ABSTRACT We developed a high-speed surface mounted permanent magnet synchronous motor (SPMSM) and a pulse width modulation (PWM)-driven inverter for a turbocharger with an additional electrically driven compressor (TEDC). These systems operate at 70 000 rpm and are intended to fit 1.6-L diesel vehicles to reduce turbo lag to within 0.4 s. There are key issues in the design of rotors for SPM type machines. To prevent the PM from scattering during high-speed operation, a retaining sleeve is necessary in the rotor. Several studies have reported on rotor design and the effect of retaining sleeve materials, but no papers have applied appropriate modeling and analysis to composite material sleeves. In this study, a specialized tool was used to obtain accurate results for a composite material sleeve. The results of theoretical analyses and finite element analyses (FEA) of rotors with metallic and composite sleeves were compared, considering various stress sources such as interference fit, centrifugal force and temperature gradient. In addition, rotor dynamics and transient response analyses were performed using the proposed models to verify structural stability and response time. The proposed rotor was assembled with the stator and tested in connection with the 1.6-L diesel engine. It was confirmed that the transient acceleration performance was improved by 49.14%.

INDEX TERMS Analytical models, carbon fiber reinforced plastic, electric turbocharger, permanent magnet synchronous motors, rotor with sleeve, shrink fitting.

I. INTRODUCTION
In the automotive industry, emission and fuel efficiency regulations are being strengthened and made compulsory with the aim of reducing greenhouse gases. This has stimulated the development of technologies to improve fuel efficiency and meet emission regulations. One of these technological approaches involves the downsizing of the internal combustion engine (ICE). The key of downsizing is to reduce emissions while increasing power. The most important component to achieve this downsizing is the turbocharger or supercharger, otherwise known as forced induction systems (FIS) [1], [2]. Turbochargers are driven by exhaust gases as they leave the engine. However, if the exhaust gas is insufficient to rotate the turbine at the initial low engine revolutions per minute (rpm), the increase in engine power is delayed until there is enough exhaust gas to spin the turbine. This response delay, called a turbo lag, is a disadvantage in a turbocharger.

To eliminate turbo lag, an electric motor can be used to supply air at low engine rpm. We developed a high-speed surface mounted permanent magnet synchronous motor (SPMSM) for the turbocharger system, as shown in Fig. 1 and Table 1. These systems operate at 70 000 rpm and a power of 2.3 kW.
Permanent magnet (PM) rotors are divided into two structural types, based on how the PMs are attached to the rotor. Interior permanent magnets (IPM) are embedded within the core of its rotor while surface mounted permanent magnets (SPM) are attached to the outer surface of the rotor [4], [5]. Rare earth magnets such as neodymium iron boron or samarium cobalt are widely used in high speed PM machines due to their high residual magnetic flux density and coercive force. These PM materials have high compressive strength but low tensile strength and can’t withstand the centrifugal force generated by high rotational speed [6]. Hence, it is a major challenge in high speed SPM type machines to ensure PMs survive at high operating speed.

For SPM motors as well as IPM motors, pre-stressing the PM using retaining sleeves (can or bandage) is essential to compensate the centrifugal force imposed by high speed operation (above 6000 rpm) [7]. Normally, two kinds of retaining sleeves are used. One is a nonmagnetic metallic sleeve such as stainless steel, titanium alloy, inconel, and the other is a carbon fiber reinforced plastic (CFRP) sleeve. Nonmagnetic metallic materials have better thermal conductivity, which makes cooling the rotor easier. Its disadvantage is its higher electrical conductivity, which usually results in much higher eddy current losses in the retaining sleeve. Its density is also higher than composite materials. In contrast, the composite materials have a high strength to weight ratio and very low electrical conductivity, and this results in very low eddy current heat generation in the retaining sleeve. The disadvantage of these materials is relatively lower thermal conductivity, which makes cooling the magnets difficult [6].

Some previous papers have discussed the design and analysis of SPM rotors in relation to the materials of the retaining sleeve. Wang et al. [8] proposed a comprehensive optimization design of a PM rotor structure retained by a nonmagnetic alloy with multiple constraints, such as deformation, stress. Hong et al. [9] and Riemer et al. [10] presented a structural analysis based on the interference fit and rotor dynamics of an SPM rotor with a metallic sleeve. Hong et al. [11] conducted a mechanical analysis according to the thickness of a metallic sleeve and shrink fitting, as well as an electromagnetic analysis considering eddy currents in a metallic sleeve. Cheng et al. [12] proposed analytical models of a metallic sleeve considering three types of stress sources, interference fit, centrifugal force and temperature gradient. Ahn et al. [13], Shang et al. [14] and Xu et al. [15] presented numerical analyses of metallic sleeves in different high speed applications.

Analytical models of composite material sleeves are more complicated than those for metallic sleeves because the composites are anisotropic materials. Some research articles have presented analytical models of composite material sleeves under the assumption that it is isotropic [16]–[23]. A few papers have conducted numerical analyses of composite sleeves as anisotropic materials. Tzeng [24] investigated mechanical issues at the interfaces of the rotor including press-fit, rotor dynamics, preload relaxation, and a potential rotor failure mechanism. Tang and Zhang [25] dealt with the stress of a rotor subjected to centrifugal loads, the initial stresses and deformation of two rings and multiple rings in interference fit were analyzed using a mathematical method and finite element method (FEM). Zhi et al. [26] addressed the analytical model for rotor strength with multiple carbon fiber rings considering wrapping process and the finite element verification was carried out under the conditions of room and high temperature. Chen et al. [27] studied the influence of PMs and pole fillers with different densities and coefficients of thermal expansion, on carbon composite sleeve rotor stress using a proposed analytical solution and FEM. Wu et al. [28] presented the unit model of a composite rotor core consisting of a multilayer winding with a preload equivalent to the interference fit between the different carbon fiber layers. Some papers have provided eddy current loss analyses of carbon composite sleeves [29]–[31]. Other papers have compared the eddy current losses in carbon composite and metallic sleeves. Based on the heat source, a thermal analysis was also conducted using an analytical model and FEM [32]–[39].

In practice, it is more difficult to control pre-stress in a composite material sleeve, while pre-stress can be adjusted.

### TABLE 1. Specifications of the electric turbocharger motor.

| Item                        | Specification |
|-----------------------------|---------------|
| Motor topology              | Brushless AC (BLAC) |
| Poles/Slots                 | 2/12          |
| Permanent magnet            | Samarium cobalt |
| DC voltage source [V]       | 48            |
| Rated power [kW]            | 2.3           |
| Rated speed [rpm]           | 70,000        |
| Efficiency [%]              | 93 (goal)     |
| Response time [sec]         | 0.4 (goal)    |
| Stator O. D [mm]            | 86            |
| Stator stack length [mm]    | 14.8          |
| Rotor O. D [mm]             | 26            |
| Permanent magnet length [mm]| 17            |
| Air-gap [mm]                | 1             |
element analysis (FEA) was also conducted using the factor gradient. To verify the design of an SPM type rotor, a finite such as interference fit, centrifugal force and temperature for the several reasons mentioned above.

but it is not suitable for carbon fiber reinforced plastic sleeve plex [40]. Jung [41] described a structural safety factor et al. [40] and PM. Accordingly, failure analysis should be conducted to verify the design of rotor and sleeves in an SPM type motor to ensure stability. Generally, failure prediction for metallic sleeves is performed by comparing stresses or strains caused by applied loads with the allowable strength or strain capacity of the material. For isotropic materials that exhibit yielding, either the Tresca maximum shear stress theory or the von Mises distortional energy theory are normally used to determine safety factor. However, composites are anisotropic and they do not yield. Failure modes in composites are generally noncatastrophic and may involve localized damage via such mechanisms as fiber breakage, matrix cracking, debonding and fiber pull-out. These can progress simultaneously and interactively, making failure prediction for composites complex [40]. Jung et al. [41] described a structural safety factor but it is not suitable for carbon fiber reinforced plastic sleeve for the several reasons mentioned above.

This paper proposes analytical models of retaining sleeves based on their materials, considering various stress sources such as interference fit, centrifugal force and temperature gradient. To verify the design of an SPM type rotor, a finite element analysis (FEA) was also conducted using the factor of safety for the metallic sleeve and the failure theory for the CFRP sleeve. In addition, rotor dynamics and transient response analyses were conducted for two type rotors to verify their structural stability and effective response time, to eliminate turbo lag.

II. THEORETICAL ANALYSIS OF RETAINING SLEEVE ACCORDING TO MATERIALS

Materials can be classified as either isotropic or anisotropic. Isotropic materials have the same material properties in all directions. In contrast, anisotropic materials have different material properties in all directions at a point in the body. However, if the material is anisotropic, such as a composite ply, its properties, such as modulus of elasticity, ultimate strength, Poisson’s ratio and thermal expansion coefficient and etc., will vary with the direction within the material. Composites are a subclass of anisotropic materials that are classified as orthotropic. Orthotropic materials have properties that are different in three mutually perpendicular directions. They have three mutually perpendicular axes of symmetry, and a load applied parallel to these axes produces only normal strains. However, loads that are not applied parallel to these axes produce both normal and shear strains.

Fig. 2 shows plane stress conditions in isotropic and orthotropic (anisotropic) materials. The coordinate system used to describe the orthotropic material is labeled with 1-2-3 axes. In this case, the 1-axis is defined to be parallel or longitudinal to the fibers (0°), the 2-axis is defined to lie within the plane of the plate and is perpendicular or transverse to the fibers (90°), and the 3-axis is defined to be normal to the plane of the plate. The angle θ between the x-axis and the 1-axis is called the fiber orientation angle. The 1-2-3 coordinate system is referred to as the principal material coordinate system. The second system, represented by x-y-z, is the structural loading direction or the direction in which loads are applied to the material [40].

A. METALLIC RETAINING SLEEVE

The mechanical behavior of rotors made of an isotropic material can be described using a thick walled model based on Lamé theory, which is founded on the fundamental principles of force equilibrium and the constitutive relationship between stress and strain within the material. Fig. 3 (a), (b) show an axisymmetric rotating pressured cylinder and an infinitesimal element, and (c) temperature distribution at the cylinder.

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A. METALLIC RETAINING SLEEVE

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$$\sigma_r + r \frac{d\sigma_r}{dr} - \sigma_\theta = -\rho \omega^2 r^2 $$

(1)

The direction relationship between strain and stress for a linear, homogenous, isotropic material can be expressed by Hooke’s law as (2)-(4) where \( \varepsilon_r, \varepsilon_\theta \) and \( \varepsilon_z \) are the radial, tangential and axial strain, \( u_r \) is the radial displacement. \( E \) and \( v \) denote the modulus of elasticity, Poisson’s ratio of
material respectively. \( \sigma_r, \sigma_\theta \) and \( \sigma_z \) are the radial, tangential and axial stress, \( \alpha \) is the coefficient of thermal expansion and \( \Delta T \) is the change in temperature [23].

\[
\varepsilon_r = \frac{du_r}{dr} = \frac{1}{E} [\sigma_r - \nu(\sigma_\theta + \sigma_z)] + \alpha \Delta T \tag{2}
\]

\[
\varepsilon_\theta = \frac{u_r}{r} = \frac{1}{E} [\sigma_\theta - \nu(\sigma_r + \sigma_z)] + \alpha \Delta T \tag{3}
\]

\[
\varepsilon_z = \frac{1}{E} [\sigma_z - \nu(\sigma_r + \sigma_\theta)] + \alpha \Delta T \tag{4}
\]

If the cylinder is very thin in the \( z \) direction, the stress field will be plane stress, where \( \varepsilon_z = 0 \). By properly combining (2), (3) with (1), the radial stress yields,

\[
r \frac{d^2 \sigma_r}{dr^2} + 3 \frac{d \sigma_r}{dr} + \alpha E \frac{d \Delta T}{dr} = - (3 + \nu) \rho \omega^2 r \tag{5}
\]

In addition, \( \sigma_r \) can be divided into three components, of contact pressure \( (\sigma_{r,p}) \), rotational speed \( (\sigma_{r,\omega}) \) and temperature variation \( (\sigma_{r,T}) \).

\[
\sigma_r = \sigma_{r,p} + \sigma_{r,\omega} + \sigma_{r,T} \tag{6}
\]

When inner and outer pressures \( (p_i, p_o) \) are applied at the cylinder surface as shown in Fig. 3 (a), the pressure component of the radial stress can be obtained by imposing both \( \omega \) and \( \Delta T \) equal to zero in (5). The pressure component of the radial stress is,

\[
\sigma_{r,p} = \frac{p_i r_i^2 - p_o r_o^2 + (r_i r_o / r)^2 (p_o - p_i)}{r_o^2 - r_i^2} \tag{7}
\]

where \( r_i \) and \( r_o \) are the inner and outer radius. \( \sigma_{r,\omega} \) is also obtained likewise by assuming no external pressure and temperature variation from (5). Hence, the result is,

\[
\sigma_{r,\omega} = \frac{3 + \nu}{8} \rho \omega^2 \left[ r_i^2 + r_o^2 - \left( \frac{r_i r_o}{r} \right)^2 \right] - r^2 \tag{8}
\]

In order to determine \( \sigma_{r,T} \) is assumed to have a linear temperature distribution within the cylinder, as shown in Fig. 3 (c). The change in temperature is given by,

\[
\Delta T = k_0 + k_1 r \tag{9}
\]

where \( k_0 \) and \( k_1 \) are defined as,

\[
k_0 = \Delta T_o - \frac{\Delta T_o - \Delta T_i}{r_o - r_i} r_o, \quad k_1 = \frac{\Delta T_o - \Delta T_i}{r_o - r_i} \tag{10}
\]

TABLE 2. Material properties and dimensions of metallic retaining sleeve rotor.

| Item | Inconel 718 | Sm2Co17 |
|---------------------------------|-----------|----------|
| Density [kg/m³] | 8190 | 8300 |
| Young’s modulus [GPa] | 200 | 120 |
| Poisson’s Ratio | 0.284 | 0.27 |
| Thermal expansion [μm/m°C] | 13 | 11 |
| Compressive yield strength [MPa] | - | 800 |
| Tensile yield strength [MPa] | 1100 | 35.3 |
| Inner radius [mm] | 12 | 0 |
| Outer radius [mm] | 13 | 12 |
| Rated speed [rpm] | 70 000 | 70 000 |
| Change in temperature @ inner [°C] | 81.71 | 80 |
| Change in temperature @ outer [°C] | 81.74 | 81.71 |

Therefore, substituting (9) in (5) under no pressure and rotational speed, the following equation is achieved,

\[
\sigma_{r,T} = \frac{\alpha E}{3} \frac{k_1}{r_o + r_i} \left[ r_o^2 + r_i^2 + r_i r_o - \left( \frac{r_i r_o}{r} \right)^2 \right] - r (r_o + r_i) \tag{11}
\]

The tangential stress also can be determined in the same manner from (5), leading to,

\[
\sigma_\theta = \sigma_{\theta,p} + \sigma_{\theta,\omega} + \sigma_{\theta,T} \tag{12}
\]

in which

\[
\sigma_{\theta,p} = \frac{p_i r_i^2 - p_o r_o^2 - (r_i r_o / r)^2 (p_o - p_i)}{r_o^2 - r_i^2} \tag{13}
\]

\[
\sigma_{\theta,\omega} = \frac{3 + \nu}{8} \rho \omega^2 \left[ r_i^2 + r_o^2 + \left( \frac{r_i r_o}{r} \right)^2 - \frac{1 + 3 \nu}{3} r^2 \right] \tag{14}
\]

\[
\sigma_{\theta,T} = \frac{\alpha E}{3} \frac{k_1}{r_o + r_i} \left[ r_o^2 + r_i^2 + r_i r_o + \left( \frac{r_i r_o}{r} \right)^2 \right] - 2r (r_o + r_i) \tag{15}
\]

The interference pressure between two cylinders according to the amount of radial interference in shrink fitting cannot be determined from equilibrium equations because the problem is statically indeterminate. Hence, a deflection analysis must be incorporated in the solution. Consider two elements to be pressed together as shown in Fig. 4, where \( b_i > b_o \) [42]. The radial interference \( \delta \) can be determined from the displacements \( (u_r)_{b_i}, (u_r)_{b_o} \) of the inner and outer member from (2).

\[
(u_r)_{b_o} - (u_r)_{b_i} = b_i - b_o = \delta \tag{16}
\]

When the two elements are mated, a pressure \( p \) develops at their interface. The \( p \) corresponds to \( p_o \) for the inner element and \( p_i \) for the outer element. The radial displacements can be calculated using (2) with (7), (13). Since \( b_i \) and \( b_o \) are almost equal, let \( b_o = b_i = b \). The pressure \( p \) can be expressed with (17):

\[
p = \left[ \frac{1}{b_o} \left( \frac{c^2 + b^2}{b^2 - a^2} + v_o \right) + \frac{1}{b_i} \left( \frac{b^2 + a^2}{b^2 - a^2} - v_i \right) \right] \frac{\delta}{b} \tag{17}
\]
If the rotational speed and the change in temperature are not zero, the pressure can be achieved by superposition of (7) and (13) with (8), (11), (14) and (15).

Table 2 and Fig. 5 show the configuration, the material properties and dimensions used for the rotor with a metallic retaining sleeve.

The temperature changes in the inner and outer radius for each part were calculated as shown in Fig. 6 by transient computational fluid dynamics (CFD) based on heat sources in Table 3 using an electro-magnetic analysis. The transient CFD analysis time was set to 10 s because this motor was intermittently driven during the run-up period. The ambient temperature was set at 100°C considering the engine room temperature. Since the convective heat transfer coefficient is large in the air gap of the high-speed rotor, it is natural that inflow and outflow of heat actively occur in the air. This is why the temperature inside the rotor is low. The temperature changes in Table 2 are the difference between the final temperature on each part and the ambient temperature of the structural analysis (22°C).

The interference δ was varied from 5 μm to 30 μm to calculate the stress generated in each part using the above equations. The range of the interference amount may vary depending on the size of the rotor, and the interference amount was selected so that the stress in PM could occur within the tensile yield strength even if the stress was generated by centrifugal force and thermal expansion at the

![FIGURE 5. Configuration of metallic retaining sleeve rotor.](image-url)

![FIGURE 6. Temperature distribution of motor with metallic retaining sleeve at rated point (2.3 kW, 70 000rpm).](image-url)

![FIGURE 7. The radial and tangential stress of PM according to interference and radius: (a) case 1, (b) case 2, and (c) case 3.](image-url)
The radial and tangential stresses in the retaining sleeve are analyzed under different conditions. Referring to Fig. 7 (a), when there is only the retaining sleeve, the stresses at the rated speed are considered. Fig. 8 shows the stress distribution of the PM and the metallic sleeve according to temperature changes. The analysis was carried out for three cases. Case 1 is when only the amount of interference, rotation, and thermal expansion exist. Finally, case 3 is when the amount of interference is 30 µm. The radial and tangential stresses of the metallic sleeve decrease due to the addition of thermal expansion. When the interference amount was 30 µm, the result shows that it was compressed to the center of the PM. Considering the rotational speed and thermal expansion, it was confirmed that the difference between the stress generated at the center of the PM with 5 µm interference and the tensile yield strength of the PM was about 10 MPa. Fig. 8 shows that, as in the case of the PM, the closer the inner diameter of the sleeve is, the larger the compression amount is, and considering the rotational speed and thermal expansion, the compressive stress decreases. As the amount of interference increased, the tangential stress increased. It was confirmed that the tangential stress occurred within the tensile yield strength (1100 MPa) of the Inconel 718, and that the rotor structure was stable.

### B. Carbon Fiber Reinforced Plastic Retaining Sleeve

A composite material is a mixture of fibers and matrix resulting in a configuration that combines some of the best characteristics of the two constituents. The composite materials are generally stacked using thin lamina, in various combinations which are classified by the angle of the fiber such as unidirectional laminates, angle-ply laminates, cross-ply laminates, balanced laminates and quasi-isotropic laminates. The analysis of composite structures is more complicated than that of conventional metallic structures because its properties also depend on the orientation of the fibers.

In this paper, the ply analysis for three cases will be initially addressed using classical lamination theory (CLT) and then for plies to the external loads and moments. In three dimensions, the normal stresses $\sigma$, strains $\varepsilon$ and the shear stresses $\tau$ completely describing the state of deformation in the composite, are respectively denoted in matrix form [40], [44], [45]:

$$\begin{bmatrix}
\sigma_1 & \sigma_2 & \sigma_3 & \tau_{23} & \tau_{13} & \tau_{12}
\end{bmatrix}$$  \hspace{1cm} (18)

$$\begin{bmatrix}
\varepsilon_1 & \varepsilon_2 & \varepsilon_3 & \gamma_{23} & \gamma_{13} & \gamma_{12}
\end{bmatrix}$$  \hspace{1cm} (19)

Stress-strain relations can be expressed by Hooke’s law:

$$\begin{bmatrix}
\sigma_1 \\
\sigma_2 \\
\sigma_3 \\
\tau_{23} \\
\tau_{13} \\
\tau_{12}
\end{bmatrix}
= \begin{bmatrix}
C_{11} & C_{12} & C_{13} & C_{14} & C_{15} & C_{16} \\
C_{12} & C_{22} & C_{23} & C_{24} & C_{25} & C_{26} \\
C_{13} & C_{23} & C_{33} & C_{34} & C_{35} & C_{36} \\
C_{14} & C_{24} & C_{34} & C_{44} & C_{45} & C_{46} \\
C_{15} & C_{25} & C_{35} & C_{45} & C_{55} & C_{56} \\
C_{16} & C_{26} & C_{36} & C_{46} & C_{56} & C_{66}
\end{bmatrix}\begin{bmatrix}
\varepsilon_1 \\
\varepsilon_2 \\
\varepsilon_3 \\
\gamma_{23} \\
\gamma_{13} \\
\gamma_{12}
\end{bmatrix}$$  \hspace{1cm} (20)

where the stiffness components $C_{ij}$ have numerical indices.
orientation was dealt with, since some of the coupling terms in (20) are zero. In addition, if the lamina is thin enough, there is no variation in stresses \( \sigma_3, \tau_{23} \) and \( \tau_{13} \). With these simplifications, the plane stress-strain relations for an orthotropic material have the form:

\[
\begin{bmatrix}
\sigma_1 \\
\sigma_2 \\
\tau_{12}
\end{bmatrix} = \begin{bmatrix}
Q_{11} & Q_{12} & Q_{16} \\
Q_{12} & Q_{22} & Q_{26} \\
Q_{16} & Q_{26} & 2Q_{66}
\end{bmatrix} \begin{bmatrix}
\varepsilon_1 \\
\varepsilon_2 \\
\gamma_{12}
\end{bmatrix}
\] (21)

Reduced stiffness coefficients \( Q_{mn} \) are given by:

\[
\begin{align*}
Q_{11} &= E_{11}/(1-\nu_{12}\nu_{21}) \\
Q_{12} &= \nu_{21}E_{11}/(1-\nu_{12}\nu_{21}) \\
Q_{22} &= E_{22}/(1-\nu_{12}\nu_{21}) \\
Q_{66} &= G_{12}
\end{align*}
\] (22)

where \( Q_{16} \) and \( Q_{26} \) are zero in UD ply (balanced and symmetric). As shown in Fig. 2 (b), if the direction of the fiber is oriented by \( \theta \) (if it is not UD ply), it is necessary to convert the 1, 2 axes into \( x, y \) axes using the transformation equations. The stresses in the \( x-y \) coordinate system can be developed and written in matrix form as

\[
\begin{bmatrix}
\sigma_x \\
\sigma_y \\
\tau_{xy}
\end{bmatrix} = \begin{bmatrix}
c^2 & s^2 & -2cs \\
s^2 & c^2 & 2cs \\
-2cs & 2cs & c^2 - s^2
\end{bmatrix} \begin{bmatrix}
\sigma_1 \\
\sigma_2 \\
\tau_{12}
\end{bmatrix}
\] (23)

and the stresses in the 1-2 system can be written as

\[
\begin{bmatrix}
\sigma_1 \\
\sigma_2 \\
\tau_{12}
\end{bmatrix} = [T] \begin{bmatrix}
\sigma_x \\
\sigma_y \\
\tau_{xy}
\end{bmatrix}
\] (24)

where \( c = \cos \theta, s = \sin \theta, \) and the transformation matrix, \( [T] \), are defined as

\[
[T] = \begin{bmatrix}
c^2 & s^2 & 2cs \\
s^2 & c^2 & -2cs \\
-2cs & 2cs & c^2 - s^2
\end{bmatrix}
\] (25)

It can be shown that the tensor strains transform the same way as the stresses, and that

\[
\begin{bmatrix}
\varepsilon_1 \\
\varepsilon_2 \\
\gamma_{12}/2
\end{bmatrix} = [T]^{-1} \begin{bmatrix}
\varepsilon_x \\
\varepsilon_y \\
\gamma_{xy}/2
\end{bmatrix}
\] (26)

Substituting (26) into (21), and then substituting the results into (23), it is written as

\[
\begin{bmatrix}
\sigma_x \\
\sigma_y \\
\tau_{xy}
\end{bmatrix} = [T]^{-1} \begin{bmatrix}
Q^* \\
Q^* \\
Q^*
\end{bmatrix} [T] \begin{bmatrix}
\varepsilon_x \\
\varepsilon_y \\
\gamma_{xy}/2
\end{bmatrix}
\] (27)

Conducting the indicated matrix multiplications and converting back to engineering strains, we find that

\[
\begin{bmatrix}
\sigma_1 \\
\sigma_2 \\
\tau_{12}
\end{bmatrix} = \begin{bmatrix}
\bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} \\
\bar{Q}_{12} & \bar{Q}_{22} & \bar{Q}_{26} \\
\bar{Q}_{16} & \bar{Q}_{26} & 2\bar{Q}_{66}
\end{bmatrix} \begin{bmatrix}
\varepsilon_x \\
\varepsilon_y \\
\gamma_{xy}/2
\end{bmatrix}
\] (28)

where the components of the transformed lamina stiffness matrix, which are given by [45]:

\[
\begin{align*}
\bar{Q}_{11} &= Q_{11}c^4 + Q_{22}s^4 + 2(Q_{12} + 2Q_{66}) s^2 c^2 \\
\bar{Q}_{12} &= (Q_{11} + Q_{22} - 4Q_{66}) s^2 c^2 + Q_{12}(c^4 + s^4) \\
\bar{Q}_{22} &= Q_{11}s^4 + Q_{22}c^4 + 2(Q_{12} + 2Q_{66}) s^2 c^2 \\
\bar{Q}_{16} &= (Q_{11} - Q_{12} - 2Q_{66}) c^3 s - (Q_{22} - Q_{12} - 2Q_{66}) cs^3 \\
\bar{Q}_{26} &= (Q_{11} - Q_{12} - 2Q_{66}) cs^3 -(Q_{22} - Q_{12} - 2Q_{66}) c^3 s \\
\bar{Q}_{66} &= (Q_{11} + Q_{22} - 2Q_{12} - 2Q_{66}) s^2 c^2 + 6Q_{66}(s^4 + c^4)
\end{align*}
\] (29)

Fig. 9 shows a laminate with a total thickness of \( t \) constructed from \( N \) plies. The individual thicknesses of each ply are \( t_1, t_2, t_3 \) and so on. The coordinate system is established such that the midplane of the laminate includes the \( x \) - and \( y \) -axes with the \( z \) -axis normal to the midplane. It can be expressed that the in-plane displacements consist of midplane displacement, designated by the superscript 0, plus a linear displacement through the thickness, as indicated by:

\[
\begin{align*}
\varepsilon_x &= \varepsilon_x^0 + zk_x \\
\varepsilon_y &= \varepsilon_y^0 + zk_y \\
\gamma_{xy} &= \gamma_{xy}^0 + zk_z
\end{align*}
\] (30)

where \( \varepsilon_x^0, \varepsilon_y^0 \) and \( \gamma_{xy}^0 \) are midplane normal strains in the laminate, \( \gamma_{xy}^0 \) is midplane shear strain in the laminate, \( k_x \) and \( k_y \) denote bending curvatures in the laminate and \( k_z \) is a twisting curvature in the laminate. It can be shown that the applied force and resulting moments are related to the midplane strains and curvatures in matrix form by:

\[
\begin{bmatrix}
N_x \\
N_y \\
N_{xy}
\end{bmatrix} = [A] \begin{bmatrix}
\varepsilon_x^0 \\
\varepsilon_y^0 \\
\gamma_{xy}^0
\end{bmatrix} + [B] \begin{bmatrix}
k_x \\
k_y \\
k_{xy}
\end{bmatrix}
\] (33)

\[
\begin{bmatrix}
M_x \\
M_y \\
M_{xy}
\end{bmatrix} = [D] \begin{bmatrix}
\varepsilon_x^0 \\
\varepsilon_y^0 \\
\gamma_{xy}^0
\end{bmatrix} + [D] \begin{bmatrix}
k_x \\
k_y \\
k_{xy}
\end{bmatrix}
\] (34)

\( N_x \) and \( N_y \) denote normal force resultants in the \( x \) - and \( y \) -directions per unit width and \( N_{xy} \) is shear force resultant per unit width. \( M_x \) and \( M_y \) are bending moment resultants in the \( yz \) and \( xz \) planes per unit width and \( M_{xy} \) is twisting moment resultant per unit width. The extensional stiffness matrix \([A]\),
The coupling stiffness matrix $[B]$ and the bending stiffness matrix $[D]$ can be expressed as per (35)-(37) [40], [45]:

$$[A] = \begin{bmatrix} A_{11} & A_{12} & A_{16} \\ A_{12} & A_{22} & A_{26} \\ A_{16} & A_{26} & A_{66} \end{bmatrix}$$ \hspace{1cm} (35)

$$[B] = \begin{bmatrix} B_{11} & B_{12} & B_{16} \\ B_{12} & B_{22} & B_{26} \\ B_{16} & B_{26} & B_{66} \end{bmatrix}$$ \hspace{1cm} (36)

$$[D] = \begin{bmatrix} D_{11} & D_{12} & D_{16} \\ D_{12} & D_{22} & D_{26} \\ D_{16} & D_{26} & D_{66} \end{bmatrix}$$ \hspace{1cm} (37)

The elements of the stiffness matrices $[A]$, $[B]$ and $[D]$ are calculated from:

$$A_{mn} = \sum_{j=1}^{N} (\bar{Q}_{mn})_{j} (h_{j} - h_{j-1})$$ \hspace{1cm} (38)

$$B_{mn} = \frac{1}{2} \sum_{j=1}^{N} (\bar{Q}_{mn})_{j} (h_{j}^{2} - h_{j-1}^{2})$$ \hspace{1cm} (39)

$$D_{mn} = \frac{1}{3} \sum_{j=1}^{N} (\bar{Q}_{mn})_{j} (h_{j}^{3} - h_{j-1}^{3})$$ \hspace{1cm} (40)

where $N$ is the total number of plies in the laminate as shown in Fig. 9, $h_{j-1}$ is the distance from midplane to the top of the $j$th ply, $h_{j}$ is the distance from the midplane to the bottom of the $j$th ply. $h$ is taken to be positive below the midplane and negative above the midplane. $(\bar{Q}_{mn})_{j}$ is the element of stiffness coefficient matrix $[Q]$ of the $j$th ply. Each element can be calculated from (38) to (40). The terms $A_{16}$ and $A_{26}$ bring in tension-shear coupling, while $B_{16}$ and $B_{26}$ represent tension-twisting coupling. Terms $D_{16}$ and $D_{26}$ are flexure-twisting coupling when a moment is applied. In symmetric laminates, $A_{16} = A_{26} = 0$, and no normal stress-shear coupling exists. The $[B]$ matrix is also zero in this case and no extension-bending occurs. When the laminates are UD or cross-ply, $D_{16}$, $D_{26}$ are zero. However, when the laminates are loaded off-axis to the material axes, these are not zero, even for symmetric laminates. Therefore, in case of the laminates which are not balanced and symmetric, it tends to bend, twist and/or warp under applied loads and moments. To avoid bending and twisting, the balanced and symmetric laminates should be used [40], [45].

In this study, the stress applied to the CFRP retaining sleeve as shown in Fig. 10 was calculated for three cases of symmetric laminates in Fig. 11. Table 4 shows the material properties and dimensions used for the rotor with the CFRP retaining sleeve. Case 1 is UD laminates formed by stacking four layers of fibers in a single direction, and case 2 is cross-ply laminates formed by alternately symmetrical fiber directions of $0^\circ$ and $90^\circ$. Last, case 3 is symmetric laminates in which the fiber is oriented $45^\circ$ by $-45^\circ$. The thickness of each ply is 0.125 mm, and the stress of each ply was calculated using (22) to (40) as shown in Fig. 12. Looking at the stress distribution of case 1, the stress tended to decrease toward the outside of the sleeve, and for case 2, stress in the opposite direction in the second and third ply was observed. In case 3, similar stress occurred in four ply, and lower stress occurred compared to case 1.

### III. FINITE ELEMENT ANALYSES OF ROTOR AND MOTOR ACCORDING TO RETAINING SLEEVE MATERIALS

In Section II, a theoretical analysis of the two-dimensional (2D) plane was conducted according to sleeve material. However, the actual model is three-dimensional (3D) and the analysis can be made accurate by considering the components in the axial direction. Therefore, in this section, a finite element analysis (FEA) was conducted under the same conditions using a 3D model.
FIGURE 12. Radial stress distribution for 3 cases: (a) case 1, unidirectional laminates. (b) Case 2, cross-ply laminates. (c) Case 3, symmetric laminates loaded off-axis.

TABLE 5. Comparison of pressure between metallic retaining sleeve and PM.

| Case  | 2D Theoretical (MPa) | 2D FEA (MPa) | Error (2D-2D) (%) | 3D FEA (MPa) |
|-------|----------------------|--------------|-------------------|--------------|
| Case 1 | 35.66                | 35.79        | 0.36              | 39.16        |
| Case 2 | 31.86                | 31.98        | 0.38              | 35.47        |
| Case 3 | 30.65                | 30.81        | 0.52              | 31.31        |

A. STRUCTURAL ANALYSIS OF ROTOR WITH METALLIC RETAINING SLEEVE

From the results in Fig. 7 and 8, considering the change of interference, rotational speed, and thermal expansion, it was confirmed that stress was generated within the yield strength of the material under all conditions, and the PM was compressed. However, in case 3, when the interference amount was 30 $\mu$m, the compressive stress was applied to the center of the PM, and the final interference amount of the metallic retaining sleeve was determined to be 30 $\mu$m. Table 5 shows a comparison of pressure for the metallic retaining sleeve and the PM. As shown in Section II, A, case 1 is when only an amount of interference exists, and case 2 is when an amount of interference and rotation at the rated speed exist. Finally, case 3 is when the amount of interference, rotation, and thermal expansion according to temperature changes are considered. There was little difference between the theoretical analysis and 2D FEA when the cases were compared, and the error was found to be around 0.3~0.5%. Fig. 13 shows the results for the 2D and 3D FEA. Fig. 14 presents the stresses of the metallic retaining sleeve and PM under the case 3 conditions. In Fig. 7. (c), when the interference amount was 30 $\mu$m, the center of the PM was found to be compressed, but as a result of 3D FEA, it was confirmed that tensile stress occurred at the center of the PM. The reason for the different results in the 3D FEA is that the amount of deformation in the axial direction was also considered. When the safety factor of the PM and sleeve was calculated, the structural stability was also confirmed to be 2 or more.

B. STRUCTURAL ANALYSIS OF ROTOR WITH CFRP RETAINING SLEEVE

As previously mentioned, composite materials do not yield structural stability through the Tresca maximum shear stress
TABLE 6. Comparison of stress of 3rd Ply (bottom) for CFRP retaining sleeve.

| Case | 2D CLT (MPa) | 3D FEA (MPa) | Error (%) |
|------|--------------|--------------|-----------|
| Case 1 | 14.87        | 14.72        | 1.01      |
| Case 2 | -7.75        | -7.88        | 1.68      |
| Case 3 | 3.10         | 3.24         | 4.52      |

FIGURE 14. Stresses of metallic retaining sleeve and PM with 30 µm interference: (a) radial stress of PM, (b) tangential stress of PM, (c) radial stress of metallic retaining sleeve, and (d) tangential stress of metallic retaining sleeve.

theory or the von Mises distortional energy theory unlike metal materials [40]. For accurate analysis, the composite sleeve was modeled using Ansys Composite Prepost (ACP), and 3D FEA was performed for the three laminations (case 1, 2, and 3), as shown in Section II, B. The results of the two analyses were shown in Table 6. The difference between the 2D CLT and 3D FEA results in three cases was about 0.13 to 0.15 MPa, which is believed to have been different because the effect of axial length in 3D was also considered. In case 3, the difference in absolute value was 0.14 MPa, but the stress value was relatively lower than cases 1 and 2, resulting in a large error rate. Fig. 15 shows the stress in the 3rd ply of the CFRP retaining sleeve and the radial stress of the PM. It was confirmed that the PM was most compressed by the CFRP retaining sleeve in case 1.

To figure out complex failure modes, several failure criteria can be used, including the maximum stress and strain criterion, the Tsai-Hill criterion [46], or the Tsai-Wu criterion [47] for composite materials [48]. The Tsai-Wu criterion is one of the earliest failure criteria originally proposed for anisotropic materials, and was applied in this paper to verify the structural stability of the CFRP retaining sleeve. Fig. 16 shows the results of the failure prediction analysis for the three cases. When the inverse reverse factor (IRF) value was 1 or more, the CFRP was unstable. In all three cases, the maximum IRF values of 1 or less were obtained, and this confirmed that the CFRP was structurally stable even at a rotation speed of 70 000 rpm. Considering the amount of PM compression, structural stability, and the productivity of the CFRP retaining sleeve, it was confirmed that the UD laminates in Case 1 seemed the most appropriate.

C. ELECTROMAGNETIC ANALYSIS OF MOTOR ACCORDING TO RETAINING SLEEVE MATERIALS

Electromagnetic analysis was conducted with the two selected rotors based on the previously analyzed results and the stator specifications in Table 1. Table 7 shows the results of both motors at 2.3 kW and 70 000 rpm which is the rated point. Comparing the current at the rated point, the current of the CFRP sleeve rotor is about 8 A rms higher due to the smaller amount of PMs. Therefore, the copper loss was relatively high and the efficiency was slightly lower than that of the Inconel sleeve rotor. The efficiency in Table 7 excludes mechanical losses such as bearing loss, windage loss, etc. and even if 2~3% of the output power is assumed to be mechanical losses, the efficiency of the two motors is 93%, which satisfies the goals in Table 1. Fig. 17 shows the magnetic flux density distribution of the two motors, and local saturation occurred at the stator teeth, and the maximum magnetic flux density was slightly higher in the Inconel retaining sleeve.
FIGURE 16. Failure prediction analysis: (a) case 1, (b) case 2, and (c) case 3.

TABLE 7. Comparison of performance results of two motors.

| Item                  | Metallic sleeve | CFRP sleeve |
|-----------------------|------------------|-------------|
| Rated speed [rpm]     | 70 000           | 70 000      |
| Rated torque [Nm]     | 0.314            | 0.314       |
| Rated power [kW]      | 2.3              | 2.3         |
| Current [A<sub>rms</sub>] | 60.74        | 68.37       |
| Rotor core loss [W]   | N/A              | 0.69        |
| Stator core loss [W]  | 32.47            | 33.80       |
| Eddy current loss @ sleeve [W] | 11.14  | 0.26        |
| Eddy current loss @ PM [W] | 4.3           | 8.34        |
| Copper loss [W]       | 44.37            | 56.22       |
| Efficiency [%]        | 96.14            | 95.86       |

IV. ROTOR DYNAMICS ANALYSIS OF RETAINING SLEEVE ACCORDING TO MATERIALS

The resonance of the rotor is a serious problem during high speed operation, and must be avoided to prevent damage to the rotor. Resonance will occur when the rotational frequency matches its natural frequency. The speed at this time is called the critical speed. Hence, when designing high-speed machines, it is essential to make the first critical speed exceed the motor operation speed by adjusting the thickness and length of the shaft and the distance between the pair of bearings. If the first critical speed cannot be designed to be greater than the motor operating speed, the critical speed must be skipped by rapidly accelerating or decelerating the motor [49]. A rotor dynamics analysis was performed using a 3D rotor model with the proposed retaining sleeves and the ANSYS mechanical module, considering the rotation and gyroscopic effect.

A. METALLIC RETAINING SLEEVE

The results of the rotor dynamics analysis when the interference amount was 30 µm are shown in Fig. 18. 1X is the ratio of rpm to frequency of the shaft’s rotational speed, and the backward whirl (BW) and forward whirl (FW) modes, respectively, which diverge as the spin speed increases. When the FW frequency is equal to the 1X line, the rotor response occurs. This speed is called the critical speed. It was confirmed that the critical speed (155 380 rpm) of the rotor was above the operating speed. The separation margin, which is the distance between the critical speed and operating speed, was about 122%, sufficient to meet American petroleum institute (API) standard 610 [50].

FIGURE 17. Flux density distributions: (a) metallic retaining sleeve motor. (b) CFRP retaining sleeve motor.

B. CARBON FIBER REINFORCED PLASTIC RETAINING SLEEVE

Fig. 19 presents the results of the rotor dynamics analysis for the UD laminates sleeve rotor. It was confirmed that

FIGURE 18. Campbell diagram of the rotor with metallic retaining sleeve.
the critical speed (163 540 rpm) of the rotor was above the
operating speed and the separation margin was 134%. The
rotors with the metallic and CFRP retaining sleeves were
stable.

V. TRANSIENT RESPONSE ANALYSIS OF RETAINING
SLEEVE ACCORDING TO MATERIALS
In Section II and III, the structural stability of the two types
of rotors was confirmed. The rotors proposed in this paper
are intended to drive an electric turbocharger, and to reduce
turbo lag the response performance is also very important.
To confirm the response time from idle speed (10 000 rpm)
to the rated speed (70 000 rpm) will be within 0.4 s, a transient
response analysis is essential.

Fig. 20 shows a block diagram of the transient response
analysis. The bearing conditions $B$, the inertia of the impeller
and rotor $J$, and fan load $T_L$ were taken into account using
the mechanical equation (41) of the motor. A PID controller
with clamping, one with anti-wind up topology to eliminate
overshoot, was applied to control the torque at changing
speeds.

$$T_m = J \frac{d\omega}{dt} + B\omega + T_L \quad (41)$$

Fig. 21 presents the results of the response time analysis
according to sleeve materials. Before overshoot was elim-
inated, it was expressed using dotted lines. The metallic
retaining sleeve rotor took 0.36 s to reach 70 000 rpm, and the
CFRP sleeve rotor took 0.31 s. In terms of transient response
performance, it was confirmed that the CFRP rotor was better.

VI. EXPERIMENTAL VALIDATION
In the transient response analysis, the performance of the
rotor with CFRP retaining sleeve was better, so the CFRP
retaining sleeve rotor was determined to be the prototype
model and manufactured as shown in Fig. 22 [51]. In order
to measure the response time from the idle speed to the rated
speed, a back-to-back dynamometer was configured as shown
in Fig. 23, and the test was conducted after adjusting the out-
put power (2.3 kW) through the resistance load. Fig. 24 shows
the results of the transient response experiment under load
condition. The response time from idle speed to rated speed
In comparison, it was confirmed that the engine with the pressure, which measured 192.94 kPa, was reached in 4.64 s. For the original engine (without ETC), it was confirmed that the target boost of the ETC from the idle speed of the ETC was rapidly increased to the rated speed of 1.6-L diesel engine, with electric turbocharger (ETC) experiment. To confirm the rapid charging using the ETC, a transient experiment was conducted to measure the time needed to reach the target boost pressure at maximum acceleration. In the engine, idle state, a 100% acceleration position signal (APS) was input to the electronic control unit (ECU) of the 1.6-L diesel engine, and the ETC was also rapidly increased to the rated speed of the ETC from the idle speed of the ETC. For the original engine (without ETC), it was confirmed that the target boost pressure, which measured 192.94 kPa, was reached in 4.64 s. In comparison, it was confirmed that the engine with the ETC reached the target boost pressure in 2.36 s. Transient acceleration performance was improved by 49.14% as shown in Fig. 26.

VII. CONCLUSION

This study developed a high-speed rotor design based on the material of the sleeve used to prevent PM scattering. For a metal retaining sleeve, an appropriate model was prepared by checking whether the PM was sufficiently compressed according to the amount of interference. For a CFRP retaining sleeve, an analysis was conducted for various laminate stacking sequences, and an appropriate model was developed which considered the amount of PM compression, stress distribution in the plies, and productivity. According to the results of the theoretical and finite element analyses, errors within 0.5% were found for the metallic retaining sleeves, and errors within 1~5% for the CFRP. Structural stability was also confirmed through rotor dynamics analysis, and results of the transient response analyses of the two finally selected models confirmed that the performance of the CFRP sleeve was better. After the CFRP sleeve rotor was manufactured as a prototype, experiments on the transient performance of the electric turbocharger motor and engine were conducted. The results of the experiments confirmed that the CFRP sleeve rotor improved transient performance.

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