals were gathered for the case of excitation frequency 200 kHz.

The propagation of two fundamental A0 and S0 modes is clearly visible, both modes are generated at the beginning of the time. Mode S0 achieved the edge located at distance 0.25 m (half width of panel – wave excitation in the middle) at time instant ~0.1 ms. Two modes exist on the edge of panel: edge reflected S0 mode and A0' as a result of S0 mode conversion on the edge (see Wandowski et al., 2018). Mode A0 achieved the edge of panel at time instant ~0.2 ms. Moreover, in Figure 4, two horizontal lines were drawn. Lines indicate the location of Teflon insert T2 in sample No. 1 (0.115–0.135 m).
signals in Figure 4 in the location of Teflon insert T2, there is a clearly visible conversion of S0 mode to A0' at time instant ~0.05 ms. At this time, instant at the Teflon location backward and forward propagation of converted A0' mode is visible (see Wandowski et al., 2018). Reduced matrices in time and space domains: $s(x, t)$ and $s(y, t)$ could be further transformed to frequency-wave number domains using 2D Fourier transform. This could be done using formulas:

$$S(k_x, f) = \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} s(x, t)e^{-j(2\pi ft - k_x x)} dt dx,$$  \hspace{1cm} (2)

$$S(k_y, f) = \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} s(y, t)e^{-j(2\pi ft - k_y y)} dt dy.$$  \hspace{1cm} (3)

Representation of signals in frequency-wavenumber domain is given in Figure 5. These signals were collected along line L2 (including Teflon insert T2) in sample No. 1 for excitation.
frequency 200 kHz. In this figure, wavenumbers related to the forward propagation of symmetric S0F and antisymmetric A0F modes are visible. Beside this, wavenumbers related to the backward propagation of symmetric and antisymmetric modes S0B and A0B can be noticed. Moreover, analyzing the results presented in Figure 5, it could be noticed that wave energy is concentrated for frequency equal to 200 kHz relating directly to the frequency of excitation. Wave energy is concentrated in the band with certain width related to the nature of Hann window modulation in tone burst signal.

Having representation of signals in the frequency-wavenumber domain, it is possible to conduct the process of modes filtration. This filtration allows us to remove the forward or backward propagation of selected modes from the wavefield. It is possible to remove the whole propagation of selected mode from the wavefield or only desired direction of propagation (forward or backward) of chosen mode using specially designed mask of the filter. After the application of filter mask for frequency-wavenumber data inverse 2D Fourier transform is utilized to transform filtered wavefield back to time-space domain. More information about this method could be found in papers (Carrara and Ruzzene, 2015; Yu and Tian, 2015; Kudela et al., 2015).

5. Experimental results

In this section, the results of analysis of S0/A0' mode conversion phenomenon caused by discontinuities in three investigated specimens are presented. In the first subsection, results of detailed analysis of mode conversion for Teflon insert with diameters 10 and 20 mm located at different depths are presented. In the second subsection, the analysis of mode conversion phenomenon caused by impact damage with different impact energies is investigated.

5.1 Teflon insert influence

In order to analyze the influence of Teflon inserts on S0/A0' mode conversion matrix containing full wavefield signals \( s(x, y, t) \) was utilized. Based on it, animation of elastic wave propagation in investigated specimen No. 1 with larger inserts Teflon (diameter 20 mm) was created.

In Figure 6, selected frame from animation illustrating the phenomenon of elastic wave propagation in this sample for excitation frequency 200 kHz was presented.

**Figure 6.** Full wavefield pattern at selected time instant for sample No. 1 with Teflon inserts with diameter 20 mm
The effects of mode conversion $S_0/A_0'$ at four Teflon inserts are observed (compare with Figure 1(a)). At the locations of circular Teflon inserts, symmetric $S_0$ mode is converted to fundamental antisymmetric mode denoted $A_0'$. Intensity of mode conversion could be characterized by the amplitude of $A_0'$ generated at the locations of inserts. Analyzing the results presented in Figure 6, it can be noticed that the amplitude of $A_0'$ mode depends on the depth at which Teflon insert was located (compare with Figure 1(a)). The largest amplitude of $A_0'$ mode was achieved in the case of Teflon insert $T_1$, whereas as the smallest amplitude of $A_0'$ mode was achieved for Teflon insert $T_4$ (see Figure 1(a)).

Then the mode filtration process described in Section 4 was utilized in order to perform detailed analysis of $S_0/A_0'$ mode conversion intensity at Teflon inserts located at different depths. In Figure 7(a), raw time-space signal gathered for sample No. 1 along line $L_1$ (Teflon insert $T_1$ – Figure 1(a)) is presented. This signal was plotted for the time instant at which the propagation of $A_0$ and $S_0$ mode is clearly visible. At this time, instant $S_0$ mode already propagated through the location of Teflon insert $T_1$ marked by vertical lines in Figure 7(a). The effect of $S_0/A_0'$ mode conversion could be noticed in Figure 7(a) in location marked by $A_0'$. Moreover, mode conversion could also be noticed in the non-regular shape of $S_0$ mode. The mode filtration process could be helpful in the analysis of mode conversion. After the mode filtration process where the forward and backward propagation of mode $A_0$ ($A_0F$ and $A_0B$) is removed, only the propagation of $S_0$ mode could be analyzed. In Figure 7(b), signal representing solely propagation of $S_0$ mode is presented. The effects of $S_0/A_0'$ mode conversion could be clearly visible after the filtration of forward propagating modes $S_0$ and $A_0$ and backward propagating mode $S_0$. There is only backward propagation of mode $A_0'$ in the wavefield. Additionally, time instant needs to be selected in such a way that $S_0$ mode already reached the location of discontinuity, like Teflon insert in analyzed case in Figure 7(c). In this case in the location of Teflon insert newly created (after conversion) and backward propagating mode $A_0'$ could be observed (Figure 7(c)). This is the same mode that is visible in Figure 6 in the location of Teflon insert.

In Figure 8(a), raw time-space signal gathered along line $L_2$ (Teflon insert $T_2$ – Figure 3) for sample No. 1 (Figure 1(a)) is presented. Similar to the case presented in Figure 7(a), the propagation of $S_0$ and $A_0$ modes is visible. Moreover, the effects of $S_0/A_0'$ mode conversion are visible. One difference in signals from Figures 7(a) and 8(a) is the larger amplitude of $A_0$ and $S_0$ for the second case. This is related to the fact that the amplitudes of $S_0$ and $A_0$ modes have non-uniform amplitudes in all directions (assuming the same material properties, the same directions of reinforcing fibers like here). This is caused by the bonding layer of transducer and its modes of vibration of transducer during the excitation (this effect was already reported in previous authors work: Wandowski et al., 2018).

In Figure 8(b), the effects of mode filtration are visible. In this case, solely propagation of $S_0$ mode is presented. In Figure 8(c), there is only backward propagation of mode $A_0'$ in the wavefield. This propagation is related to $S_0/A_0'$ mode conversion at Teflon insert $T_2$ located in specimen No. 1. This analysis was continued for Teflon inserts $T_3$ and $T_4$ in specimen No. 1. Beside excitation frequency 200 kHz for which results were presented in Figures 7 and 8, the same analysis for frequencies 100 and 150 kHz was performed.

Next, the same signal analysis was performed for specimen No. 2 with the smaller size of Teflon inserts (diameter 10 mm – Figure 1(b)). The aim of research related to specimen No. 2 with the smaller diameters of Teflon inserts was to validate that it will be still possible to see $S_0/A_0'$ mode conversion effects for discontinuities with smaller size. In Figures 9(a) and 10(a), raw time-space signals gathered for sample No. 2 along lines $L_1$
and L2 (Teflon insert T1 and T2 – Figure 1(b)) were presented. In this case, frequency was also equal 200 kHz. In both cases, the propagation of S0 and A0 modes, and S0/A0’ mode conversion effects are visible. In Figures 9(b) and 10(b), the same signals after mode filtration were presented. Here, only the propagation of S0 mode is visible. Slight differences in the amplitudes of S0 modes for lines L1 and L2 could be noticed. The amplitude of S0 mode for line L2 is slightly larger. Finally, in Figures 9(c) and 10(c), signals after mode filtration containing only backward propagation for mode

Notes: (a) Raw; (b) filtered – propagation of S0 mode; (c) filtered – backward propagation of A0; specimen No. 1

Figure 7.
Space signals gathered along line including Teflon insert T1
A0' are presented. Here, the results of S0/A0' mode conversions at the location of Teflon inserts T1 and T2 with diameters 10 mm are clearly visible. The same procedure was conducted for signals gathered along lines L3 and L4 (inserts T3 and T4) and for excitation frequencies 100 and 150 kHz.

In order to compare the effects of S0/A0' mode conversion mode conversion indicator (MCI) was formulated. It compares the amplitudes of S0 mode reaching Teflon insert with

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**Notes:**
(a) Raw; (b) filtered – propagation of S0 mode; (c) filtered – backward propagation of A0; specimen No. 1

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**Figure 8.**
Space signals gathered along line including Teflon insert T2
the amplitude of $A_0'$ mode generated as the result of the conversion of $S_0$ mode. In this research presented in this paper, the following formula was proposed:

$$MCI = \frac{\max(|A_0'|)}{\max(|S_0|)},$$

(4)
where MCI is the mode conversion indicator. The amplitude of S0 mode was taken for signal after mode filtration containing only the propagation of S0 mode (Figures 7(b) and 8(b)). The amplitude of A0 mode was taken for signal after mode filtration containing solely the backward propagation of A0 mode (Figures 7(c) and 8(c)). The amplitude of A0' for MCI is taken in the location of Teflon insert indicated by vertical lines in Figures 7(c) and 8(c). Moreover, it needs to be explained that the amplitude of S0 mode in MCI is utilized for

**Figure 10.**
Space signals gathered along line including Teflon insert T2

**Notes:** (a) Raw; (b) filtered – propagation of S0 mode; (c) filtered – backward propagation of A0; Specimen No. 2
normalization purpose. The normalization process allows to eliminate the problem of different amplitudes of S0 mode in different directions. In Table I, values of mode conversion indicator (MCI) for Teflon inserts with diameters 10 and 20 mm located at different depths in specimen Nos 1 and 2 were presented. Values of MCI were determined for three different frequencies: 100, 150 and 200 kHz. After the analysis of MCI values in Table I, characteristic trend could be noticed. Values of MCI decrease with the depth at which Teflon inserts were located. This trend is achieved for both sizes of Teflon inserts (diameters 10 and 20 mm) and for the three investigated frequencies.

In Figure 11, normalized values of MCI for specimen No. 1 with Teflon inserts with diameter 20 mm were plotted. Normalization was based in respect to the maximum value of MCI achieved for Teflon insert T1 located at smallest depth (Figure 1(a)). Results in Figure 12(a)–(c) were plotted, respectively, for excitation frequencies 100, 150 and 200 kHz. Monotonic trend related to a decrease of MCI value with the depth where insert is located could be noticed.

<table>
<thead>
<tr>
<th>Discontinuity</th>
<th>Depth (layers from the bottom)</th>
<th>100 kHz</th>
<th>150 kHz</th>
<th>200 kHz</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>$D = 10$</td>
<td>$D = 20$</td>
<td>$D = 10$</td>
</tr>
<tr>
<td>T1</td>
<td>2/3</td>
<td>0.126</td>
<td>0.463</td>
<td>0.219</td>
</tr>
<tr>
<td>T2</td>
<td>4/5</td>
<td>0.124</td>
<td>0.298</td>
<td>0.081</td>
</tr>
<tr>
<td>T3</td>
<td>6/7</td>
<td>0.063</td>
<td>0.235</td>
<td>0.053</td>
</tr>
<tr>
<td>T4</td>
<td>8/9</td>
<td>0.047</td>
<td>0.085</td>
<td>0.040</td>
</tr>
</tbody>
</table>

Table I. MCI values for Teflon inserts at different depth and for different wave frequencies

Notes: (a) 100 kHz; (b) 150 kHz; (c) 200 kHz

Figure 11. Normalized amplitudes of mode conversion indicator (MCI) for specimen No. 1 with Teflon inserts with diameter 20 mm for frequencies

Notes: (a) 100 kHz; (b) 150 kHz; (c) 200 kHz

Figure 12. Normalized amplitudes of mode conversion indicator (MCI) for specimen No. 2 with Teflon inserts with diameter 10 mm for frequencies
Similar plots of MCI values for specimen No. 2 with smaller Teflon inserts (diameter 10 mm) were presented in Figure 12. In this case, the same monotonic trend for the values could be noticed. These results indicate that the intensity of S0/A0' mode conversion depends on the depth at which discontinuity is located. Mode conversion S0/A0' phenomenon was noticed for both diameters of Teflon inserts.

5.2 Impact damage influence

Analysis of mode conversion phenomenon in Section 5.1 was related to specimens with Teflon inserts treated as artificial delaminations in the GFRP composite panels. In this section, more realistic scenario related to impact damage and its influence on mode conversion will be investigated. This research will be related to specimen No. 3 (Figure 1(c)) with three locations of impact damage. Three impact energies were considered 5, 10 and 15 J.

In Figure 13, result in the form of full wavefield pattern for specimen No. 3 (Figure 1(c)) including impact damage is presented. This measurement was performed for excitation frequency 200 kHz. Impacts were made out on the top surface of the sample while as measurements were performed on the bottom side of specimen. In this case, the propagation of fundamental S0 and A0 mode is observed. Moreover, S0/A0' mode conversions at the locations of impact damage with energies 10 and 15 J are observed (compare with Figure 1(c)). The amplitude of A0' mode is higher for impact damage with higher energy (15 J).

In Figures 14(a) and 15(a), examples of raw time-space signals gathered along lines L3 and L4 including impact damage with energies 10 and 15 J created in sample No. 3 were presented. The frequency of excitation was equal to 200 kHz. The propagation of fundamental modes S0 and A0 can be noticed. However, the effects of S0/A0' mode conversion are hard to noticed. The locations of impact damage were indicated by vertical line. In Figures 14(b) and 15(b), these signals after mode filtration were presented. Now, only the propagation of S0 mode is visible. The amplitudes of S0 modes in two directions also slightly differ.

Finally in Figures 14(c) and 15(c), signals after mode filtration showing only the backward propagation of A0b mode are presented. In the case of Figure 15(c) for the case of impact with energy 15 J, the S0/A0' mode conversion effect is clearly visible. However, for
the case of impact with energy 10 J (Figure 14(c)), the amplitude of $A_0'$ generated after conversions is at the level of noises.

Next, values of MCI were calculated for lines L1–L4 (Figure 1(c)). Line L1 is referential (without impact), and lines L2–L4 contain the locations of impacts with energies 5, 10 and 15 J, respectively. MCI values were extracted for all investigated excitation frequencies: 100, 150 and 200 kHz. The results are presented in Table II.

In the case of frequency 100 kHz, MCI values for referential case and impact with energy 5 J are similar. Higher values of MCI were obtained in the case of impact with energies 10 and 15 J. This is clearly visible in Figure 16(a) where normalized values of MCI were plotted for all investigated cases for frequency 100 kHz. Normalization was performed in respect to the

**Figure 14.** Space signals gathered along line including impact with energy 10 J
Figure 15. Space signals gathered along line including impact with energy 15 J.

Notes: (a) Raw; (b) filtered – propagation of S0 mode; (c) filtered – backward propagation of A0; Specimen No. 3

Table II. MCI values for impact damage with different energies and wave frequencies

<table>
<thead>
<tr>
<th>Discontinuity</th>
<th>100 kHz</th>
<th>Frequency 150 kHz</th>
<th>200 kHz</th>
</tr>
</thead>
<tbody>
<tr>
<td>REF</td>
<td>0.085</td>
<td>0.037</td>
<td>0.037</td>
</tr>
<tr>
<td>I1 – 5 J</td>
<td>0.097</td>
<td>0.049</td>
<td>0.034</td>
</tr>
<tr>
<td>I2 – 10 J</td>
<td>0.258</td>
<td>0.051</td>
<td>0.045</td>
</tr>
<tr>
<td>I3 – 15 J</td>
<td>0.272</td>
<td>0.247</td>
<td>0.191</td>
</tr>
</tbody>
</table>
maximum value of MCI achieved for impact with largest energy – 15 J. In the case of frequency 100 kHz based on S0/A0’ mode conversion, it is possible to detect impacts with energies 10 and 15 J. In the case of these energies, MCI differs significantly from referential state. In the case of frequency 150 kHz, MCI values are similar for referential case and impacts 5 and 10 J. Only in the case of impact 15 J, values of indicator significantly differ from referential state. This is clearly visible in Figure 16(b) where the normalized values of MCI for all cases for frequency 150 kHz were plotted.

Similar situation was observed for frequency 200 kHz where MCI values are similar for referential case and impacts 5 and 10 J. This is visible in the values of Table II and in plot presented in Figure 16(c). This could also be noticed comparing results presented in Figures 14(c) and 15(c) where the amplitude of A0’ mode is much higher than noise only in the case of impact with energy 15 J (Figure 15(c)). The value of MCI is much higher than for referential case only for impact 15 J.

6. Conclusions

The results of conducted research indicate that Teflon inserts (internal located discontinuity) and impact damage (surface located discontinuity) are sources of S0/A0’ mode conversion. The mode conversion phenomenon could be utilized as the indicator of discontinuities located in structures with uniform thickness.

The effects of mode conversion could be observed even if the discontinuity is located at the surface opposite to surface where the measurements are taken. This shows potential for non-destructive testing in the case of damage located in the surface of structure which is not available for measurements.

In the case of Teflon inserts, the amplitude of A0’ mode generated after the conversion of S0 mode depends on the depth at which insert is located in the composite specimen. The effects of S0/A0’ mode conversion were observed for Teflon insert with diameters 10 and 20 mm for all investigated depths and wave frequencies (100, 150 and 200 kHz). The results show that depth at which Teflon insert is located can be assessed based on the values of mode conversion indicator MCI.

The amplitude of A0’ mode is also related to impact energy. Minimal impact energy causing visible effects of mode conversions is equal to 10 J in the case of frequency 100 kHz and 15 J for frequencies 150 and 200 kHz.

The results of conducted research show that S0/A0’ mode conversion and proposed conversion indicator MCI could be utilized for the detection of discontinuities like artificial delamination or impact damage.

Obtained results presented in this paper should be validated for a larger set of samples and experimental cases.

![Figure 16](image-url)

**Notes:** (a) 100 kHz; (b) 150 kHz; (c) 200 kHz

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References


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Capability of non-destructive techniques in evaluating damage to composite sandwich structures

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Abstract

Purpose – Defects can be caused by a number of factors, such as maintenance damage, ground handling and foreign objects thrown up from runways during an in-service use of composite aerospace structures. Sandwich structures are capable of absorbing large amounts of energy under impact loads, resulting in high structural crashworthiness. This situation is one of the many reasons why sandwich structures are extensively used in many aerospace applications nowadays. Their non-destructive inspection is often more complex. Hence, the choice of a suitable non-destructive testing (NDT) method can play a key role in successful damage detection. The paper aims to discuss these issues.

Design/methodology/approach – A comparison of detection capabilities of selected C-scan NDT methods applicable for inspections of sandwich structures was performed using water-squirt, air-coupled and pitch-catch (PC) ultrasonic techniques, supplemented by laser shearography (LS).

Findings – Test results showed that the water-squirt and PC techniques are the most suitable methods for core damage evaluation. Meanwhile, the air-coupled method showed lower sensitivity for the detection of several artificial defects and impact damage in honeycomb sandwiches when unfocussed transducers were used. LS can detect most of the defects in the panels, but it has lower sensitivity and resolution for honeycomb core-type sandwiches.

Originality/value – This study quantitatively compared the damage size indication capabilities of sandwich structures by using various NDT techniques. Results of the realised tests can be used for successful selection of a suitable NDT method. Combinations of the presented methods revealed most defects.

Keywords Aerospace, Sandwich, Carbon fibre-reinforced composite, Non-destructive testing, Ultrasonic, Laser shearography

Paper type Research paper

1. Introduction

A structural sandwich is a special form of a laminated composite material, which typically consists of thin stiff face sheets separated by a thicker lightweight core (Herrmann et al., 2005). Given this advantageous arrangement, the structure may exhibit low weight, high flexural stiffness and high buckling strength simultaneously. Hence, sandwich structures are extensively used in many aerospace applications nowadays. Regarding the non-destructive testing (NDT), these structures are often more complex, presenting major challenges in some cases (Heida and Platenkamp, 2013; Takeda et al., 2007; Hsu, 2009).

Presently, coin tap tests (Prior, 2016) or manual acousto-ultrasonic contact A-scan type methods (pitch-catch (PC)) are still widely used. However, these types of inspections can be significantly affected by human factors and their output capabilities are very limited.

Over the last few decades, utilisation of automatisation and C-scan methods to improve probability of detection and quality of NDT inspections has become a general trend. C-scan imaging (2D data presentation) significantly increases the quality of inspections and allows detection of very slow and small changes in properties of tested structures (Roach and Rice, 2016). Currently, C-scans are commonly used for standard immersion ultrasonic inspections.

The research leading to these results was funded by the institutional support for the long-term conceptual development of a research organisation provided by the Ministry of Industry and Trade of the Czech Republic.
of monolithic composite structures, and they can also be used for acousto-ultrasonic dry-coupled methods, such as PC. Implementation of C-scans with this technique offers a considerable potential for the improvement in the quality of NDT of bonded joints and sandwich structures.

Sandwich structures are typically characterised by considerably high attenuation of ultrasound waves of frequencies above 1 MHz. This fact sometimes complicates usage of conventional ultrasonic methods. That is why various alternative methods and, in particular, vibrational acousto-ultrasonic methods with frequencies less than 100 kHz were developed. At the present time, a range of acoustic multi-mode NDT systems are commercially available such as Bondmaster (Olympus), Bondascope (NDT Systems), Sondicator (Zetec), etc. The systems usually include several acoustic and ultrasonic methods. Typically, five inspection methods (resonance, three modes of PC, MIA) are available. The simple PC mode, which uses no couplant, is generally preferred for sandwich structures when applicable. The PC probe in its simplest form normally incorporates two transducers, one configured as a dedicated drive (pitch) and the other as a dedicated receive channel (catch). The first transducer transmits a burst of acoustic energy into the test part, and the second one receives the sound propagated across the test part between the probe tips. All parts of sandwich structures including any defects beneath the two probe tips (transducers) affect the characteristics of the acoustic energy that is transmitted between the two tips. These characteristics can be displayed in terms of phase and amplitude change. Excitation in various commercially available devices is usually a short tone burst or swept sine chirp within the frequency range of 2–70 kHz (Dickinson and Thwaites, 2002).

Some research study performed previously showed that the detectability of some kinds of defects in sandwich structures is highly frequency-dependent. That is why it is generally appropriate to use a broadband technique for an initial scan and then to refine the probe pulse to give more information about the detected defects (Dickinson and Fletcher, 2009).

PC is a relatively low-cost, couplant-free method. The great advantage of the method is ability to detect far-side defects. On the other side it is not suitable for inspections of very rough or damaged surfaces. The measuring tips of the PC probe can be easily damaged in the case of some open surface defects. It can also show a limited detection performance for in-service defects in some cases (Heida and Platenkamp, 2013).

Contactless methods by using air-coupled low-frequency (LF) (Peters et al., 2003) or water squirter-based ultrasonic transducers are also suitable for the type of tested structures. They can be successfully used for inspecting materials which cannot be inspected using the total immersion method, owing to damage possibility from water ingress.

An alternative to total immersion method is the water-squirt technique in which the ultrasound is coupled by a water jet applied by specially designed probes (e.g. Roye and Schieke, 2006). This propagates the ultrasound along a narrow column of water, which is projected onto the test part surface. The water flows down of surface and it is then taken off. In this way, the exposure of the inspected object to water can be considerably reduced. Both the pulse-echo (PE) and through-transmission technique (TT) can be implemented in water-squirt method (Segreto et al., 2018). Water-squirt systems are often used for testing of high attenuating materials like sandwich structures (e.g. NOMEX) or GLARE (GLass-fibre REinforced aluminium). TT testing can be performed in single channel, dual channel, multi-channel or even phased array setup (Schwabe et al., 2010). It can be used for very large test parts when total submersion is impractical because of the excessive cost of immersion tank or when the material of the test part cannot be directly immersed into water. However, water coupling delivers also disadvantages like pressure variations, air-bubbles, lime scales, algae and possible corrosion of the scanner mechanism. Water-coupled UT can be also difficult and time-consuming to implement on complex shaped parts.
Air-coupled ultrasonic inspection has the distinct advantage of being couplant-free, but also suffers from several significant disadvantages. The main limitation of the method is the large reflection loss at the air-solid interface and the large attenuation of high-frequency ultrasound in air. If the sound has to move between the test part and air, only around 1 per cent of the sound energy is transmitted. Thus after four transitions very little sound energy is left. Typically the overall path loss may be 100 dB higher using air as a couplant than when water is used. Because the frequency attenuation in air increases exponentially with the frequency, air-coupled ultrasonic is carried out in the frequency range below 1 MHz. The scanning rate of air-coupled ultrasound is limited by the relatively low pulse repetition frequency (PRF), which is possible with the long transit times in air. Typically, a PRF of around 200 Hz is possible. If we accept a relatively coarse scan pitch of 3 mm as acceptable, this implies a maximum linear speed of 600 mm/s and in two dimensions, an absolute minimum time of around 10 min to scan one square metre (Buckley, 2000; Hillger et al., 2014).

The arrangement of the probes can be one or two-sided using techniques like TT, shear waves, plate waves or pseudo PE technique with sound barrier between transmitter and receiver. Air-coupled ultrasonic testing has experienced rapid growth within the last years. For example, annular array transmitters and receivers for air-coupled ultrasound testing were developed. They are based on dice-and-fill composites. The electronically focussing of these transducers significantly enhances the lateral resolution and of the measured C-scans (Steinhausen et al., 2016). Also, the utilisation of cellular polypropylene offers new possibilities for construction of new types of air-coupled transducers. A twin probe with a transmitter and a receiver placed on the same ferroelectret film was developed (Gaal and Kotschate, 2018).

Another applicable contactless method is laser shearography (LS). It is a very fast full-field real time optical technique that uses interferometric comparison of a test object in two states, that is, neutral and loaded (Kadlec and Růžek, 2012; Růžek et al., 2006; Gryzagoridis and Findeis, 2008). The method requires no coupling or complex scanning equipment. The loading can be created by different excitation methods such as vacuum, thermal, vibration (Findeis and Gryzagoridis, 2014) or mechanical load, which induces deformation of the inspected object. Shearography systems can operate with extraordinary efficiency reaching throughputs from 25 to 1,200 sq. ft per hour, 2.5 to 120 times the typical 10 sq. ft/hour inspection rate for ultrasonic C-Scan (Newman, 2008; Pieczonka et al., 2014).

LS has also several limitations. First, it can only be applied on objects in which the surface roughness is of the order of one wavelength of light or more. Objects with smooth surface can produce random interference that may tend to wrong results of inspection. Next drawback is that the interpretation of the results is quite complex and requires considerable experience. Because the inspected object must be loaded to reveal defects, there is a potential possibility that damage might be incurred as a result of the inspection procedure (Kapadia, 2006). Some researchers reported also that the detectability of delaminations and disbonds in honeycomb sandwiches can be poor to moderate when compared with ultrasonic inspection. The detectability of some defects rapidly decreases with increasing defect depth (Heida and Platenkamp, 2013). However, LS seems to be a very promising NDT method with high potential for the inspection of sandwich structures.

This study aims to quantitatively compare the damage size indication capabilities of sandwich structures by using various NDT techniques.

2. Experiments and methods
2.1 Experimental material and boundary conditions
Four reference panels (500 mm × 220 mm in size) with carbon fibre-reinforced plastic (CFRP) skins (1.3 mm thick) (prepreg EHKF420-C20-45, Gurit) and honeycomb (Nomex, 15 mm) or foam core (10 mm, Airex R82.60) were used for the NDT experiments.
EHKF420-C20-45 is carbon fibre-reinforced thermosetting prepreg consisting of 3k HTA carbon fabric (204 g/m², twill 2/2) impregnated with 40 per cent epoxy resin EH420 with curing temperature 120°C. The reference panels were manufactured in two steps. All skins were initially produced by autoclave technology, inspected by immersion ultrasonic C-scan method, and subsequently bonded to core in a heated press.

Two panels from both core materials were verified to have artificial defects. Figure 1 shows the honeycomb inspection reference panel IRP1 with artificial defects. The design of panel IRP2 with foam core was very similar but potted core cells were replaced with 3 mm deep flat-bottom holes.

Artificial defects imitating foreign bodies and delaminations in the skin were realised using one- and two-layer Kapton foil inserts placed between the middle layers of the top skin. Four insert sizes were employed, and their dimensions varied from $3 \text{ mm} \times 3 \text{ mm}$ to $25 \text{ mm} \times 25 \text{ mm}$.

The skin-to-core disbonds defects were simulated in two ways. In the first method, two-layer Kapton foil inserts placed between the foil adhesive and honeycomb core were used. These inserts were situated at a larger distance (min. of 50 mm) from the edges of the reference panels. In the second method, pull tabs with the length of 25 mm and the width of 3, 6, 12 and 25 mm were used for the separation of the skin from the core at the edges of the panels. The tabs were removed after curing.

Other defects and manufacturing irregularities, such as potted core cells, bonded core joints and cracked honeycomb core, were also simulated. The cracked honeycomb core defect was realized by cutting through the core parallel to the inspection surface at the two corners of the reference panels.

The other two panels included real visible impact damage (VID) or barely visible impact damage (BVID) with impact energy range of 2.5–15 J by using two types of impactor diameters (12.7 and 25.4 mm). The impacted panels H1 and F1 had the same dimensions as the IRP panels. Panel H1 after impacting is presented in Plate 1. The BVID impact damage was defined as an impact damage with an initial dent depth of less than 1.0 mm. Impacting was performed on an SUPR drop impact tower in accordance with ASTM D7136M. The drop tester was equipped with an accelerometer and anti-rebound system (Figure 2). The impacts were performed in two rows with an 80 mm spacing. The data obtained from the accelerometer were used for the complete analysis of the behaviour of the specimens.
based on the information related to the real absorbed energy. Dent depth was measured using an indicator gauge immediately after the impact and after relaxing (30 min). Indent depth after relaxing vs total impact energy measured on panels H1 and F1 is presented in Figure 3, indicating a difference in results due to the impactor diameter. Generally, dents induced by smaller impactor diameter were markedly deeper for the same impact energy. Limit impact energy for 0.5” impactor was approximately 5.1 J for both types of core material. All damage induced by impacts with energy under the limit can be indicated as BVID because the dent depth was less than 1.0 mm. The impact energy limit for 1” impactor was 8.4 J for the honeycomb core sandwich panel and 9.2 J for the foam core panel.
2.2 Water-squirt technique

The panels were initially scanned using the PE and TT water-squirt techniques, in which the ultrasound was coupled with a narrow water jet. Universal squirter nozzles made by 3D printing and single-element pencil case-type unfocussed transducers (Olympus V323-N-SU 2.25 MHz (TT) and V310-N-SU 5 MHz (PE)) were used (Plate 2(a)). The expected high sensitivity and resolution were confirmed, but C-scans of the panels with foam core cannot be conducted because a suitable pencil case-type probe with frequency below 1 MHz was not available.

2.3 PC technique

Subsequent scans were performed via the PC method by using an innovative Omniscan Bond Tester and the standard ECA module. Utilisation of ECA module for bond testing (BT) is possible thanks to several electrical and conceptual similarities of BT and eddy current testing (ECT). Both methods use very similar frequency range and circuitry. The main difference lies in voltage level used. PC probes need a little higher voltage than it is used for ECT. That is why special voltage booster (BT probe adaptor) has to be used. Both methods also use XY flying dot/impedance plane display for signal representation. When the PC probe is used with an OmniScan MX ECA flaw detector, dedicated MXB software and probe adaptor, it is possible easily turn EC flaw detector into a BT instrument with intuitive C-scan capabilities. A very important advantage of the solution based on Omniscan MX

Notes: (a) TT water-squirt technique; (b) PC method utilising an eddy current array (ECA) module
ECA flaw detector is that it can use eight frequencies simultaneously, greatly increasing the probability of detection and sizing performance. Every frequency recorded can be analysed using the amplitude or phase setting, giving the inspector up to 16 C-scans to analyse.

SPO-5629-PHV broadband high-voltage PC probe, with measuring tip spacing of 12.7 mm, was used (Plate 2(b)). C-scans were performed for test frequencies of 8–42 kHz during the optimisation phase. Superior results were obtained in the frequency of 11–13 kHz and probe drive of 4 V.

2.4 Air-coupled ultrasound technique
The air-coupled LF technique was implemented at frequencies below 1 MHz and is suitable for contactless inspections of high ultrasonic attenuation materials which did not contact with water. Special dual element transducers in two separate housings working separately as the transmitter and receiver are required for most applications (Plate 3(a)). Thus, a standard PE testing is not feasible.

STARMAN DIO 2000 LF system, analysis software C-scan and unfocussed 120 kHz probes PLN-19-012 with a nominal element size of 19 mm were used for the experiments.

2.5 Laser shearography
LS is a large-area optical inspection system that uses laser light to detect very slight surface deformations owing to subsurface discontinuities (Newman, 2005; Cawley, 2006). Shearography uses the interference of monochromatic laser light to detect surface displacements in the order of nanometres (Plate 3(b)). It is an “active” NDT method and measures the material’s reaction to an applied stress. The test panels were loaded with heat energy (500 W from a 20 cm distance for 5 s). Horizontal shear was used and the wrapped phase image was evaluated.

3. Results
The water-squirt ultrasonic technique, air-coupled ultrasound, PC and LS were used to detect defects in four composite sandwich panels with honeycomb or foam core. Two panels included different types of artificial defects and two panels included real BVID and VID impact damage.

The selected final C-scans and a more detailed description of the results are provided in the following subsections.

Plate 3. Contactless NDT methods compared in this study

Notes: (a) Air-coupled ultrasound; (b) LS
3.1 Artificial defects

Figure 4 shows the C-scans of inspection reference panel IRP1 with artificial defects. All the defects in the test specimen were detected using a combination of the PE and TT water-squirt methods. The TT technique was insensitive to the detection of thin one-layer inserts (foreign bodies) in the skin.

Two types of C-scans were used for the PC method, namely, amplitude C-scans displaying colour variation based on the amplitude of the signal (ideal for disbonds and potted core detection) and phase C-scans that used a $0^\circ$–$360^\circ$ colour palette to display the changes in the phase angle (ideal for delamination and foreign body detection in skins). All disbonds, potted core cells and core damage were very well evident in the amplitude C-scan obtained at the optimised frequency of 13 kHz (Figure 4(c)). Two-layer inserts (delaminations) greater than 12 mm $\times$ 12 mm were well evident in the phase C-scan obtained at 42 kHz, but no one-layer inserts were detected.

Based on the obtained results, air-coupled LF method and LS cannot reliably detect two-layer inserts smaller than 25 mm $\times$ 25 mm and all one-layer inserts in honeycomb sandwich structures.

In addition to artificial defects, the compared NDT methods revealed also some manufacturing defects in the IRP1 panel (see Figures 4 and 5). These findings were confirmed using photos taken during manufacturing of the panel. Several cells of the honeycomb core were mechanically damaged and one cell was inadvertently potted by glue during manufacturing. All the tested NDT methods showed high sensitivity in case of the potted core

![Manufacturing defects](image)

**Notes:** (a) PE water-squirt technique (C-scan of the upper skin); (b) TT water-squirt technique; (c) PC; (d) TT air-coupled ultrasound; (e) LS
cell (only one potted cell was detected!), but they significantly differed in core damage
detection. TT water-squirt technique showed extremely high sensitive and ability to detect
even the smallest damage of the core cells. On the other hand, the LS method was practically
insensitive for the kind of defect.

Figure 6 shows the C-scans of the Airex foam core sandwich panel IRP2. However, the
TT C-scan by using the water-squirt technique cannot be performed because suitable
transducers with frequency under 1 MHz were not available. Thus, only the PE C-scan was
performed using this method (Figure 6(a)). The amplitude PE C-scan was very interesting
because all artificial defects were detected, except the foam core defects in the right-hand
corners of IRP2.

Moreover, the air-coupled LF method and LS can reliably detect all defects up to $6 \text{ mm} \times
6 \text{ mm}$. The defects with the size of $3 \text{ mm} \times 3 \text{ mm}$ were not reliably detected.

The detection capability of the PC method was significantly worse for the test panel in
comparison with the above-mentioned methods. The method did not reliably detect defects
smaller than $25 \text{ mm} \times 25 \text{ mm}$.

3.2 Impact damage
The subsequent part of the study aims to quantitatively compare real impact damage size
indication in sandwich structures by using various NDT techniques. Two sandwich panels,
H1 and F1 with Nomex honeycomb and Airex R82.60 foam core, respectively, were impacted
on the SUPR drop impact tower. Ten impacts with different impact energies were applied on
each test panel. BVID damage was achieved for total impact energies of less than 5 J for 0.5”
impactor and 7 J for 1” impactor.

Final C-scans of the impacted panels are shown in Figures 7 and 8. All the compared NDT
methods detected all impact damage, but these methods differed regarding damage sizes.

Several defects were detected also at the edges of the honeycomb sandwich panel H1 by
using the TT water-squirt and PC methods (Figure 7). These defects occurred during
manufacturing of the sandwich panel (bonding of the panel in a heated press) or more likely
during impacting due to excessive clamping force. Local compression damage of the
honeycomb core (core buckling or cracking) probably occurred as a result of excessive
clamping force applied by clamping fixture at the edges of the test panel (see Figure 2).
The honeycomb core damage was confirmed by the PC method. The presence of defects in
the skin in the areas was excluded by C-scan obtained by PE water-squirt technique after
impacting. Visibility and detectability of impact damage planned for the comparison study
were not influenced. That is why the defects at the edges did not affect results of the study.

The water-squirt technique was proven to be very sensitive to any damage to the
honeycomb core. This finding confirms some conclusions regarding the IRP1 reference
panel inspection. The comparison of the C-scans obtained via the TT and PE water-squirt

![Figure 5. Manufacturing defects found in the IRP1 panel](image)

**Notes:** (a) Top side, plotted core cell; (b) bottom side, honeycomb cells damage
techniques is very interesting. While the PE C-scan shows damage only in the top skin, TT C-scan presents the overall damage to the impacted sandwich. The difference in the indicated damage size is significant in this case. This finding shows that the honeycomb core is sensitive to impacts and the TT water-squirt method can detect even very small damage to the core.

Similar comparison of the capabilities of NDT methods was performed for the impacted foam core sandwich. Figure 8 shows the final C-scans of test panel F1 with Airex R82.60 foam core.

All impact damage indications in the C-scans obtained by using the four NDT methods were measured in horizontal and vertical directions. The average values for each indication are presented in Figure 9. Apparently, the TT water-squirt and PC methods were the most suitable for core damage evaluation. Meanwhile, LS and the PE water-squirt method detected the skin damage only.

4. Discussion
The comparison of the detection capabilities of the four C-scan NDT methods applicable for inspections of sandwich structures was performed in this study. Table I summarises the results in a comprehensive form.

The water-squirt ultrasonic, air-coupled ultrasound, PC and LS techniques were used for the detection of artificial defects and real impact damage in the previously mentioned sandwich panels. The water-squirt technique was proven to be extremely sensitive to any damage to the honeycomb core. The comparison of the C-scans obtained by using the TT and PE water-squirt techniques for the honeycomb sandwich panels showed significant differences regarding the indicated damage size. The comparison result showed that the honeycomb core is sensitive to impacts and that the TT water-squirt method can also detect even very minor damage to the core. Meanwhile, the impact results indicated that the larger diameter impactor produced significantly different damage states, which was also observed by Shyprykevich et al. (2003).

The PC method was also proven to be very sensitive, but its resolution was limited to defects greater than 12 mm × 12 mm because of the tip spacing on the PC probe (12.7 mm). It showed high penetration; far-side defects were readily detected. Some problems occurred during scanning of areas with opened surface damage. There was a risk that the measuring
tips could be trapped and damaged. The method was found to be unsuitable for inspections of very rough or damaged surfaces.

The air-coupled LF method showed a slightly lower sensitivity for the detection of several artificial defects and impact damage in honeycomb core-type sandwiches when
Unfocussed transducers were utilised. Utilisation of focussed transducers or apodisation of the receiver could significantly increase the method resolution (Peters et al., 2003). The sensitivity of the method was fully comparable with that of the PC method for foam core-type sandwiches. In addition, it was able to detect all defects in the foam core test panels. LS detected most of the artificial and real impact defects. However, its sensitivity was significantly worse for honeycomb sandwiches, and the sizing of impact damage was not reliable. The method was able to detect impact damage mostly in the skins, but it was not able to detect fine damage in the core.

5. Conclusion
Four sandwich test panels with CFRP skins (1.3 mm thick) (prepreg EHKF420-C20-45) and honeycomb (Nomex, 15 mm) or foam core (Airex R82.60, 10 mm) were used in the study focussed on comparison of detection capabilities of selected C-scan NDT methods. While the first two panels included different types of artificial defects imitating selected types of defects (delaminations and foreign objects in the skin, unbonded skin-core joint, potted core cells, cracked core, etc.), the other two panels included real impact damage. Ten impacts with different impact energies were applied to the each test panel, and BVID and VID damage were induced.

<table>
<thead>
<tr>
<th>Defect/method</th>
<th>Water-squirt PE/TT</th>
<th>Pitch-catch</th>
<th>TT air-coupled UT (unfocussed probes)</th>
<th>Laser shearography</th>
</tr>
</thead>
<tbody>
<tr>
<td>Two-layer inserts</td>
<td>++</td>
<td>++</td>
<td>++ &gt; 12 mm</td>
<td>++</td>
</tr>
<tr>
<td></td>
<td>++</td>
<td>++</td>
<td>+ &gt; 25 mm</td>
<td>++</td>
</tr>
<tr>
<td>One-layer inserts</td>
<td>++</td>
<td>++</td>
<td>–</td>
<td>++</td>
</tr>
<tr>
<td>Skin/core disbond</td>
<td>++</td>
<td>++</td>
<td>+ &gt; 12 mm</td>
<td>++</td>
</tr>
<tr>
<td>Cracked core</td>
<td>++</td>
<td>++</td>
<td>+</td>
<td>++</td>
</tr>
<tr>
<td>Potting</td>
<td>++</td>
<td>na</td>
<td>++ &gt; 12 mm</td>
<td>na</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>–</td>
<td>na</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>0/–</td>
<td>–</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>0</td>
<td>–</td>
</tr>
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<td></td>
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<tr>
<td></td>
<td></td>
<td></td>
<td>na</td>
<td>++</td>
</tr>
</tbody>
</table>

Table I. Detection effectivity overview of the evaluated NDT methods

Figure 9. Indication diameters for various NDT methods in relation to total impact energy

Composite sandwich structures
The comparison of the detection capabilities of water-squirt ultrasonic, air-coupled ultrasound, PC and LS techniques was performed. The main results obtain in this work can be summarised as follows:

- The best results were achieved using water-squirt ultrasonic method. All types of defects, including the smallest ones were found using a combination of TT and PE techniques. The method proved to be the best choice for inspection of the structures but some problems occurred in case of sandwiches with higher attenuation. UT probes with nominal frequency lower than 1 MHz were needed in case of foam core sandwiches.

- TT air-coupled LF method proved to be also very suitable for inspection of sandwiches, which are characterised in addition by higher attenuation or sensitivity to damage due to water ingress. Unfortunately, the method with used configuration (unfocussed 120 kHz probes) showed low sensitivity to detect smaller delaminations (< 12 mm × 12 mm) and one-layer foreign objects in case of sandwiches with honeycomb core.

- PC was tested as another “dry” alternative NDT method. Great advantage of the method is that it allows one-sided inspection and defect detection on the far side. Performed C-scans of the test panels disclosed insensitivity of the method to detect one-layer foreign objects (e.g. forgotten foils) in the skins. Also, the resolution of the method was significantly limited. It was not possible to reliably detect smaller defects than 12 mm × 12 mm because of tip spacing on the standard PC probe.

- LS showed several distinct advantages over other NDT methods. In particular, the contactless portable method proved the ability of quick inspection of large areas. Performed tests showed that the method is able to detect defects and impact damage larger than 6 mm × 6 mm in the skins, but it was not able to reliably detect some fine damage of the core. Problems with detection of some skin/core disbonds and one-layer foreign objects in the skin occurred especially in the case of the honeycomb type sandwich panel. Also, the sizing of real impact damage was not reliable.

- Significantly better detection results were achieved in case of foam core sandwiches. This was proved for all the used NDT methods and types of defects. The fact was probably caused by a different way of ultrasonic waves transmission (water-squirt, air-coupled LF and PC methods) and the heat load response for honeycomb and foam core.

- In frame of the NDT study, the impact behaviour of the sandwich panels made of above-mentioned materials was also investigated. The main goal of the activities was determination of limit impact energy causing VID or BVID damage and to obtain reference sandwich panels with real impact damage. Limit impact energy for 0.5” impactor was approximately 5.1 J for panels with both types of core material. All damage induced by impacts with energy under the limit was indicated as BVID (dent depth was less than 1.0 mm). The impact energy limit for 1” impactor was 8.4 J for the honeycomb core sandwich panel and 9.2 J for the foam core panel.

- Final impact damage of the sandwich panels was very complex. It consisted of different kinds of damage of the skins and the core such as delamination, matrix cracking, fibre breakage, disbonding, core buckling and cracking, etc. Detection capability and sensitivity of the compared NDT methods varied for the different kinds of damage. This fact effected determining the size of the real impact damage.

Above-mentioned results obtained in frame of the study showed that each NDT method has its own set of advantages and disadvantages and, therefore, some are better suited than
others for a particular type of sandwich structure and defect type. When selecting the suitable method for inspection, it is important that the NDT staff be aware of the type of core material that is used in the sandwich structure since the type and configuration of the core will affect the type of test method to be selected. The NDT specialists must select the method that will detect the defects with the highest accessible sensitivity and reliability.

Unfortunately, the situation is a little more complicated in case of impact damage of sandwich structures. The type of damage is very complex because it consists of several different kinds of defects. That is why the selection of a suitable NDT method can be a matter of compromise in some cases. Utilisation of combination of two or more NDT methods can be beneficial but cost effectiveness must also be taken into consideration. The results obtained in the study will be used to devise additional research focussing on structural health monitoring of impacted sandwich structures by using guided waves. R&D activities focussing on the utilisation of the air-coupled NF method for NDT inspection of sandwich structures are also planned in the future.

References


Further reading


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A numerical approach to determine fiber orientations around geometric discontinuities in designing against failure of GFRP laminates

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Abstract

Purpose – Determining fiber orientations around geometric discontinuities is challenging and simultaneously crucial when designing laminates against failure. The purpose of this paper is to present an approach for selecting the fiber orientations in the vicinity of a geometric discontinuity; more specifically round holes with edge cracks. Maximum stresses in the discontinuity region are calculated using Classical Lamination Theory (CLT) and the stress concentration factor for the aforementioned condition. The minimum moment to cause failure in a lamina is estimated using the Tsai–Hill and Tsai–Wu failure theories for a symmetric general stacking laminate. Fiber orientations around the discontinuity are obtained using the Tsai–Hill failure theory.

Design/methodology/approach – The current research focuses on a general stacking sequence laminate under three-point bending conditions. The laminate material is S2 fiber glass/epoxy. The concepts of mode I stress intensity factor and plastic zone radius are applied to decide the radius of the plastic zone, and stress concentration factor that multiplies the CLT stress distribution in the vicinity of the discontinuity. The magnitude of the minimum moment to cause failure in each ply is then estimated using the Tsai–Hill and Tsai–Wu failure theories, under the aforementioned stress concentration.

Findings – The findings of the study are as follows: it confirms the conclusions of previous research that the size and shape of the discontinuity have a significant effect on determining such orientations; the dimensions of the laminate and laminae not only affect the CLT results, but also the effect of the discontinuity in these results; and each lamina depending on its position in the laminate will have a different minimum load to cause failure and consequently, a different fiber orientation around the geometric discontinuity.

Originality/value – This paper discusses an important topic for the manufacturing and design against failure of Glass Fiber Reinforced Plastic (GFRP) laminated structures. The topic of introducing geometric discontinuities in unidirectional GFRP laminates is still a challenging one. This paper addresses these issues under 3pt bending conditions, a load condition rarely approached in literature. Therefore, it presents a fairly simple approach to strengthen geometric discontinuity regions without discontinuing fibers.

Keywords Classical Lamination Theory, Geometric discontinuities

Paper type Research paper

1. Introduction

Fiber reinforced laminates have become optimum material alternatives to metals in major applications and industries. A specific example with major applications in the current R&D sectors of modern industry is fiber reinforced laminated leaf springs in vehicle suspension systems, which constitute a focus area for the current research. This type of composites, having high strength to weight ratio, offers flexibility in selecting and optimizing the reinforcing phase (fibers) orientation in order to obtain the desired stiffness, mechanical properties, and performance of the structure. Research has shown that replacement of monoleaf steel springs with a laminated Glass Fiber Reinforced Plastic (GFRP) leaf significantly reduces the weight of the leaf spring by more than 20 percent, without sacrificing strength and performance (Shokrieh and Razaei, 2003).
Fibers hold the load carrying capacity of Fiber Reinforced Plastics (FRP). Long unidirectional fibers offer high strength and stiffness to laminated FRP beams. However, geometric discontinuities, such as holes, fillets, and tapered edges, interrupt the unidirectionality of the fibers and may affect the properties and performance of the laminate. Similar to metal beam structures, a high stress concentration area is observed around the discontinuity region. A simple FE analysis shows a 40 percent increase in the stresses developed around a 0.6 mm hole in a laminate of stacking sequence (0°),3 (Fragoudakis, 2012).

Composite laminate failure may be due to microscopic failure through fracture of individual material constituents (i.e. fibers and/or matrix), debonding at the constituents interface, or macroscopic delamination. As a result, during manufacturing of the geometric discontinuity catastrophic failure may occur due to any of the above failure modes.

Failure criteria and models help in the prediction of failure and evaluation of critical stresses. Different criteria predict different failure modes. For example, Hashin’s (1980) theory focuses on fiber failure, while newer approaches can predict fiber and interfiber failure (Ribeiro et al., 2013). The Waddoups–Eisenmann–Kaminski model and its modifications discussed in literature (Waddoups et al., 1971; Kannan et al., 2010) use the stress intensity factors to evaluate the strength of notched composite specimens. However, evaluating failure is one aspect of composite laminate performance, while designing against failure is another.

Choosing an optimum stacking sequence can improve the onset of first-ply failure in GFRP laminates. Previous research on laminated GFRP leaf springs examined the effect of fiber orientations and consequently stacking sequences and laminate symmetry showed that symmetric general stacking (0/45)s laminates can withstand higher loads before first-ply failure occurs, compared to other stacking sequence configurations such as cross-ply (90/0)s or (60/90/45)s (Fragoudakis, 2012; Fragoudakis and Gallagher, 2017). Consequently, choosing the appropriate stacking sequence in the vicinity of the discontinuity is of importance.

As the fibers are interrupted at the discontinuity, in order to strengthen the vicinity of the region different fiber orientations should be selected around it. Selection of fiber orientation in this region is a challenging process and depends on a variety of parameters. Different methods have been presented in literature, in an effort to design algorithms to determine such orientations. Goteti and Reddy (2014) used Classical Lamination Theory (CLT) and the stress intensity around a circular hole, to examine the effect of fiber angle, hole size, and volume fraction of the reinforcing phase on the stress concentration around a hole. Sharma determined the stress concentration around circular/elliptical/triangular cutouts using Muskhelishvili’s complex variable method and fiber orientation as inputs (Sharma, 2011). An effort to fully determine the fiber orientation in a broader domain around the discontinuity, while keeping the lamina fiber orientation unidirectional far from the prescribed domain around the discontinuity, is given by Huang and Haftka (2005). All of the above works agree that the parameters affecting fiber orientation around discontinuities, such as holes, depend on the size of the discontinuity, the load type and direction, as well as volume fraction of the fibers, which is widely known to affect the mechanical properties of the fiber/epoxy material, as determined by the Rules of Mixtures equations (Fragoudakis, 2018).

This paper discusses a slightly different approach to determine fiber orientation around circular discontinuities, focusing on the immediate vicinity of the discontinuity and not affecting the fiber orientations past the plastic region of the discontinuity. In this region, the stacking sequence remains intact in order to provide the laminate with the appropriate performance as expected by the chosen stacking sequence (Fragoudakis, 2012). The loading condition examined is not axial loading as in the aforementioned research, but rather three-point bending. This loading condition is chosen to be examined as it is the main loading condition of rear leaf springs in heavy duty vehicles, which constitute the focus of the paper as well as previous research that showed general stacking sequence as an
appropriate selection for leaf springs (Fragoudakis, 2012); and it is a loading condition not widely examined in the literature as is axial loading. The approach focuses on using CLT in conjunction with the Tsai–Hill and Tsai–Wu failure theories to determine the minimum moment to cause first-ply failure in the laminate in the absence of a discontinuity; and the effect of the geometric stress concentration factor under bending, to determine the moment to cause failure in the presence of a circular hole. The fiber orientation in the vicinity of the hole is determined when this minimum moment is applied.

Finally, while the majority of the aforementioned work is not concerned with the interruption of the fibers at the discontinuity, i.e. fibers start at the edge of the hole or cutout, this paper is concerned with the viability of fibers and makes an effort to predict orientations where fibers run uninterrupted along the length of the laminate, therefore, maintaining their continuity and unidirectionality.

2. Choosing the fiber orientation around a circular hole

2.1 Modeling of the laminate beam

CLT determines the stress distribution in a semi-infinite beam. As a result, it is important to determine the number of laminae in the laminate, the thickness of each lamina and the fiber orientation in each lamina.

To determine the minimum moment to cause failure in the laminate, a beam with no discontinuities and unidirectional fibers is considered. The laminate is considered to be symmetric, with a general stacking sequence (0/45/0)s (Figure 1). A total of six layers, each of thickness 1 mm, are considered. The number of layers and their thickness are chosen arbitrarily to facilitate calculations. Typically, GFRP laminae tend to be thinner than 1 mm. The number of layers in the laminate has a significant effect on the stiffness and consequently the stress distribution in the laminate. The beam material is S2 glass fiber/epoxy at 55 percent volume fraction, a typical material in GFRP leaf spring research (Fragoudakis, 2012). This is a medium to high stiffness GFRP, with properties that are commonly used in the industry. The mechanical properties of the GFRP material are given in Table I. As it can be seen from the properties of the GFRP, the lamina is considered to be transversely isotropic, with direction 1 along the fibers and the isotropy plane being 23.

![Figure 1. Stacking sequence](image)

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_1$</td>
<td>34 GPa</td>
</tr>
<tr>
<td>$E_2$</td>
<td>8.9 GPa</td>
</tr>
<tr>
<td>$G_{12}$</td>
<td>4.5 GPa</td>
</tr>
<tr>
<td>$v_{12}$</td>
<td>0.27</td>
</tr>
<tr>
<td>$X$ (longitudinal tensile strength)</td>
<td>2,000 MPa</td>
</tr>
<tr>
<td>$X'$ (longitudinal compressive strength)</td>
<td>1,240 MPa</td>
</tr>
<tr>
<td>$Y$ (transverse tensile strength)</td>
<td>49 MPa</td>
</tr>
<tr>
<td>$Y'$ (transverse compressive strength)</td>
<td>158 MPa</td>
</tr>
<tr>
<td>$S$ (shear strength)</td>
<td>63 MPa</td>
</tr>
</tbody>
</table>

Table I. Properties of S2 glass fiber/epoxy
A beam of the exact same dimensions, stacking sequence and fiber orientations with a central circular hole of 1 cm in diameter is considered in the investigation of the fiber orientations around the discontinuity (Figure 2).

In both of the above cases, the loading condition is a bending moment, which is left to be determined using the two failure theories.

2.2 Designing the discontinuity region

The region around a geometric discontinuity affected by the maximum stress concentration can be approximated to the plastic zone. To determine this zone, a very small crack is assumed just on the verge of the hole. This crack, in the epoxy matrix of the GFRP, has the potential to propagate and lead to fracture of the matrix, which is one of the failure modes in FRP composites. Assuming a very small crack means that the ratio of crack length to hole radius approaches 0. Using this ratio, the mode I stress intensity factor \( K_I \) can be calculated from Equation (1), the equation for cracks at the edge of a circular hole. Assuming that the stress intensity factor becomes equal to the critical stress intensity factor \( K_c \), the instant the crack begins to propagate and cause failure, the radius of the plastic zone \( r_y \) can be estimated by Equation (2):

\[
K_I = \sigma \sqrt{\pi \alpha f(z/r)}, \quad (1)
\]

\[
r_y = \frac{1}{2\pi} \left( \frac{K_c}{\sigma_y} \right)^2, \quad (2)
\]

where \( \alpha \) is the crack length; \( r \) the hole radius; and \( \sigma_y \) the applied yield stress.

The radius of the plastic zone begins at the crack tip. Assuming that the crack length is very small, this radius can be assumed to begin on the rim of the circular hole and determine the extent of the crucial vicinity around the discontinuity. At this region, a different reinforcing stacking sequence, from that of the remaining lamina, should be selected. The above equations clearly show the effect of the hole size \( r \), to the selection of this zone.

2.3 CLT, failure theories, and maximum stress

CLT uses the information of the material properties and fiber orientation in each lamina to generate a stiffness matrix \( [Q] \) for every lamina (Equation (3)). The position of each lamina in the laminate \( (z) \), the lamina thickness \( (t) \), the loading of the laminate \( (F \) and/or \( M \) – in the case examined in this paper only a bending moment along the longitudinal
direction of the beam is considered) builds the stress distribution in the \( k \)th lamina (Equation (4)):

\[
\mathbf{\bar{Q}}_k = \begin{bmatrix}
\bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} \\
\bar{Q}_{21} & \bar{Q}_{22} & \bar{Q}_{26} \\
\bar{Q}_{61} & \bar{Q}_{62} & \bar{Q}_{66}
\end{bmatrix},
\]

(3)

\[
\begin{bmatrix}
\sigma_x \\
\sigma_y \\
\tau_{xy}
\end{bmatrix}_k = \begin{bmatrix}
\varepsilon_x^0 \\
\varepsilon_y^0 \\
\gamma_{xy}^0
\end{bmatrix}_k + \begin{bmatrix}
\kappa_x \\
\kappa_y \\
\kappa_{xy}
\end{bmatrix} + \begin{bmatrix}
x_x \\
y_y \\
\Delta T
\end{bmatrix}_k,
\]

(4)

where \( \varepsilon_{ij} \) are the mid-surface strains developed on the mid-surface plane, which divides the laminate into two equal top and bottom portions (Figure 1); \( \kappa_{ij} \) the laminate curvatures; and \( \alpha_{ij} \) the coefficient of thermal expansion, which accounts for the thermal strains developed in the laminate. Such strains may occur during manufacturing when \( \Delta T \) becomes significantly large and residual stresses may be induced between layers (i.e. laminae). Thermal effects are ignored in the current study. Moisture absorption may also affect the stress distribution in the lamina. However, hygral effects appear to be insignificant in S2 fibers, and therefore, are not accounted for in Equation (4) and the current investigation (Fragoudakis and Gallagher, 2017).

The above stress distribution is evaluated at global axis \((x, y)\) which remain the same for all laminae. However, since CLT focuses on planar analysis, and failure theories account for failure within a single lamina, the above stresses are translated to principal stresses, along the local axis of each lamina \((1, 2)\), where direction 1 is always along the fibers. The failure theories considered in this study are interactive failure theories, i.e. account for the interaction of stresses in different directions. To determine the minimum moment to cause failure in each ply, the Tsai–Hill (Equation (5)) and Tsai–Wu (Equation (6)) failure theories are considered:

\[
\frac{\sigma_1^2}{X^2} + \frac{\sigma_2^2}{Y^2} + \frac{\tau_{12}^2}{S^2} < 1,
\]

(5)

\[
F_{11}\sigma_1^2 + 2F_{11}\sigma_1\sigma_2 + F_{22}\sigma_2^2 + F_{66}\sigma_6^2 + F_1\sigma_1 + F_2\sigma_2 < 1,
\]

(6)

where the coefficients in the Tsai–Wu failure theory depend on the strengths of the material.

The stresses calculated in Equation (4) are the stresses of a single lamina, of unidirectional fibers and no geometric discontinuities. To account for these discontinuities, the geometric stress concentration factor \((K_t)\) is used to multiply these stresses and maximize them to account for the effect of the hole (Equation (7)):

\[
\sigma_{\text{max}} = K_t\sigma_k.
\]

(7)

For the aforementioned loading and dimensions of the laminate and hole \(K_t = 2.7\). However, this factor changes depending on the size of the hole and dimensions of the laminate. This is another area where the effects of the hole size not just to the stresses in the laminate, but also to the failure of each lamina, can be seen.
2.4 The optimization process

Using the equations and the constraints mentioned above, the smallest moments to cause failure in each lamina are calculated and given in Table II. These moments become the applied load to the plate with the central circular hole.

Table II shows the absolute values of the minimum moments. Tsai–Hill, with the minimum condition to cause failure when Equation (5) is equal to 1, gives two possible moments that can cause failure. They both have the same magnitude, while the set of two moments accounts for both clockwise and counterclockwise direction. On the other hand, for the same failure condition, the Tsai–Wu theory gives a positive and negative moment at each layer; however, their magnitudes are not the same. Table II shows the moment with the smaller magnitude. Either failure theory predicts that the 0° orientation is the strongest, and the symmetry of the matrix allows for a mirror image of the moments above and below the mid-surface plane. A final conclusion of this analysis is that Tsai–Wu tends to overestimate failure, as the lamina fails at lower loads, while Tsai–Hill tends to underestimate failure. Since failure theories are used just to approximate failure, without physical experiments it is not easy to choose one over the other.

Assuming that the discontinuity is at the center of a semi-infinite plate, the lamina can be treated as symmetric in both the $x$ and $y$ directions of its plane. To choose the number of points required around the hole, in order to determine the path of the fiber orientation, the finite element concept of appropriate seeds for geometric discontinuities is applied. A minimum of 16 seeds around a circle is recommended, as a result a total of 4 seeds in a quadrant are chosen. Following Huang and Haftka’s (2005) model of fiber orientation, outside the plastic zone the fiber orientation returns to that of the lamina, as prescribed from the general laminate stacking sequence. The number of seeds inside the plastic zone depends on the accuracy of the results and radius of the plastic zone. In this paper, the optimization for the fiber orientation around the hole is constrained to the first four seeds. The possible orientations for theses seeds can vary between 0° and 90°.

The failure theories, as applied above, assume a load carried along the direction of the fibers. However, in the vicinity of the discontinuity the fibers will be constantly changing orientation (along the new four orientations as prescribed by the seed selection). To account for this effect on the load, the minimum load factor is calculated as the positive root of a polynomial constructed based on the Tsai–Hill failure theory at the instance it is equal to 1 (Equation (8)):

\[
\left(\frac{\sigma_1^2}{X^2} + \frac{\sigma_2^2}{Y^2} + \frac{\tau_{12}^2}{S^2}\right)\rho^2 - 1 = 0. \tag{8}
\]

The positive load factor $\rho$ is a function of the orientations around the discontinuity.

3. Results

The positive roots of Equation (8) are examined in order to determine the appropriate orientations around the vicinity of the discontinuity. The orientations are estimated for the

<table>
<thead>
<tr>
<th>Lamina/Fiber orientation</th>
<th>Moment (Nm/m)</th>
<th>Moment (Nm/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/0°</td>
<td>269</td>
<td>171</td>
</tr>
<tr>
<td>2/45°</td>
<td>72</td>
<td>67</td>
</tr>
<tr>
<td>3/0°</td>
<td>808</td>
<td>512</td>
</tr>
<tr>
<td>4/0°</td>
<td>808</td>
<td>512</td>
</tr>
<tr>
<td>5/45°</td>
<td>72</td>
<td>67</td>
</tr>
<tr>
<td>6/0°</td>
<td>269</td>
<td>171</td>
</tr>
</tbody>
</table>

Table II. Absolute values of minimum moment to cause lamina failure.
case that the positive load factor becomes 1, and Equation (8) becomes Equation (5) at the first possible occurrence of failure. A similar approach can be followed using the Tsai–Wu criterion. Huang and Haftka follow this approach for the case of axial loading (Huang and Haftka, 2005; Sharma, 2011).

Table III shows possible orientations for the four angles around the discontinuity. It can be observed that there is a gap in the calculation of these orientations between 40° and 71°. Solutions indicate a very large range of orientations within these two bounds. To narrow this range, a genetic algorithm approach can be followed. However, this approach is outside the scope of this paper. Figure 3 plots orientations in the regions immediately next to the discontinuity, and far away from it, at a radius of 4 cm. This larger radius is calculated as the end of the plastic zone using Equation (2), for a crack of length 2.5 mm. To prevent the crack tip from coinciding with the fibers, the fiber orientations begin 5 mm away from the hole edge. This distance of 5 mm is taken as a safety precaution at twice the length of the crack. As a result, fibers begin away from the end of the plastic region and the risk of fiber cracking, one of the most catastrophic failure modes in composites, is avoided. However, a more detailed fracture mechanics analysis paired with experiments can help determine this distance with higher accuracy.

The fiber orientations of Figure 3 are those for Lamina 3, where the general fiber orientation is at 0°. Similar results are obtained for the other laminae (1, 4 and 6) whose general orientation is at 0°. On the other hand, laminae at 45° appear more challenging when determining these orientations. These cases are not examined in this paper, and results for these layers are expected to be found using a genetic algorithm, which could potentially reduce the wide range of solutions.

Although the above approach is constrained around the first quadrant of the hole, by symmetry it can be expanded to the remaining quadrants of the hole, and as a result produce fiber paths that are not interrupted along the whole domain of the lamina.
4. Conclusions

Reinforcing the regions around discontinuities can have a significant effect in the life and performance of a laminate. This paper presents a simple numerical approach to determine a range of possible fiber orientations around a circular hole. The approach depends on the fundamental definitions of stress intensity and stress concentration factors, while CLT and failure theories calculate the stress distribution and failure loading conditions. Analysis is constrained to loading under three-point bending conditions, which is not typically found in similar research in the literature.

The analysis and results show that for 0° fiber orientation laminae, the fiber orientations in the vicinity of the discontinuity are not affected by the location of the lamina in the laminate. For the case of 45° fiber orientation, there appears to be a very large range of such orientations for the vicinity of the discontinuity and further optimization is required.

However, the fiber orientations in the vicinity of the discontinuity are linked to the minimum moment to cause failure in the lamina. Consequently, the position of the lamina in the laminate plays a crucial role in determining these orientations. Therefore, a different set of such orientations should be expected at each lamina of different minimum moment to cause failure.

The preliminary analysis of this approach confirms the conclusions of previous research (Goteti and Reddy, 2014; Sharma, 2011; Huang and Haftka, 2005) that the size and shape of the discontinuity have a significant effect on determining such orientations. Additionally, the dimensions of the laminate and laminae not only affect the CLT results, but also the effect of the discontinuity in these results.

The findings of this research and its expansion to further fiber orientations and stacking sequences may have direct applications in the manufacturing of GFRP laminated leaf springs. Filament winding is among the most popular manufacturing processes of such leaf springs (Fragoudakis, 2012). The optimum stacking sequence around and farther away from the discontinuity may be used directly in this manufacturing process to build strong structures with reinforced regions around holes for bolts and/or tapered edges.

References


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Combining FEM and MD to simulate C₆₀/PA-12 nanocomposites

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Abstract
Purpose – The purpose of this paper is the computation of the elastic mechanical behaviour of the fullerene C₆₀ reinforced polyamide-12 (PA-12) via a two-stage numerical technique which combines the molecular dynamics (MD) method and the finite element method (FEM).

Design/methodology/approach – At the first stage, the proposed numerical scheme utilizes MD to characterize the pure PA-12 as well as a very small cubic unit cell containing a C₆₀ molecule, centrally positioned and surrounded by PA-12 molecular chains. At the second stage, a classical continuum mechanics (CM) analysis based on the FEM is adopted to approximate the elastic mechanical performance of the nanocomposite with significantly lower C₆₀ mass concentrations. According to the computed elastic properties arisen by the MD simulations, an equivalent solid element with the same size as the unit cell is developed. Then, a CM micromechanical representative volume element (RVE) of the C₆₀ reinforced PA-12 is modelled via FEM. The matrix phase of the RVE is discretized by using solid finite elements which represent the PA-12 mechanical behaviour predicted by MD, while the C₆₀ neighbouring location is meshed with the equivalent solid element.

Findings – Several multiscale simulations are performed to study the effect of the nanofiller mass fraction on the mechanical properties of the C₆₀ reinforced PA-12 composite. Comparisons with other corresponding experimental results are attempted, where possible, to test the performance of the proposed method.

Originality/value – The proposed numerical scheme allows accurate representation of atomistic interfacial effects between C₆₀ and PA-12 and simultaneously offers a significantly lower computational cost compared with the MD-only method.

Keywords Molecular dynamics, Finite element method, Nanocomposite, Polyamide, Fullerenes

Paper type Research paper

1. Introduction
Polymer composites are increasingly required for numerous industrial applications such as electrical and thermal insulators, strong but lightweight components in automotive and aeronautical applications and many others. However, the use of polymeric materials, generally, presents specific limitations which need to be overcome, and it seems that the discovery of carbon-based nanostructured materials showed the appropriate path for eliminating them.

Carbon is capable of forming many nanostructured allotropes such as graphene, carbon nanotubes (CNTs) and fullerenes which present excellent structural, electronic, optical, chemical and mechanical properties. Evidently, these carbon allotropes due to their unique material characteristics have a strong potential to be used as fillers in polymeric materials with the main purpose of enhancing their mechanical, thermal,
chemical and electrical performance. Nowadays, despite the high cost of the relevant manufacturing processes, carbon nanomaterial-based nanocomposites have already illustrated a significant growth potential in various industry sectors such as manufacturing, automotive, aerospace, electronics, green technologies and medical applications (Mittal et al., 2015). Nanocomposites which are reinforced with carbon allotropes belong to an emerging class of organic hybrid materials that exhibit improved mechanical properties at very low loading levels compared with conventional microcomposites (Tjong, 2006; Bhattacharya, 2016). Possibly, graphene and CNTs play the most promising role because of their superior structural and functional properties such as high aspect ratio, high mechanical strength and high electrical properties (Dinadayalane and Leszczynski, 2010). Both graphene and CNT-based nanocomposites usually possess the properties of fracture toughness, lightweight, fatigue strength and cost-effectiveness. Nanocomposites reinforced with graphene and CNTs are also utilized in energy storage and conversion devices as well as in biomedical applications such as gene delivery and nano-medicine. However, despite the fact that CNTs presents a similar mechanical behaviour with graphene, graphene nanocomposites have gained most of the research attention due to their better electrical and thermal performance.

Although graphene and CNT-based nanocomposites have attracted tremendous scientific and industrial interest, the reinforcement of conventional materials with fullerenes has not be studied to the same extent, despite the fact the specific nanomaterials have gained one of the first places in modern material science related to carbon since their discovery in 1985 (Kroto et al., 1985). The C$_{60}$ molecule (Krätschmer et al., 1990), which is the most known and representative of the fullerene family, has 12 pentagons located around the vertices of an icosahedron and 20 hexagon rings placed at the centres of icosahedral faces. This spherical molecular structure leads to unique mechanical property characteristics (Dinadayalane and Leszczynski, 2010; Ruoff and Ruoff, 1991; Peón-Escalante et al., 2014) which may be proved valuable for the development of new promising isotropic-like nanocomposites. In order to utilize the outstanding properties of fullerenes, they may be incorporated into polymeric matrices in the form of filler materials.

Considering the above, the present study aims to investigate the mechanical behaviour of a polymeric material reinforced with fullerenes C$_{60}$. In view of the fact that experimental procedures (Ballav, 2005; Wondmagegn and Curran, 2006; Bronnikov et al., 2017; Zuev, 2011, 2012) are excessively costly for the material characterization of polymer-fullerene nanocomposite systems, numerical techniques such as molecular dynamics (MD) (Sharma et al., 2015, 2016) seem to be very attractive for predicting and investigating their performance. Therefore, in the present study a hybrid numerical multiscale technique based on MD (Sharma et al., 2015, 2016), finite element method (FEM) (Tserpes and Papanikos, 2014) and unit cell methods (Marur, 2004) has been implemented to characterize the elastic mechanical properties of a polymeric material reinforced uniformly with molecules C$_{60}$. Among numerous candidate polymer materials, the polyamide-12 (PA-12) has been chosen as the matrix material of the investigated nanocomposite since it has already been experimentally demonstrated that the addition of graphene nanoplatelets or CNTs, which are analogous to fullerenes, at even small concentration up to 2 wt% within pure PA-12, may lead to significant improvements of its mechanical and electrical behaviour (Zuev, 2011; Chatterjee et al., 2011). To the authors’ best knowledge, there is a lack of studies which combine MD and FEM for evaluating the mechanical performance of materials and, yet, the majority of them (Higuchi et al., 2014) refer to the microscopic numerical investigation of conventional material problems.

2. MD-only formulation details
In all MD simulations performed in this work, a room temperature of 298 K is assumed. The dynamic analyses are performed by using the NPT ensemble (constant number of atoms,
pressure and temperature) with a constant time step of 1 fs. The simulations are realized by utilizing the “Materials Studio 2017” software package (Materials Studio 2017, 2016). The potential energy of the system is assumed to be described by the Condensed Phase Optimized Molecular Potential for Atomistic Simulation Studies (COMPASS) (Sun, 1998), the first and only ab initio force field that enables an accurate and simultaneous prediction of various gas-phase and condensed phase properties of organic and inorganic materials. The total energy function in COMPASS force field is composed of 12 terms including valence and non-bonded interaction terms as follows (Sun, 1998):

\[
E_{\text{total}} = \sum_{b} \left[ k_2(b-b_0)^2 + k_3(b-b_0)^3 + k_4(b-b_0)^4 \right] + \sum_{\theta} \left[ k_2(\theta-\theta_0)^2 + k_3(\theta-\theta_0)^3 \right] + \sum_{\psi} \left[ k_1(1- \cos \phi) + k_2(1- \cos 2\phi) + k_3(1- \cos 3\phi) \right] + \sum_{\lambda} k_2\lambda^2
\]

\[
+ \sum_{b,\theta} k(b-b_0)(b'-b'_0) + \sum_{b,\theta} [k(b-b_0)(\theta-\theta_0) + \sum_{b,\psi} (b-b_0)[k_1 \cos \phi + k_2 \cos 2\phi + k_3 \cos 3\phi]
\]

\[
+ \sum_{\theta,\theta'} k(\theta-\theta_0)(\theta'-\theta'_0) + \sum_{\theta,\theta',\phi} k(\theta-\theta_0)(\theta'-\theta'_0) \cos \phi
\]

\[
+ \sum_{ij} \frac{q_i q_j}{r_{ij}} + \sum_{ij} \epsilon_{ij} \left[ 2 \left( \frac{r_{ij}}{r_{ij}} \right)^6 - 3 \left( \frac{r_{ij}}{r_{ij}} \right)^{10} \right].
\]

The first four terms in this equation are sums that describe the energy needed to stretch bonds \((b)\), bend angles \((\theta)\) away from their reference values denoted by subscript 0, rotate torsion angles \((\phi)\) by twisting atoms about the bond axis that determines the torsion angle and distort planar atoms out of the plane \((\psi)\) formed by the atoms they are bonded to. The next five terms are cross-terms that corresponding to the interactions between the four types of internal coordinates. The final two terms are related with the non-bond Coulombic electrostatic interactions due to the charges \(q\) and the non-bond van der Waals (vdW) interactions as a sum of repulsive and attractive Lennard-Jones terms. Both terms are functions of the distance \(r_{ij}\) between atom pairs. The force field defines the stiffness like parameters \(k, k_1, k_2, k_3, k_4\) as well as the functional form of each term \(q_i, \epsilon_{ij}, r_{ij,0}\) in this equation. The force field also describes internal coordinates as a function of the Cartesian atomic coordinates. It should be noticed that the simulations that are carried out here, the atom-based summation method with cut-off radii of 12.5 Å and long range corrections are adopted in the calculation of the vdW interactions. Moreover, the electrostatic interactions are estimated by the Ewald summation method with the accuracy of 0.001 kcal/mol. Finally, the Nose thermostat and Berendsen barostat are adopted to control, respectively, the temperature and pressure of the system during the MD simulations.

3. Combined FEM–MD analysis of C_{60}/PA-12 nanocomposite

3.1 MD simulation of pure PA-12

Initially, the pure PA-12 matrix material is numerically investigated via a MD-only simulation. The PA-12 is assumed to be composed by polymer chains of five repeated monomer units each as Figure 1 depicts.

A polymeric unit cell of 35 PA-12 chains is constructed by assuming an initial density value of 0.9 g/cm³ and by using the “Amorphous Cell Construction” module.
The developed unit cell, which is illustrated in Figure 2, is geometrically optimized by performing energy minimization.

Afterwards, a dynamic analysis using “Forcite Dynamics” module (Materials Studio 2017, 2016) accompanied with the NPT ensemble is performed under an external pressure of 0.2 GPa for 20 ps, in order to minimize the close contacts between atoms. A similar NPT analysis follows, using an external pressure of 1 atm for 40 ps though, in order to achieve an equilibrium state of the unit cell. As Figure 3 presents, the final computed density of the polymeric material after the 60 ps time period converges to the value of $\rho_{PA-12} = 0.936 \text{ g/cm}^3$, which is a representative PA-12 density under typical environmental conditions.

The elastic mechanical properties of the pure PA-12 are finally computed by performing a static analysis on the equilibrated PA-12 unit cell, via the “Forcite Mechanical Properties” module (Materials Studio 2017, 2016) using a maximum strain of 0.01. The mechanical
properties of the unit cell are estimated via its Lamé parameters $\lambda, \mu$ which are computed by the MD static simulation and by assuming that the unit cell behaves as an isotropic medium. Then, the elastic modulus and the Poisson’s ratio of the unit cell are calculated, respectively, via the following well-known equations:

$$E = \frac{\mu(3\lambda + 2\mu)}{\lambda + \mu},$$  \hspace{1cm} (2)

$$v = \frac{\lambda}{2(\lambda + \mu)}. \hspace{1cm} (3)$$

The above described numerical procedure leads to an elastic modulus equal to 2.474 GPa and a Poisson’s ratio equal to 0.362 for the pure PA-12 unit cell. Note that the utilization of more than 35 PA-12 chains during the unit cell construction leads to negligibly different computational estimations, fact that proves that the chosen unit cell size offers converged numerical solutions.

It should also be mentioned that at the end of each mechanical property simulation, the “Forcite Mechanical Properties” module (Materials Studio 2017, 2016) additionally computes the velocities of sound observed within a unit cell by adopting the well-known formulas which associate wavespeeds with the lame’s parameters.

3.2 MD simulation of PA-12 reinforced with a high concentration of C$_{60}$

Here, C$_{60}$ molecule, which is shown in Figure 4, is introduced as reinforcement of the PA-12 at a rather high mass fraction of $M_{C_{60}} = 0.108$ (10.8 wt%).

The uniform dispersion of carbon nanofillers in the polymer matrix is a general requirement for accomplishing the desired mechanical and physical properties. Thus, in the present analysis, the reinforcements C$_{60}$ are assumed to be periodically and homogenously distributed in the PA-12 matrix material. Given the previous assumption, it becomes evident that the nanocomposite with the mass fraction $M_{C_{60}} = 0.108$ may be analyzed according to the abcdefgh cubic unit cell which is illustrated in Figure 5.

The specific unit cell is constructed and optimized in such a way so that it contains a centrally positioned single C$_{60}$ crystal at a high concentration of 10.8 wt% using the
“Amorphous Cell Packing” module (Materials Studio 2017, 2016). To realize the unit cell construction, an initial assumption should be made regarding its density and side length according to the following analytical considerations.

The mass of the fullerene \( m_{C_{60}} \) is as follows:

\[
m_{C_{60}} = 60m_c,
\]

where \( m_c = 1.9927 \times 10^{-23} \text{ g} \) is the mass of a carbon atom.
Given that the C\textsubscript{60} molecule has a vdW diameter of \(d_{C_{60}} = 10.35\ \textnormal{Å}\) (Krätschmer \textit{et al.}, 1990), its volume may be approximated by the following equation:

\[
V_{C_{60}} = \pi \left(\frac{d_{C_{60}}}{2}\right)^3 / 6.
\]  

The actual mass of the PA-12 material in the unit cell may be found by the fullerene mass \(m_{C_{60}}\) and the mass fraction \(M_{C_{60}} = 0.108\) by the following equation:

\[
m_{PA-12} = m_{C_{60}} \left(1 - M_{C_{60}}\right) / M_{C_{60}}.
\]  

Consequently, the volume of the polymeric matrix may be estimated by the following equation:

\[
V_{PA-12} = m_{PA-12} / \rho_{PA-12}.
\]  

Finally, the initial approximation of the density as well as the side length of the unit cell may be made, respectively, by the next relationships:

\[
\rho = \frac{m_{C_{60}} + m_{PA-12}}{V_{C_{60}} + V_{PA-12}},
\]  

\[
a = \sqrt[3]{V_{C_{60}} + V_{PA-12}}.
\]  

Using the MD predicted PA-12 density of \(\rho_{PA-12} = 0.936\ \textnormal{g/cm}^3\), Equations (4)–(9) lead to a unit cell density and length of \(\rho = 0.995\ \textnormal{g/cm}^3\) and \(a = 22.33\ \textnormal{Å}\), respectively. 

Next, a dynamic analysis under NPT ensemble for 20 and 40 ps under a pressure of 0.2 GPa and 1 atm, respectively, is performed to get the real density and size of the unit cell. The unit cell density changes during the 60 ps of the simulation may be seen in Figure 6. Note that the final unit cell density at 60 ps almost converges to the initial assumed density value at 0 ps which is calculated by Equation (8).

To prove the effectiveness of the analytical Equations (4)–(9), Figure 7 is presented which illustrates, among other data, the analytically predicted initial unit cell size in contrast with the final unit cell size after the 60 ps dynamic analysis.

\[\text{Figure 6.} \]  
\text{Materials' density variation with time during the MD analysis.}
Having achieved an equilibrated nanocomposite structure, a static analysis is performed and the elastic modulus and the Poisson’s ratio of the unit cell are found equal to 3.464 GPa and 0.3471 according to Equations (2) and (3), respectively.

3.3 FEM–MD combined analysis of PA-12 reinforced with low concentrations of \( \text{C}_{60} \)

Note that small amounts of fullerenes are commonly added into a polymeric matrix in order to prevent agglomerate phenomena. Moreover, it is proved that carbon-based nanofillers in the range of 3–5 per cent by weight promise substantial improvement of mechanical properties and may achieve the same reinforcement as 20–30 per cent of microsized fillers (Bhattacharya, 2016). Thus, rather small loadings of \( \text{C}_{60} \) reinforcement up to 4 per cent by weight are studied in this section. However, small concentrations of nanofillers inevitably lead to large representative volume elements (RVEs), which require substantial computational effort when atomistic methods such as MD are to be followed exclusively. On the other hand, the mechanical properties of the nanocomposites are significantly influenced by interfacial interactions between the nanofillers and the polymer matrix. Therefore, the accurate representation of interfacial effects is essential and may only be achieved via a numerical analysis at an atomistic level. Given the above, a multiscale approach is proposed here, which simulates the nanocomposite via continuum mechanics FEM by using the LUSAS finite element analysis system (LUSAS, 1999). The proposed analysis pre-requires the above described MD-only simulations of the pure PA-12 and the \( \text{C}_{60}/\text{PA-12} \) nanocomposite of a high mass fraction \( M_{\text{C}_{60}} \) (taken here equal to 0.108). Due to the assumed periodic dispersion of \( \text{C}_{60} \) molecules, the nanocomposites with small reinforcement concentrations may be analyzed according to the ABCDEFGH cubic RVE which is illustrated in Figure 8.

In this second stage of the analysis, the MD-based mechanical property data regarding the abcdedfg unit cell with \( M_{\text{C}_{60}} = 0.108 \) are utilized to develop a continuum same sized solid unit within the whole ABCDEFGH RVE of the nanocomposite with \( M_{\text{C}_{60}} = 0.01, 0.02, 0.03, 0.04 \). Note that Equation (9), which leads to Figure 7, is appropriate for the estimation of the size of the whole RVE as well. According to the followed micromechanical analysis and due to the symmetry, only the one-eighth of the problem domain may be formulated, which is denoted by LMNOKFIJ in Figures 8 and 9, given that appropriate symmetry conditions are adopted.
An orthogonal Cartesian coordinate system is used as reference with X, Y and Z axes aligned with the main dimensions of the one-eighth of the RVE. Both the central C$_{60}$/PA-12 solid unit as well as the outer PA-12 volume are discretized with isoparametric, hexahedral, linear, eight-noded finite elements having three degrees of freedom per node (displacements $u_X$, $u_Y$ and $u_Z$) (LUSAS, 1999) as Figure 9 illustrates. It should be noticed that Figure 9 is
representative for the discretization adopted for all the developed models. Specifically, each finite element mesh is achieved by using 4,000 solid elements denoted as HX8M.

Since the interfacial phenomena are already incorporated in the first stage atomistic analysis, the connection between the central unit cell volume and the outer PA-12 volume is obviously considered perfect. The elastic material properties which are computed via the MD-only simulations are adopted for the outer polymeric volume. In order to determine the elastic properties of the nanocomposite, appropriate boundary conditions must be applied. For the calculation of the elastic modulus and Poisson’s ratio, a small uniform strain \( \varepsilon_z = 0.01 \) is applied on the boundary LKFM. In addition, the constraints \( u_x = 0, u_y = 0 \) and \( u_z = 0 \) are applied on the boundaries OJKL, OLMN and OJIN, respectively, whereas the boundaries JKFI and NIFM are kept parallel to their original shape by nodal coupling since shear stresses on these boundaries must be zero due to the symmetry. The aforementioned simple boundary conditions are appropriate for the problem under consideration given the uniform dispersion and the spherical shape of the nanoparticles as well as the symmetry of the loading. The elastic modulus of the nanocomposite is computed via the ratio of average stress, obtained from the sum of reactions in the ground \( Z = 0 \), to the applied strain \( \varepsilon_z = 0.01 \). The Poisson’s ratio is computed through the ratio of the arisen average transverse strain, to the applied normal strain.

4. FEM–MD-based numerical results
Small filler loadings of 1, 2, 3 and 4 wt% are numerically tested. Figure 10(a) and (b) depicts the contours of the equivalent von Mises stresses and strains for the filler loading case of 1 per cent by weight. The stress concentrations within the reinforced unit cell become obvious, while strains tend to increase at the outer PA-12 volume and especially on the deformed boundary above the reinforcement.

The variation of the nanocomposite elastic modulus vs the \( C_{60} \) weight percentage is given in Figure 11(a). The numerical outcome shows that the addition of \( C_{60} \) nanoparticles within PA-12 by 4 per cent by weight may lead to a stiffness improvement up to 13 per cent. Some corresponding experimental results are as well illustrated in Figure 11(a). Very good agreement between the numerical results and those measured via appropriate experimental procedures (Zuev, 2011) may be observed for the filler loading cases up to 3 per cent. The dependence of the nanocomposite Poisson’s ratio on the percentage mass fraction of the \( C_{60} \) reinforcement is illustrated in Figure 11(b).

Generally, the stiffness of the nanocomposite increases linearly as the percentage mass fraction of the nanofiller increases. On the other hand, the Poisson’s ratio of the nanocomposite is reduced as the filler mass fraction gets higher values due to the small Poisson’ ratio of \( C_{60} \). It has to be noted here that the use of denser meshes than that depicted in Figure 9 has a negligible effect on the FEM numerical solutions.

5. Conclusions
A three-dimensional, multiscale micromechanical numerical method has been formulated in order to predict the elastic mechanical behaviour of PA-12 polymer which is uniformly reinforced with \( C_{60} \) nanoparticles at small loading levels.

The complicated interfacial phenomena between the nanofiller and the polymeric matrix require analysis at an atomistic level, while small nano-reinforcement mass fractions demand continuum numerical treatment to reduce the enormous computational cost.

Therefore, a two-stage analysis is proposed according which MD simulations are first performed to extract the mechanical behaviour of pure PA-12 as well as a \( C_{60}/PA-12 \) unit cell of a rather high filler mass fraction. Second, an appropriate volume element, which contains a same sized unit cell and, thus, represents the whole nanocomposite, is simulated in a continuum manner via FEM. Elastic material parameters such as the
elastic modulus and the Poisson’s ratio have been computed for several loading levels up to 4 per cent by weight. The results have shown that the increase of C_{60} mass fraction leads to a significant increase of the nanocomposite stiffness but also to a decrease of its Poisson’s ratio.

Figure 10.
Contours of the equivalent von Mises strain

Notes: (a) Stress; (b) strain of the RVE describing the C_{60} reinforced PA-12
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Further reading


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Contact modeling with a finite element model in piston ring–liner conjunction under dry conditions

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Abstract

Purpose – The purpose of this paper is to focus on the creation of an isothermal elastic ring-liner model to highlight, through stresses, the occurrence of the plastic deformation in certain crank angles under extreme dry lubrication conditions.

Design/methodology/approach – The stresses that are exported from this analysis are pointing out not only the necessity for an elastoplastic model to be created, but also the importance of predicting the correct friction coefficient, as pointed out by both the contact surface stress and that in depth of the two bodies in contact.

Findings – The comparison between the finite element model and the adhesion mathematical model of Johnson, Kendall and Roberts seals the importance to calculate the interaction forces, acting on the common solid surface, in the pursuit of defining a proper contact patch. Additionally, a three-dimensional ring model is built, highlighting the importance of the modeling surface’s micro asperities for a solid stress analysis. Also, numerical experiments are conducted, in contact with the cylinder and a piston ring made of an iron alloy and of two different plating materials, such as Chromium (Cr) and Chromium–Nickel Alloy (CrN). The ability to calculate the stress concentration factor is also described.

Originality/value – A three-dimensional ring model is built, highlighting the importance of the modeling surface’s micro asperities for a solid stress analysis. Also, numerical experiments are conducted, in contact with the cylinder and a piston ring made of an iron alloy and of two different plating materials, such as Chromium (Cr) and Chromium–Nickel Alloy (CrN). The ability to calculate the stress concentration factor is also described.

Keywords FEM, Piston ring, Contact modeling, Stress analysis, Dry lubrication conditions

Paper type Research paper

Nomenclature

\( z \) force per unit area Lennard–Jones \( K_0 \) the initial contact stiffness

\( \gamma \) work of adhesion \( [K] \) Stiffens matrix

\( \alpha \) Hertz contact area \( u \) displacements

\( \delta \) Deflection due to Hertzian contact \( \varepsilon_{ij} \) deformations

\( \rho \) Hertz contact pressure \( \sigma_i \) stresses

\( E \) Modulus of elasticity \( F_T \) ring tension force

\( E' \) combined modulus of elasticity \( \rho_{el} \) elastic pressure

\( \nu \) Poisson’s ratio \( F_G \) back gas force

\( \delta \) virtual work \( U \) sliding velocity

1. Introduction

The pioneering work by Holmberg et al. (Holmberg and Erdemir, 2017) showed that the highest energy losses in engines and transmissions appear due to boundary friction. Among these, the ring-pack assembly is the primary source of friction (Zavos and Nikolakopoulos, 2018). This contact pair, of the piston ring pack and the cylinder liner, is also one of the most
complex to analyze because of the transient nature of this conjunction and the variation of the regimes of lubrication. The basic theory of contact between elastic bodies has been of great importance for many years. Initially, variety of models of Johnson (1987) and Sackfield, Hills and Nowell (2013) approached consistently the mechanical problem of contact, whereas Gladwell (1980) and Galin (1953), along with Kikuchi and Oden (1988) and Khudnev and Sokolowski, (2012) reported the same problem from both a mathematic and a finite element perspective, respectively. Additionally, Bhushan conducted research for both a single (Hertzian) (Bhushan, 1996) and multiple asperity (Bhushan, 1998) approach, laying the foundations of the so-called contact mechanics of rough surfaces. Schwarz (2003) was also focused on calculating the elastic deformation of a sphere on a flat surface with JKR and DMT models, whereas Adams and Nosonovskiy (2000) categorized the theoretical contact models according to adhesion, friction and single or multiple asperity. Shi and Zhao (2004) compared the adhesion JKR and DMT models with the adhesionless Hertzian model, mentioning the influence of dimensionless load parameter. A complete research in the field of surface energy, which is the factor that distinguishes the JKR from the Hertzian model, was conducted by Leite et al. (2012). They provided an extended list of measured materials in terms of intermolecular interactions and surface forces along with a complete theoretical approach.

Even a highly polished surface with today’s manufacturing methods has surface roughness on many different scales. The contact regions can be visualized as small areas where asperities from one solid are compressive pressured against asperities of the other solid part; depending on the conditions, the asperities may deform elastically or plastically.

The size of the real contact area between a solid bodies is very important, especially for practical implications. For example, it determines the contact resistivity and the heat transfer between the solids. It is also of direct importance for sliding friction, that is the sliding friction between a piston ring and the cylinder surface, or of a tire and the road, and it has a major influence on the adhesive force between two solids blocks in direct contact, evaluating the elastic or the plastic contact (Persson et al., 2002).

Our approach is coming to an end for the creation of an isothermal elastic model to highlight stresses distribution, in certain crank angles, under extreme dry conditions. The stresses that are obtained from the current work are pointing out the necessity for an elastoplastic model to be developed, in order to predict more accurately the friction coefficient. Furthermore, the comparison between the finite element model and the mathematical model of Johnson, Kendall & Roberts seals the importance to calculate the interaction forces, acting on the common solid surface, in the pursuit of defining appropriate contact patch. Numerical techniques and limitations are analytically presented and discussed. Furthermore, a three-dimensional model is proposed for further stress field investigation and for highlighting the importance of modeling surface’s micro asperities in several distributions for a solid stress analysis. The friction coefficient in a contact state between coated ring and cylinder is calculated, and the ability for the stress concentration factor estimation in rough surface in contact is also discussed.

The reminder of this paper is organized as follows: in Section 2, the theoretical considerations are presented, along with the used methods for the solution with its limitations. In Section 3, the numerical simulation details concerning the two-dimensional analysis are given, whereas in Section 4, the detail results are presented and discussed. The three-dimensional model for stresses evaluation is also presented in this section. Finally, the major conclusions obtained from the analysis are presented in Section 5.

### 2. Theory

#### 2.1 Surface energy–work of adhesion

The difference between the two compared mathematical models (Hertzian and JKR) is on the interaction forces that are considered to act in the contact surface of the compression ring.
and the cylinder liner. This case, in which the contact between two surfaces is elastic, results in no energy dissipation during the interaction, and so both the adhesive and the surface forces are equal in magnitude (Schwarz, 2003). Schwarz mentioned that the area between the force curve and the base line, right from the equilibrium distance \( z_0 \), is considered as the work of adhesion or Dupré's energy:

\[
\gamma = \int_{z_0}^{\infty} \sigma(z)dz,
\]

where \( \sigma(z) \) is the force per unit area; Lennard–Jones potential typically represents the adhesive stress \( z \) (Schwarz, 2003):

\[
\sigma(z) = -\frac{8\gamma}{3z_0} \left[ \left( \frac{z}{z_0} \right)^{-3} - \left( \frac{z}{z_0} \right)^{-9} \right],
\]

where \( z \) is the separation between atomic planes.

Another method consists of calculating the work of adhesion and then relating it to the Hamaker constant through the work of Adams and Nosonovsky, (2000). Separation will occur only when contact area \( a_{\text{JKR}} \) becomes equal to 0.63\( a_{o} \), where \( a_{o} \) is the contact area in the absence of external load. If two phases \( (i, j) \) in contact are pulled apart inside a third phase \( k \), then the work of adhesion according to Adams and Nosonovsky, (2000) is given by the expression the following equation:

\[
\gamma = \gamma_{ik} + \gamma_{jk} - \gamma_{ij}.
\]

Another method consists of calculating the work of adhesion and then relating it to the Hamaker constant through the work of Leite et al. (2012). Separation will occur only when contact area \( a_{\text{JKR}} \) becomes equal to 0.63\( a_{o} \), where \( a_{o} \) is the contact area in the absence of external load.

As an attempt to build an adhesive contact model, the work of adhesion parameter used to model JKR was obtained from Chong et al. (Chong, 2012).

### 2.2 Adhesionless single asperity contact – Hertz model

The two bodies, the compression ring and the cylinder liner, can be treated as a two-dimensional cylindrical contact problem. Boundary friction condition is assumed, as in top and bottom dead center of a four-stroke cycle lubricant is usually absent. (Stachowiak and Batchelor, 1994) formulated the basic contact parameters between two parallel cylinders. It is assumed that the dimensions of the contact area are small compared to the dimensions of each body and to the radii of curvature of the surfaces, and that the strains are sufficiently small for linear elasticity to be valid. Neglecting any attractive forces \( (\gamma = 0) \) and considering only the external load acting in the back of the ring, the contact radius, maximum deflection and maximum contact pressure, which occurs at \( r = 0 \) (center of contact), are predicted from the Hertzian model as follows (Hertz, 1896):

\[
a = \left( \frac{4PR}{\pi LE^*} \right)^{1/2},
\]

\[
\delta = 0.319 \left( \frac{P}{E^*L} \right) \times \left[ \frac{2}{3} + \ln \left( \frac{4R_1R_2}{a^2} \right) \right],
\]
\[ p_{0}^{H} = \frac{P}{\pi aL}, \]  \hspace{1cm} (6)

\[ \tau_{\text{max}} = 0,304 p_{0}^{H}, \]  \hspace{1cm} (7)

at depth of:

\[ z = 0,786a, \]  \hspace{1cm} (8)

where:

\[ \frac{1}{E^{*}} = \frac{1-v_{1}^{2}}{E_{1}} + \frac{1-v_{2}^{2}}{E_{2}}, \]  \hspace{1cm} (9)

and:

\[ \frac{1}{R} = \frac{1}{R_{1}} + \frac{1}{R_{2}}, \]  \hspace{1cm} (10)

with \( E_{1}, E_{2}, \nu_{1}, \nu_{2} \) being the Young’s modulus and Poisson’s ratio for the compression ring and the cylinder liner, respectively, \( \delta \) is indentation depth and \( \alpha \) is the radius of contact area. \( E^{*} \) is the reduced Young’s modulus and \( R \) the effective radius of the two-cylinder bodies, with \( R_{2} = \infty \) for the infinite cylinder radius (the cylinder is considered as plane). The pressure distribution over the contact area, considering equally distributed external load, is calculated by the following equation:

\[ p^{H}(r) = p_{0}^{H} \left( 1 - \frac{r^{2}}{a^{2}} \right)^{1/2}, \]  \hspace{1cm} (11)

where \( r^{2} = x^{2} + y^{2} \), although considering two-dimensional model, \( r = x \).

2.3 Adhesion single asperity contact – JKR model

Extending the Hertzian theory and setting \( \gamma \neq 0 \), the contact equations are modified including the effect of the work of adhesion. The effective external load applied according to the work of Johnson et al. (1971) is given in the following equation:

\[ p^{\text{JKR}} = P + 3\pi R_{\gamma} \pm \sqrt{6\pi R_{\gamma} P + (3\pi R_{\gamma})^{2}}, \]  \hspace{1cm} (12)

which is larger than the external force used in Hertzian approximation. The minus symbol in the above expression denotes unstable conditions. The contact radius, the maximum deflection and the maximum contact pressure, which are dependent on the external applied load, are given by Equations (13)–(15):

\[ x^{\text{JKR}} = \left( \frac{R p^{\text{JKR}}}{K} \right)^{1/3}, \]  \hspace{1cm} (13)

\[ \delta^{\text{JKR}} = \frac{a^{2}}{R} \sqrt{\frac{8\pi a}{3K}}, \]  \hspace{1cm} (14)
\[ p_0^{JR} = \left( \frac{3\gamma K}{2\pi a} \right)^{1/2}, \]  

where:

\[ K = \frac{4E^*}{3}. \]

The pressure distribution over the cyclic contact area, which is compared with the Hertzian approach, including an equally distributed external load is expressed as:

\[ p^{JR}(r) = p^H(r) - p_0^{JR} \left( 1 - \left( \frac{r}{a} \right)^2 \right)^{-1/2}. \]  

2.4 Penalty and Lagrange methods – solvers of the contact problems

The penalty, the Lagrange and the augmented Lagrangian methods are widely used for solving contact problems (Yastrebov, 2011). Compared to the other methods, the Lagrange multiplier method was chosen here to solve the present ring–cylinder bore contact problem. The basic constrained optimization problem can be formulated using the principle of the minimum total potential energy. This principle is a fundamental concept used in structure analysis calculating structure deformation. It states that a structure should deform to a stationary state that minimizes its total potential energy, including the elastic strain energy and the potential energy from the applied force.

This principle is used to formulate the ring–cylinder bore contact problem. In this work, according to several numerical tests done, it was concluded that the augmented Lagrange method is not suitable for the solution of the certain contact model, since the solution of the finite elements does not satisfy the Gauss points. The same conclusion was arisen using the internal multipoint constrain (MPC) method. So, the Lagrange multiplier method combined with the penalty method is used in order to obtain solution (Hirmand et al., 2015). This means that convergence was achieved by also regulating the penetrations between the cylinder’s elements and ring (slave and body) (Figure 1).

Penalty means that any violation of the contact condition will be punished by increasing the total virtual work:

\[ \delta\Psi = \int_V \sigma^T \delta\epsilon dV + \int_f (\epsilon_N \delta g_N + \epsilon_T \delta g_T) dA. \]  

**Figure 1.** The contact problem simulated in finite elements

Source: Drive Ansys (2015)
Any violation of the contact condition will be then manipulated with Lagrange multiplier:

\[
\delta \Psi = \int_V \sigma^T \delta \varepsilon \, dV + \int_{\Gamma} (\lambda_N \delta g_N + \lambda_T \delta g_T) \, dA. \tag{19}
\]

Contact constraint conditions are defined as:

\[
g_N \geq 0, \tag{20}
\]

\[
\lambda_N \leq 0, \tag{21}
\]

\[
g_N \lambda_N = 0. \tag{22}
\]

In case of linear elastic and node-to-node contact, the equation is linear. In certain cases wherein non-linearity appears, an iterative method, usually the Newton–Raphson, is used to solve inside the contact area, by using the following equations:

\[
[K][u] = \{f\}, \tag{23}
\]

\[
\{u\} = [K]^{-1}\{f\}, \tag{24}
\]

where \([K]\) is the stiffness matrix containing \(n \times n\) terms, and \([u], \{f\}\) are \(n \times 1\) displacements and force matrices. Further, the boundary conditions vary in the contact surface, resulting in a successful iteration approach to achieve convergence. To this end, the forces and the stiffness of the system are functions of the node displacements into the contact surface:

\[
[K(u)][u] = \{f(u)\}. \tag{25}
\]

The general principle behind Newton–Raphson iterative method is based on the principle that it uses, starting with an initial value, the previous values of displacement \(u\), which defines the matrix \(K(u)\), according to the following equation, converging to the nearest solution (Figure 2):

\[
\{u^{+1}\} = [K(u)^{+1}\{f\}. \tag{26}
\]

The isotropic materials’ plane loads and the generalized Hooke’s Law are:

\[
\varepsilon_{ij} \equiv \frac{1}{2} \left( \frac{\Delta u_i}{\Delta x_j} + \frac{\Delta u_j}{\Delta x_i} \right) \forall (i, j) = \{1, 2, 3\}. \tag{27}
\]

Stress and strain calculation on finite elements is conducted as a first-order approximation:

\[
\sigma_i = C_{ij} \varepsilon_j \forall (i, j) = \{1, 2, \ldots, 6\}. \tag{28}
\]

2.5 Approach of the contact solution

A CFD analysis, similar to (Shahmohamadi et al., 2013), needs to be conducted, first, to obtain the critical crank angles where the thickness of the lubricant is insufficient. The above analysis is carried out for four critical angles. To include the worst of the scenarios among those angles, the absence of lubricant is decided. Therefore a two-dimensional single asperity model is created. The correctness of the model is attested by Hertz (1896), in static load conditions. Following the application of a force in the circumferential of the piston ring,
a second lateral force is applied, to simulate the sliding phenomenon that occurs inside the combustion cylinder. Finally, a three-dimensional model is created, to magnify inside the contact area and examine the stresses envelope for a variety of operating and surface conditions. The flow chart on how the problem is approximated is shown in Figure 3.

3. Contact modeling of ring–liner conjunction
3.1 Ring–liner lubrication problem
Figure 4 shows the typical piston ring–liner lubrication problem. The area of interest is the compression ring near to the top dead center.
The cylinder and the ring are considered very long in the vertical of sliding direction, a state that leads safely to a 2D model in-plane stress condition. The sliding bodies have a perfect elastoplastic behavior. In some of the cases analyzed, the friction coefficient is considered to be zero in contact, avoiding thus the plasticity. The sliding is modeled as pseudo static process. The temperature influenced during the sliding process is not taken into account. This point is for further research. All the mechanical properties are defined for environmentally temperature conditions. The cylinder is modeled as a stationary, whereas the ring as a moving wall.

Radial forces act in the cylinder–piston ring system, whereas the axial forces are considered negligible. According to Haddad and Tjan (1995), the problem could be reduced to 1D problem.

With regards to the in-plane ring behavior, two outward forces are applied between the piston and the back profile of the ring. The ring tension force is obtained as:

$$F_T = 2\pi r_o b p_{el},$$  \hspace{1cm}\text{(29)}$$

where the elastic pressure is taken as $p_{el} = \frac{d_{gap} E_{ring} I_{ring}}{3\pi b (r_o)^2}$, whereas the ring cross section is defined as $I_{ring} = \frac{b w^2}{12}$ Additionally, the back gas force is given as:

$$F_G = 2\pi r_o b p_{bk}(\phi).$$  \hspace{1cm}\text{(30)}$$

Supposing zero sliding velocity of the compression ring in the radial direction and also relative zero velocity in the piston groove, the relative sliding velocity according to Shahmohamadi et al. (2013) can be expressed by the following equation (Figure 5):

$$U(\phi) = -r \omega \sin \phi \left\{ 1 + \cos \phi \left[ \left( \frac{l}{r} \right)^2 - \left( \frac{l}{r} \right)^2 \right] (-1/2) \right\},$$  \hspace{1cm}\text{(31)}$$

where $\omega$ is the angular velocity, $\phi$ is the crank angle, $l$ the length of the piston rod and $r$ is the radius of the crank pin.

**Figure 4.**
Typical piston ring–liner contact problem
3.2 Two-dimensional elastic ring–liner model

The contact performance of the top compression ring–liner conjunction for a high performance four-stroke engine at dead centers has been studied through numerical analysis. The material of the ring is Steel SAE 925, and its mechanical properties are received from Johnson et al. (1971). Both the cylinder and the compression ring are deformable solids. Table I presents the basic dimensions. The forces acting on the back of the ring in radial plane are the ring’s elastic tension force and the gas force, both obtained from the work of Shahmohamadi et al. (2013). The back gas pressure is also presented for this investigation in Table II.

A total of 33,810 quadrilateral elements are used and a penalty method algorithm solves the contact problem in ANSYS Multiphysics package. The analyses are conducted for non-linear geometrical behavior of the finite element whose stiffness matrix is given by the non-linear Equation (18):

\[ K(u) = K_0 \left( 1 - \frac{u}{U} \right) = \text{stiffness matrix}, \]

\[ (32) \]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Compression ring</th>
<th>Cylinder liner</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial face width (m)</td>
<td>$1.15 \times 10^{-3}$</td>
<td>$1.15 \times 10^{-3}$</td>
</tr>
<tr>
<td>Radial face width (m)</td>
<td>$0.085 \times 10^{-3}$</td>
<td>$0.085 \times 10^{-3}$</td>
</tr>
<tr>
<td>Crown height (m)</td>
<td>$10 \times 10^{-6}$</td>
<td>–</td>
</tr>
</tbody>
</table>

Table I. Ring–liner tribo-pair dimensions

<table>
<thead>
<tr>
<th>Degrees</th>
<th>Operation time</th>
<th>Back-ring pressure (Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$-180$</td>
<td>Starting compression</td>
<td>134,043,806</td>
</tr>
<tr>
<td>$0$</td>
<td>Starting combustion</td>
<td>2,322,582.23</td>
</tr>
<tr>
<td>$180$</td>
<td>Starting exhaust</td>
<td>138,727,801</td>
</tr>
<tr>
<td>$360$</td>
<td>Starting inlet</td>
<td>142,498,386</td>
</tr>
</tbody>
</table>

Table II. Pressure data in the back-ring profile

Note: This figure is originally obtained by the paper algorithm

Figure 5. Sliding ring velocity
where $K_0$ is the initial contact stiffness ($N/m$), $U$ is the sliding velocity ($m/s$) and $u$ is the nodal momentarily velocity ($m/s$). The initial stiffness is suggested to take values in an order of magnitude larger than the elastic modulus of the underlying geometry. All in all, this value must be a balanced value between the accuracy of the results and the run time. A comparison is conducted for two different coefficients of friction, $\mu = 0$ and $\mu = 1$, in order to examine the equivalent normal stresses along the depth of the compression ring. The model is solved iteratively, using the Newton–Raphson method. The vertical stress distribution along piston ring’s depth for the friction, the frictionless and analytical solution is plotted in Figure 6.

Comparing the case of analytical (line contact in two cylinders with parallel axis) with the frictional numerical solution, a lack of convergence is spotted. Both analytical and frictionless cases do match with the friction, one in the boundary lubrication of the contact surface and in the back surface of the ring, respectively. As we approach the highest contact point between the two contact surfaces ($Y$-cord $= 10^{-4}$), a significant deflection in normal mechanical stress between the two cases is observed. In the frictionless case (sliding surfaces), thermal load is neglected; therefore, stress is fully translated into mechanical (shear), whereas in the case of $\mu = 1$ (sticking), a great amount of energy is converted into thermal stresses (which are not presented in the current work), justifying the lower mechanical stress value and denoting the pure contact in dead centers of an internal combustion engine.

3.3 Boundary conditions

Figure 7 shows the simulation model for ring–liner contact. The first step of every finite element model is to validate an existing theoretical model. Therefore, an analysis should be conducted in the absence of any axial forces, considering only the radial ones acting on the ring. The number of elements used is 38,410 and the number of nodes is 38,577. The convergence is achieved using CONTACT172 and TARGE169 type of elements. The number of nodes that are in sliding denotes the contact surfaces. In Figure 7(b), the grid of nodes for one of the solutions obtained is indicated. It is obvious that the nodes that are in sliding condition, and therefore in pure contact, define, thus, the contact surface. Dividing the stresses in $x$-direction over the stresses in $y$-direction, the kinematic friction coefficient in the certain stresses angle can be calculated. These values are used later on as inputs in the contact model.
4. Results and discussion

4.1 Validation – comparing Hertzian and JKR theoretical model with the finite element model

After examining the dead center angles of every stroke (−180°, 0°, 180°, 360° and 540°), a comparison was conducted for 0°, which turned out to be the most critical among them (Shahmohamadi et al., 2013).

Notes: (a) Boundary conditions and magnified contact mesh detail, (b) node state during sliding contact

Figure 7. (a) Input boundary conditions and (b) meshing details
In Figure 8, the contact pressure distribution is presented over the normalized contact area for the Hertzian, JKR and FE models. The deviation of models is approximately 10 percent, on the one hand, owing to the absence of surface energy in case of the Hertzian solution and, on the other hand, owing to dependence of the FEA model from the number of its elements. Furthermore, the equations formulating both Hertzian and JKR theory refer to cylindrical geometry, which slightly differs from our model’s case. However, the maximum value of the contact pressure has a good agreement for three models. In order to achieve the most accurate results, grid sensitivity tests are performed.

The analysis of the work of Zavos and Nikolakopoulos (2018) is used here in order to analyze the system in a mixed lubrication regime. This analysis indicates the critical point through the minimum film thickness that there is possibility for roughness contact.

4.2 Contact stress and frictional analysis of several coating materials

The finite element method is used in order to model the problem and to verify it with Hertz’s fundamental contact theory. In total, three numerical experiments are conducted to contact a cylinder with a piston ring made of an iron alloy spring as well as of two plating materials, such as Chromium (Cr) and Chromium–Nickel Alloy (CrN). The mechanical properties of the compression ring and the cylinder materials are studied, and they are summarized in Table III (Zavos and Nikolakopoulos, 2018; Shahmohamadi et al., 2013).

After extensive tests, the final model geometry is divided into 38,410 elements and 38,576 nodes. The contact between the cylinder and the ring is a rigid surface-to-surface contact. CONTA172 and TARGET169 elements are used to model the contact.

Further, two types of analysis are performed, and the results are also compared.

First, the large-deflection effects are deactivated with the suitable command (NLGEOM, OFF), and in the second analysis, the nonlinear geometry and the structural deflections are taken into account.

![Figure 8. Comparison between the Hertzian theory, the JKR theory and the FEA](image)

<table>
<thead>
<tr>
<th>Material</th>
<th>Steel SAE 9254</th>
<th>Chromium plated (Cr)</th>
<th>Chromium Nitride (CrN)</th>
<th>Grey cast iron</th>
</tr>
</thead>
<tbody>
<tr>
<td>Modulus of elasticity (GPa)</td>
<td>203</td>
<td>276</td>
<td>400</td>
<td>92.3</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.211</td>
<td>0.21</td>
<td>0.2</td>
<td>0.3</td>
</tr>
</tbody>
</table>
In Table IV, the solution from the current work using FEM is compared (as a validation) with the analytical solution to investigate the effect of the structure deformations in several crank angles and to show the possible errors between the two ways.

In Figure 9, the coefficient of friction (COF) for five crank angles is presented. The data are obtained in both cases, deactivating (geometrical linearity) and activating (geometrical non-linearity) the structural deformations.

It is observed that the COF values converge to the majority of the input values, with the exception of the zero degrees for the non-plating ring and the angle of 540° for chromium-plated ring.
The case of the non-plated ring approaches the unit value concerning the COF. On the contrary, the ring plated with chromium–nickel alloy exhibits the smallest COF of the three examined cases, which is justified, as it also has the greatest modulus of elasticity. Taking into account the isothermal conditions of the model, no comprehensive conclusion can be drawn, in terms of how the above three coating materials react to such extreme loading conditions.

However, a worth noting change in the friction coefficient value, the isothermal approach, confirms the undefined nature of the dry friction problem. Practically, such a change in friction coefficient in extreme engine operating speed conditions, where the accumulated roughness peaks on the surfaces of the ring and cylinder will come into contact, may cause the spinning effect of the ring. This phenomenon in its turn is undesirable for the manufacturers, which indicates the possible contact between the ring and the piston groove.

The value of the current work, in the presence of vertical loads only, has a two-fold meaning. On the one hand, it is possible to understand at an early stage the process of the contact and how it can be solved by a computer code, as well as to define the correct parameters and the problem indicators. On the other hand, we can assume that the actual contact surface (or contact line for a two-dimensional model) can be defined as the periphery (or line for the two-dimensional model) of the nodes that are in sliding mode.

4.3 Two-dimensional stress analysis – sliding process

In this section, the ring cylinder sliding will be studied in dry friction conditions, concerning the critical area of the ignition initiation. The study covers the range 0°–0.1° space in crank shaft rotating. The choice of the above space is small enough to capture as much realistically the phenomenon of dry friction on the cylinder surface. In the simulation procedure, the role of friction as well as the non-linearities of the geometry is taken into account.

On the microscale level, the occurrence of plastic deformation of the surface roughness is well known. Thus, it is important to know what percentage of the roughness plastic deformation affects the functionality of the structure and its impact on the overall COF variation on the wear and the deformation of the surface.

Over the last four decades, problems of elastic and elastoplastic spherical contact have been analyzed by numerous researchers who relied on the roughness behavior, which they extended later on using statistical patterns of gravel distribution on the surface. However, everyone shared the following assumptions (Thomas and Thomas, 1999; Greenwood and Williamson, 1966): replacing the two rough surfaces with one, sharing the average of the properties of both surfaces and the other being considered rigid, simulating the random geometries of the asperities with simple basic shapes, triangles or arcs and predicting the type of roughness distribution.

Furthermore, Vijaywargiya (2006) found some of the limitations coming from the previous approaches, concerning the stress distribution, the actual forces that are developed as the sum of the loads of the roughness peaks in the contact area, and the energy losses due to the plasticity of an elastoplastic pair. Thompson, (2007) proceeded further and focused on the deviation due to a analytical prediction of the roughness characteristics, which are introduced as variables in the any simulation model, with those in which the statistical distribution of the roughness is directly introduced in a finite element model.

The analysis here will be performed for a given displacement corresponding to the range of angles, as the roughness of the two surfaces, of the compression ring and the cylinder, is in contact. In the previous section, the back force of the ring was calculated consisting of the combustion load and the pre-tension of the ring material. It has been found that the values of
the total pressure $P_{\text{gap}}$ are changed, so as not to significantly influence the contact force resulting from the solution.

During the first cycle of the solution, a table is defined of pressure values $P_{\text{gap}}$, which are set at any defined interval of 0.01° for 10 repetitions. During the second solution cycle, the load remains the same for 0.1° interval. This results in a quicker solution and smaller deviation in the results.

The use of the three different materials (steel, Cr and CrN), and the crank angle to correspond to combustion phase, which is the worst-case scenario (0°), and for 0.2 mm ring thickness, the below results were obtained (Figures 10–12).

From the above diagrams, it can be observed that the harder the material, like Cr and CrN, the bigger are the stresses, with CrN being the material with higher stresses in the contact points. This leads to a conclusion that the model is reliable for predicting stresses in the contact points.

In order to see the influence of the friction and frictionless considerations in the contact stresses and the effects of the geometrical linearity and non-linearity, Figures 13 and 14 are presented; the figures are presented for the harder material (CrN).

In both cases of Figures 13 and 14, there is a shift of the curve in the direction of movement in the case of frictional contact, as well as a noticeable stress development at the end of the stress curve due to the stick–slip phenomenon due to accumulated roughness peaks at this point.

4.4 The effect of Gaussian-distributed ring surface roughness on contact stresses

A numerical three-dimensional model with Gaussian-distributed surface roughness is created, simulating a cylinder liner and piston ring contact pair in a small area of their contact patch on microscale (Figure 4). The model is created using a 3D sample, $9 \times 9 \times 3$ mm with roughness distribution according to Gauss statistical distribution.

![Figure 10. Equivalent von Mises stresses for Cr, in contact with friction and geometrical non-linearity (ON)](image)
Figure 11. Equivalent Von Mises stresses for CrN, in contact with friction and geometrical non-linearity (ON). Slide_Contact

Figure 12. Equivalent von Mises stresses for unplated steel SAE 9254, in contact with friction and geometrical non-linearity (ON). Slide_Contact
Figure 15 shows that as the asperity's density becomes higher, the maximum equivalent stress also becomes higher. Furthermore, an obvious reduction in maximum pressure is observed disproportionately to the asperity peaks rise, in which more surface (peaks) share an equal amount of load. Among them, the necessity of a golden section is pointed out to be found between the topographical
parameters compared. The Gaussian roughness distribution (or the asperities density) is developed using one, two and five roughness peaks per mm. The height of the roughness varies as $0.02 \times 10^{-4}$, $0.2 \times 10^{-4}$ and $10^{-4}$ mm. The model simulated surface-to-surface contact using the elements TARGE170 and CONTA174. Ring’s material was Steel SAE 9254 and the crank velocity was set to 6,000 rpm.

For the current analysis, after grid sensitivity test, various elements and nodes are used: 3,971 elements and 3,140 nodes, 15,230 elements and 12,104 nodes, 145,800 elements and 166,464 nodes. The cylinder and ring contact is considered as a surface-to-surface contact, with the two bodies retaining the mechanical properties of their materials, and therefore they are being considered flexible.

The contact pair elements are CONTA174 and TARGE170 (Figure 16).

In Figure 17, the average contact pressure is illustrated as a function of the roughness height and roughness density.

Several conclusions can be drawn from the above diagram. Regarding the roughness height, 0.002 mm, which is the smallest height of the three defined, there is a satisfactory convergence for all of the three proposed grids. The convergence is not maintained for height roughness of 0.1 mm. This resulted due to the coarser surface. It also indicates that minimal roughnesses that stand for this load are significantly more numerous than the other two cases in the case of 5,000 peaks per unit length.

In case of the roughness surface of 0.002 mm height, it is obvious that the peaks that receive the external load are sufficiently close to the other cases, resulting in a more uniform load distribution. The intermediate value of 0.02 mm comes to verify the almost linear relationship that governs all the numerical measurements from the roughness heights.
Notes: (a) One peak per mm; (b) two peaks per mm; (c) five peaks per mm for an asperity mean height of $0.2 \times 10^{-4}$ mm (material Steel SAE 9254)

Figure 16. 3D contact modeling total stresses on compression ring surface for three different asperity dense contact surfaces

Figure 17. Comparative diagram of mean contact stress for the three different roughness cases
As a general conclusion, it is worth adding the linear error resulting from the calculations, which becomes bigger as the sum of the finite elements on the contact surface increases.

4.5 Roughness distribution – stresses and stress intensity factors

The surface quality of the mechanical structures as parameter of their integrity is an important parameter to assess the material behavior under variable for static loads. This is related to a series of processes occurring in the superficial layer after mechanical processing and interaction with the environment (Thompson, 2007).

In this work, as described in the previous sections, the sliding process is simulated once the problem is considered as a static problem. However, the roughness peaks are singular points in a surface, and the point in the bottom of the roughness hills can be considered as notch points. To this end, and specially with the 3D sliding simulation, the stress can be calculated in the bottom and in the peak of the roughness height, resulting in the calculation of the stress intensity factor. The stress concentration factor is very significant parameter, either in static analysis or, and more specifically, to assess the evolution of the fatigue process.

Having defined (Arola and Williams, 2002) the roughness average, $R_a$, peak–peak height, $R_p$, the ten point height, $R_z$, and the maximum valley depth, $R_v$, the models can be used instead of the linear parameters used in the original models.

These relationships will be obtained as:

$$ K_t = 1 + n \left( \frac{R_a}{\rho} \right) \left( \frac{R_y}{R_v} \right), $$

where $n$ is empiric constant for uniform tension $n = 2$; and for shear $n = 1$ and $\rho$ is the notch root radius.

Of course, alternatively, the stress concentration factor can be calculated dividing the point maximum stress over the initial nominal stress.

For example, Table V, refers to the Figure 17. The stress concerning the roughness height $10^{-9}$ and 1,000 peaks per unit length can be considered as the nominal stress. So, in any examine case, the stress concentration factor can be estimated. This leads to a surface design (regulating the machining parameters to achieve the surface roughness), controlled in sliding friction fatigue conditions.

5. Conclusion – future research

The main conclusions can be summarized here:

- The maximum value of the contact pressure has a good agreement for three models, which declares the accuracy of an adhesionless (Hertz) and an adhesion (JKR) models (Figure 3).

<table>
<thead>
<tr>
<th>Peaks’ Roughness picks per unit length</th>
</tr>
</thead>
<tbody>
<tr>
<td>1,000</td>
</tr>
<tr>
<td>Average contact pressure/stress concentration factor</td>
</tr>
<tr>
<td>--------------------------------------------------</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Height of roughness ($10^{-9}$)</th>
<th>0.02</th>
<th>0.2</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_t = 1$</td>
<td>1.7E + 08</td>
<td>1.88E + 09</td>
<td>9.19E + 09</td>
</tr>
<tr>
<td>$K_t = 2.76$</td>
<td>6.40E + 07</td>
<td>6.36E + 08</td>
<td>3.66E + 09</td>
</tr>
<tr>
<td>$K_t = 7.5$ (unacceptable value)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$K_t = 2.93$</td>
<td>4.79E + 08</td>
<td>4.87E + 09</td>
<td>2.17E + 10</td>
</tr>
<tr>
<td>$K_t = 7.6$ (unacceptable value)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$K_t = 2.51$</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$K_t = 5.92$ (unacceptable value)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table V. Roughness with average contact stress and stress concentration factor
In the plated tribo-pairs (Figure 9), the harder the material, the bigger are the stresses, with CrN being the material with the higher stresses in the contact points.

Regarding the three-dimensional model, a remarkable reduction in maximum pressure is observed due to the asperity peaks rise, in which more surface (peaks) share an equal amount of load. Additionally, considering both the error of non-linear calculations inside the contact area, which increases proportionally to the number of elements, and also the unpredictable nature of asperity distribution, there is a necessity for finding the “golden ratio” between density and mean peak height parameters (Figures 1 and 7).

The ability to predict the extended adhesion forces between two contact surfaces, on a multi-asperity model, as well as the stress concentration factor estimation is a function of roughness height and peaks density, is a step further of this work. The immediate effect on the existing single asperity models like the ones of DMT (Muller et al., 1983), JKR (Johnson et al., 1971) and Maugis–Dugdale (Maugis, 1992) should also be investigated.

References


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Investigations on prying forces in flexible connections of steel beams

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Abstract

Purpose – The use of flexible connections throughout the steel structures provides a high level of stiffness compared to that of fully welded connections. Flexible connections allow for rotation to an extent, which make them perform better during earthquake than welded connections. In hanger connections, the applied load produces tension in the bolts and bolts are designed for tensile forces. When the deformation of the flange plate is equal to that of the bolts, a plastic hinge is formed in the flange plate at the weld line and the bolts are pulled to failure. If the attached plate is allowed to deform, additional tensile forces called prying forces are developed in the bolts. The paper aims to discuss these issues.

Design/methodology/approach – This paper includes the results of investigation on prying force in T-stub connection fabricated with normal grade bolts and high strength friction grip (HSFG) bolts. Finite element analysis has been carried out by creating models and analyzing the effect of external tensile force and bolt force. For different grades of bolt (4.6, 8.8, 10.9, 12.9), the prying force is calculated.

Findings – It is found that prying force is increasing with the change in grade of bolt used from normal to HSFG. The results obtained from analysis using IS 800:2007 codal provision are also included. It is observed that HSFG bolts do not allow for any slip between the elements connected and hence rigidity is increased.

Originality/value – The prying force mainly depends on geometrical parameter of the connection. In this research work, the variation of prying force was studied based on the variation in dimensions of T-stub angle section and bolt grade (4.6, 8.8, 10.9, 12.9). The method of obtaining prying force from bolt load and applied load is a unique approach. The results of FE analysis is validated with the analytical calculation as per IS 800:2007 code provisions, which shows the originality of the research.

Keywords Beam–column connection, Bolt grade, Flexible connection, Prying force, T-stub connection

Paper type Research paper

1. Introduction

Steel structures are generally the assembly of structural members by means of connection. Among all type of connections, the design of bolted connections is much more complicated. Bolts are subjected to tensile load when they are assembled in the connection. Because of the large deformation of thin connected plates, bolts are subjected to additional force called prying force. The prying action between connected parts and bolts need to be investigated because of its complexity. The past researchers have proposed a number of design strategies for bolted connections. Chasten et al. (1992) stated that for unstiffened end-plate connections of large beam sections, a plastic mechanism may occur in the end plate, while beam stresses remain elastic, due to the large beam to column strength ratio. Sherbourne and Bahaari (1994)
developed a finite element model to evaluate the moment-rotation relationship for steel bolted end-plate connections. They also studied the prying action for extended end-plate connections. It is stated that prying forces in the projected portion of the end plate increase with decrease of end-plate thickness. Swanson and Leon (2001) performed modeling for bolted T-stub connection. The main variables tested include the size of the T-stub, the gauges of the bolts, and the type and diameter of the bolts. Most of the T-stubs failed by net section fracture through the stem and by tension fracture of the bolts, but generally the failure had occurred after significant plasticization. The major contributions to the overall deformation of T-stub components were made by the flange deformation, tension bolt elongation, stem deformation and relative slip. Maggia et al. (2005) stated that the lowest value obtained for the strength capacity of each bolt line should be adopted to check flexure. Danesh et al. (2005) evaluated the initial stiffness reduction rate of connections under shear force by comparing the shear capacity of the connecting web angle and its reference shear. Pirmoz et al. (2008) proposed an equation to determine the reduction factor for initial rotational stiffness of connection using connection initial rotational stiffness, yield moment, the expected shear force and web angle dimension. Vrakas et al. (2012) studied the moment-rotation response of characteristic joints subjected to static loads. Hantouche et al. (2012) and Hantouche and Abboud (2014) mentioned the importance of finding a secondary prying force, which is related to the additional forces that are introduced into the tension bolts due to significant bending of the column flange. This study provides an insight as to whether continuity plates are necessary in columns when designing and detailing full-scale T-stub connections. Prinz et al. (2014) conducted advanced experimental and numerical investigations on bolted beam–column connections. Rui Bai et al. (2014) stated that the prying force and the dimensions of connection plates should be considered for the bolt design. Liu et al. (2017) investigated the failure modes and their corresponding tensile strengths of a thread-fixed one-side bolted T-stub. The thread resistances of bolts fixed directly to plates were evaluated by experimental analysis. The previous studies on the bolted connections focus on strength and stiffness. The present research is aimed at finding the variation of prying force in normal and high strength friction grip bolts in a T-stub connection.

2. Methodology
A finite element model of bolted connection has been created using ABAQUS software (ABAQUS, 2010) and validated with the results of Packer et al. (1989). In the present research work, the T-stub connection model was created with different grades of bolts. The connection was applied with incremental tensile load. With that applied load and bolt force, the prying force was calculated. The prying force was calculated using provisions given in section 10 of IS 800:2007 (Bureau of Indian Standards, 2007) and the results were compared.

3. Analytical investigation
3.1 Validation with past research
The present analysis is validated by previously done experimental work of Packer et al. (1989). Square and rectangular hollow sections of size 101.6 mm x 101.6 mm x 6.35 mm and 177.8 mm x 127 mm x 4.78 mm, respectively, were connected with four numbers of bolts of ASTM A325 grade. The simulation model was created and was loaded up to failure in a quasi-static manner in axial tension. The prying force was calculated using applied load and bolt force. The experimentally calculated prying force by Packer et al. (1989) was 101 kN. In the present validation, the prying force calculated using finite element simulation model was within the range of accuracy (around 10 percent).

The deformed shape of square and rectangular hollow sections connected by bolts as obtained from analytical study is shown in Figure 1.
3.2 **FE model specification**
The details of specimens which are taken for the present analysis are shown in Table I.

3.3 **Finite element model**
A finite element model consisting of solid elements was created as shown in Figure 2. The finite element model of the T-stub connection was developed using ABAQUS by merging various small components using assembly module as a single unit.

3.4 **Material property**
From experimental study, it was observed that for all members of the connection, Young’s modulus, $E$, was assumed as 203,000 MPa and Poisson’s ratio $\mu$ as 0.30. The sections were of solid category and homogeneous type. The same properties were assigned to all finite element connection models and were assembled.

3.5 **Bolt preloading**
Since bolts have preload within them, it has to be given to the bolts for the FE study also. The centroidal axis of the bolts and internal layer of the bolts has been used for creating preload of the bolt, as shown in Figure 3.

<table>
<thead>
<tr>
<th>Specimen ID</th>
<th>Column</th>
<th>T-stub</th>
<th>Bolt grade</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specimen 1</td>
<td>ISMB 150</td>
<td>ISNT 150</td>
<td>4.6</td>
</tr>
<tr>
<td>Specimen 2</td>
<td>ISMB 150</td>
<td>ISNT 150</td>
<td>8.8</td>
</tr>
<tr>
<td>Specimen 3</td>
<td>ISMB 150</td>
<td>ISNT 150</td>
<td>10.9</td>
</tr>
<tr>
<td>Specimen 4</td>
<td>ISMB 150</td>
<td>ISNT 150</td>
<td>12.9</td>
</tr>
<tr>
<td>Specimen 5</td>
<td>ISMB 550</td>
<td>ISST 250</td>
<td>4.6</td>
</tr>
<tr>
<td>Specimen 6</td>
<td>ISMB 550</td>
<td>ISST 250</td>
<td>8.8</td>
</tr>
<tr>
<td>Specimen 7</td>
<td>ISMB 550</td>
<td>ISST 250</td>
<td>10.9</td>
</tr>
<tr>
<td>Specimen 8</td>
<td>ISMB 550</td>
<td>ISST 250</td>
<td>12.9</td>
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<td>Specimen 9</td>
<td>ISMB 600</td>
<td>ISST 250</td>
<td>4.6</td>
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<td>Specimen 10</td>
<td>ISMB 600</td>
<td>ISST 250</td>
<td>8.8</td>
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<tr>
<td>Specimen 11</td>
<td>ISMB 600</td>
<td>ISST 250</td>
<td>10.9</td>
</tr>
<tr>
<td>Specimen 12</td>
<td>ISMB 600</td>
<td>ISST 250</td>
<td>12.9</td>
</tr>
</tbody>
</table>

Table I. Details of specimens used in FE analysis
3.6 Boundary conditions and meshing

While creating the FE model, interaction was created by means of surface to surface contact. The top and bottom portion of the column is completely fixed, so that displacement and rotation will be made equal to zero. In ABAQUS, by the use of option ENCASTRE ($U_1 = U_2 = U_3 = UR_1 = UR_2 = UR_3 = 0$), a fixed condition was achieved, as shown in Figure 4.

Figure 2.
FE model of bolted T-stub connection

Figure 3.
Bolt preload
The elements were created by keeping aspect ratio (ratio of length to width) as small as possible. The various global seed sizes used in the model is given in Table II. The meshing of the created model is shown in Figure 5.

3.7 Analysis
The incremental tensile load was applied at increments of ten steps within the tip of the T-stub. Once the bolt reaches its yield point, it started to fail. The failure was observed to occur at the shank of the bolt where the fasteners are tightened. The variation of stress in the T-stub connection and the bolt is shown in Figure 6(a) and (b), respectively.

4. Theoretical study
As per provisions of IS 800:2007 (Bureau of Indian Standards, 2007) (ABAQUS, 2010), the prying force, $Q$, shall be calculated using the following equation:

$$Q = \frac{l_e}{2t_e} \left[ T_e - \frac{\beta n f_y b_t t^4}{2l_e d_t} \right].$$

<table>
<thead>
<tr>
<th>Component</th>
<th>Global seed size</th>
</tr>
</thead>
<tbody>
<tr>
<td>I-section</td>
<td>50</td>
</tr>
<tr>
<td>T-stub</td>
<td>50</td>
</tr>
<tr>
<td>Bolts</td>
<td>20</td>
</tr>
</tbody>
</table>

Table II.
Global seed size
where $l_v$ is the distance from the bolt center line to the toe of the fillet weld; $l_q$ is the distance between the prying force and bolt center line; $\beta = 2$ for non-pre-loaded bolt and 1 for pre-loaded bolt; $\eta = 1.5$; $b_e$ is the effective width of flange per pair of bolts; $f_p$ is the proof stress; $t$ is the thickness of the end plate; and $T_e$ is the tensile capacity of the bolt.
5. Results and discussions

5.1 Stress vs strain behavior
The stress vs strain behavior of bolts of different grades used in columns of various size/flange thickness is shown in Figure 7(a)–(c). It can be identified that stress increases up to 3.4 times with an increase in the bolt strength from normal to high grade, while the maximum strain values, as expected decreases up to 45 percent, keeping column flange thickness the same. For the same grade of bolt under consideration, maximum strain values are found to decrease up to 98 percent with an increase in column flange thickness. It can be inferred that high strength bolts with larger column size/flange thickness are preferred for better stress–strain behavior.

5.2 Bolt load vs applied load behavior
The bolt load vs applied load behavior of bolts of different grades used in columns of various size/flange thickness is shown in Figure 8(a)–(c). It is found that bolt load of grade 12.9 increases up to 3.43 times that of grade 4.6, at maximum applied load for a specific column dimension. Keeping the bolt strength same, the bolt load taken of larger column size/flange thickness is found to increase up to 8.31 times that of the smaller one. But for columns of larger dimensions, the difference in the bolt load between bolts of different grades is only marginal.

5.3 Applied load vs deformation behavior
The applied load vs deformation behavior of bolts of different grades used in columns of various size/flange thickness is shown in Figure 9(a)–(c). It is observed that when column of...
lesser size/flange thickness is used, the difference in deformation is only marginal. The
deformation is found to be reducing up to 44 percent with an increase in bolt strength when
columns of larger size/flange thickness are used. The reduction in deformation with increase
in column size/flange thickness is identified, keeping bolt grades the same. This reduction
varies from 47 to 72 percent from normal to high grade of bolts used.

5.4 Comparison of prying force, Q, from FEM and as per provisions of IS 800:2007
(Bureau of Indian Standards, 2007)
The values of prying force obtained from FEM and as per provisions of IS 800:2007
(Bureau of Indian Standards, 2007) (theoretical results) are compared and is shown in
Table III. It is observed that the mean and standard deviation of force ratio (last column
of Table III) when larger column size/flange thickness is used are 1.13 and 0.2 percent,
respectively, whereas these values are 1.47 and 4 percent for smaller column size/flange
thickness used in connections.

6. Conclusion
The behavior of a T-stub connection is simulated through a model and the prying force
for bolts of different grade used in columns of varied sizes was calculated successfully.
The conclusions arrived from this analytical study are listed as follows:

- High strength bolt with larger column size/flange thickness is preferred because of its
  better stress–strain behavior.
For column of larger dimensions, there is only marginal difference in bolt load between bolts of different grades.

The reduction in deformation with increase in column size/flange thickness varies from 47 to 72 percent from normal to high strength grade of bolts used. Hence, high grade bolts are preferred.

Notes: (a) Column-ISMB 150; (b) Column-ISMB 550; (c) Column-ISMB 600

<table>
<thead>
<tr>
<th>Description</th>
<th>Bolt grade</th>
<th>Q (theory) (kN)</th>
<th>Q (FEM) (kN)</th>
<th>Q (FEM)/Q (theory)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ISMB 150/ISNT 150</td>
<td>4.6</td>
<td>2.45</td>
<td>3.77</td>
<td>1.53</td>
</tr>
<tr>
<td></td>
<td>8.8</td>
<td>6.93</td>
<td>10.05</td>
<td>1.44</td>
</tr>
<tr>
<td></td>
<td>10.9</td>
<td>10.19</td>
<td>14.76</td>
<td>1.44</td>
</tr>
<tr>
<td></td>
<td>12.9</td>
<td>11.92</td>
<td>17.27</td>
<td>1.44</td>
</tr>
<tr>
<td>ISMB 550/ISST 250</td>
<td>4.6</td>
<td>30.19</td>
<td>33.91</td>
<td>1.12</td>
</tr>
<tr>
<td></td>
<td>8.8</td>
<td>80.68</td>
<td>90.43</td>
<td>1.12</td>
</tr>
<tr>
<td></td>
<td>10.9</td>
<td>118.50</td>
<td>132.82</td>
<td>1.12</td>
</tr>
<tr>
<td></td>
<td>12.9</td>
<td>138.68</td>
<td>155.43</td>
<td>1.12</td>
</tr>
<tr>
<td>ISMB 600/ISST 250</td>
<td>4.6</td>
<td>30.19</td>
<td>33.91</td>
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<tr>
<td></td>
<td>12.9</td>
<td>138.68</td>
<td>155.43</td>
<td>1.12</td>
</tr>
</tbody>
</table>

Source: Bureau of Indian Standards (2007)
The results of prying forces obtained using FEM and calculated values as per IS 800:2007 (Bureau of Indian Standards, 2007) are agreeing well for columns of larger size/flange thickness used in a T-stub connection. Hence, it needs revision when columns of smaller size/flange thickness are used.

References


Further reading


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Carbon footprint and financial evaluation of an aeronautic component production using different manufacturing processes

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Abstract

Purpose – The purpose of this paper is to quantify the environmental footprint and cost and thus compare different manufacturing scenarios associated with the production of aeronautical structural components.

Design/methodology/approach – A representative helicopter canopy, i.e., canopy of the EUROCOPTER EC Twin Star helicopter described in Pantelakis et al. (2009), has been considered for the carbon footprint (life cycle energy and climate change impact analysis) along with the life cycle costing analysis. Four scenarios – combinations of different manufacturing technologies (autoclave and resin transfer molding (RTM)) and end-of-life treatment scenarios (mechanical recycling and pyrolysis) are considered.

Findings – Using the models developed the expected environmental and cost benefits by involving the RTM technique have been quantified. The environmental impact was expressed in terms of energy consumption and of Global Warming Potential-100. From an environmental standpoint, processing the canopy using the RTM technique leads to decreased energy demands as compared to autoclaving because of the shorter curing cycles exhibited from this technique and thus the less time needed. As far as the financial viability of both processing scenarios is concerned, the more steps needed for preparing the mold and the need for auxiliary materials increase the material and the labor cost of autoclaving as compared to RTM.

Originality/value – At the early design stages in aeronautics, a number of disciplines (environmental, financial and mechanical) should be taken into account in order to evaluate alternative scenarios (material, manufacturing, recycling, etc.). In this paper a methodology is developed toward this direction, quantifying the environmental and financial viability of different manufacturing scenarios associated with the production of aeronautical structures.

Keywords Composite materials, Cost analysis, Aeronautic component, Autoclave, Life cycle analysis, Resin transfer moulding (RTM)

Paper type Research paper

1. Introduction

Carbon fiber reinforced thermosetting materials demonstrate superior specific properties and corrosion resistance as compared to traditional materials. These desirable properties have made composites a common material in aerospace applications thus leading to reduction of aircraft fuel consumption and greenhouse gases. Representative examples are the new aircrafts Boeing 787 and Airbus A350 which both utilizes more than 50 percent of composite materials in their structural weight.

Several studies have been conducted for evaluating the environmental and financial viability of structures made from composite materials. Suzuki and Takahashi (2005) quantified the energy intensity of carbon fiber reinforced composites for applications to passenger cars. The results demonstrated that both the high energy intensity and cost of carbon fibers pose significant burdens for the adoption of composites.
for such applications. However, it was pointed out that their environmental performance can be improved through reusing and recycling. Duflou et al. (2012) and Song et al. (2009) carried out Life Cycle analysis (LCA) studies for quantifying the environmental footprint of composites when they are used in place of steel in automotive applications. Both studies demonstrated that composite materials outperformed steel due to the weight savings that they offer during the in use phase. In another study, Das (2011) quantified the energy demands for the production of two alternative carbon fiber types: lignin based and the conventional polyacrylonitrile (PAN) based. The analysis revealed that lignin-based carbon fiber demanded 5 percent less energy. This reduction was attributed to the energy required for the production of the carbon fiber precursor. Timmis et al. (2015) evaluated the environmental footprint reduction occurred from the adoption of composite materials in aviation. The results pointed out that although carbon fiber reinforced polymers demonstrate increased environmental impact during the manufacturing and disposal phase, they demonstrate a more environmentally friendly behavior as compared to other traditional materials (aluminum) when the in use phase of the aircraft is taken into consideration. Furthermore, the increased environmental impacts during the production of composite materials have made the development of both environmentally friendly and financially viable end-of-life treatment techniques essential. In this context (Li et al., 2016), the environmental and financial credits of different waste treatments routes (disposal, incineration for energy recovery and mechanical recycling) of carbon fiber reinforced composites are investigated. The results demonstrated that the environmental benefits occurred from mechanical recycling are impaired from the severe deterioration of the mechanical properties of the fibers leading to a low market value product. Additional efforts have been made from Shuaib and Mativenga (2016) and Howarth et al. (2014) who developed models for calculating the energy intensity of mechanical recycling of glass fiber and carbon fiber reinforced thermoset composites, respectively. The results revealed the environmental advantages of recycling carbon fibers as compared to produce new ones. Additionally, these studies pointed out that further reduction of the energy intensity of this process is achievable for higher processing rates.

In parallel, efforts have been undertaken in optimizing structural components with regard to their quality and cost (Pantelakis et al., 2009; Katsiropoulos et al., 2009; Weiland et al., 2013). In the study of Pantelakis et al. (2009), the manufacturing of a helicopter’s canopy produced using the cold diaphragm forming technique is optimized from a financial and qualitative standpoint. To this end, an optimization concept taking into account both the quality and the cost performance of a composite product was introduced and implemented. However, despite these efforts, tools and concepts allowing the simultaneous optimization of a product with regard to quality, cost and environmental impact are still not available.

The aim of this work is to contribute on developing a model for estimating the environmental footprint and cost occurring from the production of an aeronautical structural component. For this purpose for the same canopy optimized in Pantelakis et al. (2009) with regard to its cost and quality, LCA and Life cycle costing models (LCC) are implemented. For developing these models, the component is assumed to be made of a conventional carbon fiber reinforced thermosetting composite. As far as the manufacturing stage is concerned, two alternative processing routes commonly used in aeronautics, namely, the conventional autoclave process and the resin transfer molding (RTM) technique were considered.

2. Definition of the case study
The present study explores alternative manufacturing and recycling scenarios in terms of financial and environmental viability, associated with the canopy of the EUROCOPTER EC
Twin Star helicopter described in Pantelakis et al. (2009). The canopy is assumed to be made of a carbon fiber reinforced epoxy composite material, with a 60 percent fiber volume fraction. The total mass of the canopy was calculated based on Pantelakis et al. (2009) and found equal to 0.64 kg. As a result the mass of the carbon fibers and the epoxy were found equal to 0.384 kg and 0.256 kg, respectively. The canopy taken for the present study has been assumed to be the same with the thermoplastic one shown in Figure 1.

3. Methodology

3.1 Life cycle analysis

Life cycle analysis is a well-established technique for assessing the environmental impact associated with a product or a process, from raw material extraction to disposal or recycling. The goal of this technique is to quantify all the energy and material inputs and outputs as well as the emissions to the environment associated with a process or a product so as to provide a sound basis for adopting environmental-oriented decisions. In the present work, the ISO 14040:2006 was selected (ISO, 2006). ISO 14040:2006 is a commonly used tool that describes the principles for carrying out an LCA. According to the above standard, LCA involves four stages. The first stage is the goal and scope definition where the purpose of the study, the system boundaries and the level of detail are defined. The second stage of the LCA is the inventory analysis where all the inputs (raw materials, energy, etc.) and outputs (emissions to the environment) of the system are quantified. The third stage of the LCA is the impact assessment which is a vital stage because it associates the inventory data with impact categories providing additional information for deeper understanding a product’s environmental footprint. Finally, during the interpretation stage, the results of the analysis are summarized and discussed, as well as recommendations and decisions are made according to the initial goal of the analysis (Figure 2).

Based on this approach, LCA models are developed for conducting a comparative analysis between different production and recycling scenarios of the canopy. The stages taken into consideration for the present LCA were: carbon fiber production, resin production, manufacturing (autoclave and RTM) and Recycling (Figure 3). For the recycling stage, two different routes of end-of-life treatment were considered: mechanical recycling and pyrolysis. In mechanical recycling, the material is initially cut into smaller pieces by shredder and then is grinded for further size reduction (Li et al., 2016). Pyrolysis, on the other side, is a thermal process where the resin is degraded producing oil, gases and solid products (fibers, fillers and char) (Oliveux et al., 2015).

For each of the abovementioned stages, the total energy consumption as well as the Global Warming Potential-100 (GWP$_{100}$) for a period of 100 years was calculated.
The GWP$_{100}$ is a common index among LCA studies for evaluating the environmental footprint and is suitable for this study because the prevalent pollutant emitted from electricity generation is CO$_2$. The total energy consumption was calculated by multiplying the mass of each material with the energy intensity of each process (Equation (1)). The energy intensity defined as the necessary energy for the production of a material or a product per unit mass of material/product produced and was derived from literature (Table I). The GWP$_{100}$ was calculated by multiplying the kg CO$_{2eq}$ produced from the consumption of 1 kWh of electricity with the total energy demands of each process under investigation (Equation (2)). The kg CO$_{2eq}$ produced from the consumption of 1 kWh were considered equal to 0.34 kg CO$_{2eq}$/kWh (www.electricitymap.org). The functional unit of the study is the canopy of the EUROCOPTER EC Twin Star helicopter:

\[ E_i = m_i e_i, \]  
\[ \text{GWP}_{100} = E_i \frac{m_{CO_{2eq}}}{\text{kWh}}. \]
3.2 Life cycle costing

An LCC model based on the principles of the activity-based costing (ABC) method is implemented for evaluating the total cost as well as tracking off the main contributors to the total cost. In ABC all work steps, with their costs for material, work, etc., are added to build the final cost of a product. This technique demands deep understanding of the process and is able to provide a meticulous insight into the total cost. In the performed LCC, costs associated with labor, material and energy were calculated. The performed cost analysis accounts also for recycling cost which was assumed equal to the energy cost of the recycling process. For calculating the cost and indentify the major cost drivers, cost estimation relationships (CERs) were either formulated or adopted from Pantelakis et al. (2009). The CERs are functions that take into account the geometrical features of a material or a product like perimeter (PAP), surface (PAA), length (L), shape complexity, mass (m), as well as the processing parameters for calculating the final cost.

For the purpose of the LCC analysis, data available from European Commission (ec.europa.eu) associated with electricity and labor cost was taken into account. The cost of 1 kWh was considered equal to 0.114 euros and the labor cost equal to 32.6 euros/hour. The cost of the raw material was considered equal to the cost of its constituents. Additionally, the empirical assumption of 80 percent of scrap material based on Pantelakis et al. (2009) was made for both the RTM and autoclave. The energy cost for both the manufacturing and the recycling stage is calculated as follows:

$$K_i = E_i k,$$

where $K_i$ is the total energy cost; $E_i$ the total energy consumption; and $k$ the cost of 1 kWh of electricity.

Although in this study non-recurring costs are not considered, an estimation of the investment cost of autoclaving and RTM is made (Tables II and III).
4. Results

4.1 Life cycle analysis

From the performed environmental analysis, the production of raw material (carbon fiber and epoxy) is responsible for 90 percent of the total environmental impact (Figures 4 and 5). Producing carbon fibers demands temperatures up to 1,500°C (during carbonization) which in turn results in high energy consumption. For this reason, it is critical to optimize the recycling techniques in order to avoid energy consumption for the production of new carbon fibers. The energy needs for carbon fiber production are 23 and 2,000 times higher as compared to pyrolysis and mechanical recycling, respectively (Figure 6). However, despite the environmental benefits, recycling of composites materials causes a significant deterioration of the mechanical properties of the recycled material posing a burden for the wide scale application of such techniques.

Producing the canopy involving RTM as the manufacturing process outperforms autoclaving both in terms of energy consumption (Figures 4(a) and 5(a)) and Global warming Potential-100 (Figures 4(b) and 5(b)). This is because of the lower energy intensity demonstrated by the RTM technique. In RTM, dry reinforcement is placed into the mold,
low viscosity resin is injected and then the mold is heated leading to reduced curing cycles and thus reduced energy consumption as compared to autoclaving (Figure 7). However, despite the environmental credits of RTM, its involvement is expected to lead to a lower product quality (lower mechanical properties, more defects and possibly less complete impregnation) as compared to autoclave.

4.2 Life cycle costing

The results from the cost analysis (Figure 8(a)) demonstrated that the main contributors to the total cost are the material and labor costs. The high material cost is a result of the high cost of carbon fibers. The energy as well as the recycling costs occupies a negligible portion of the total cost which is less than 1 percent. For this reason, only the scenarios where pyrolysis is considered as the recycling technique are shown. Moreover, the more steps needed for preparing the mold and the need for auxiliary materials (vacuum bags, vacuum tapes, breather films, etc.) increases both the labor and the material cost of the autoclave technique up to 29 percent (cutting the consumables is the activity with the second higher cost after cutting the prepreg in autoclaving responsible for 28 percent of the labor cost) (Figure 9) and 24 percent, respectively, as compared to RTM. For both manufacturing
scenarios cutting the raw material is the main contributor to the labor cost being responsible for 46 and 64 percent of the total cost of autoclaving and RTM, respectively. Additional activities needed for preparing the final product (storing the material, non-destructive testing inspection, rework, etc.) are common in both processes and responsible for 14 percent of the labor cost for autoclaving and 19 percent for RTM. Furthermore, the reduced heating time of RTM leads to a decreased energy cost (Figure 8(b)).

Figure 10 provides an insight into the effect of the main cost drivers to the total cost. It exhibits the cost variation for storing the component revealing that this activity is more

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**Figure 8.** Total cost and energy cost for both processes

**Figure 9.** Labor cost breakdown structure

**Figure 10.** Labor cost for storing the component as a function of weight and perimeter

**Notes:**
- (a) Total LCC for autoclave and RTM; (b) energy cost comparison for autoclave and RTM
- (a) Autoclave; (b) RTM
- Non-recurring costs are not included
sensitive to a perimeter increase than to a weight increase. Non-recurring costs associated with these activities are not included.

As far as the investment cost of an autoclave is concerned, this is strongly dependent on size (diameter, length), maximum working temperature and pressure as well as heating and cooling rates. Similarly, the investment cost of RTM varies, depending on injection capacity, tool size, as well as temperature and pressure capacity. A cost comparison between an autoclave and RTM (injection/mixing machine) equipment was made based on data available from a company specializing in online commerce (www.alibaba.com) and is shown in Figure 11. From this figure, it is obvious that the investment cost of autoclaving is more than three times higher as compared to RTM. On the other hand, increased cost of the molds/tools used in RTM as well as additional costs related with the design of the tool (e.g. for defining the injection line) and are not taken into account in this study, may impair the financial performance of RTM.

For both processes, RTM and autoclaving, no relative learning curve data were available for this kind of component in order to make an estimation for massive production. However, a serial production with higher volumes will certainly lead to lower cost per part, while in parallel the environmental effect would stay the same.

5. Discussion
In this study, LCA and LCC models were developed so as quantify the environmental and financial impacts of alternative manufacturing processes (autoclave and RTM) associated with the production of a helicopter's canopy. The canopy was assumed to be made from carbon fiber reinforced thermosetting composite. The results from the analyses demonstrated that RTM outperforms the conventional autoclave technique from an environmental and financial standpoint. The environmental benefits of RTM is the result of the lower energy intensity of the process as compared to autoclaving. In all scenarios, the production of the raw material (carbon fiber and resin) was responsible for almost all the environmental footprint arising significant concerns about their environmental performance. Although there are recycling techniques developed (mechanical recycling, pyrolysis) the resulting material suffers severe deterioration of the mechanical properties. Furthermore, the necessity for auxiliary materials (vacuum bag, breather films, etc.) needed in autoclaving leads to an increased labor and material cost. Additionally, the investment cost of autoclaving was found to be more than three times higher as compared to RTM. On the other hand, autoclaving is still the mainstream process for producing aeronautical components because it provides higher product
quality as compared to RTM. Given that quality is of primary concern when producing an aeronautical structural component, the above findings underline the need for a trade-off between quality, cost and environmental footprint in the process of optimizing the component and selecting the technique for manufacturing it.

6. Conclusion

The environmental performance in terms of energy consumption, CO₂ footprint and climate change and the financial viability of two different processing scenarios of the canopy developed in Pantelakis et al. (2009) are quantified through the implementation of two broadly applied methods, the LCA and LCC. The results demonstrated the advantage of RTM both in terms of cost and environmental footprint which is the combining result of the lower heating times and the less steps needed for preparing the mold. Furthermore, autoclaving exhibits a higher investment cost as compared to RTM. On the other hand, processing by using the autoclave technique is expected to lead to a higher product quality as compared to the use of RTM. This makes evident the need of a trade-off between different design, manufacturing and recycling options in the process of optimizing a component as well as the necessity of developing tools and concepts making it manageable.

References


Further reading


European Commission, available at: ec.europa.eu


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