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Abstract. The silent flange used for coupling different machines/systems to an eddy current dynamometer represents one of the modular components each test-bench should use. By introducing a silent flange into a dynamometer, the coupling steps are easier and faster. For an appropriate design, the silent flange was analyzed using dedicated software during different operation procedures and scenarios, for materials that allow easy manufacturing. This study shows that the design for this silent flange model has no danger of failure due to the small deformation and the values for the equivalent stresses. The silent flange coupling is suitable for the dynamometer for his high positioning accuracy, the zero backlash and the fact that there is no motion between the shafts.

1. Introduction

The eddy current dynamometer is one of the best known dynamometers solutions, being used to measure the force, torque or power, produced by an engine, an electric machine or a transmission, by measuring simultaneously the torque and the rotational speed. Its usage is often met to determine the power characteristics of a machine under test, the fuel consumption and the harmful emissions for internal combustion engines on different loads and scenarios, and even if for complete powertrain solution, including hybrid powertrains and electric powertrains. Beside the apparently simple measurements of power, fuel and emissions, the eddy current dynamometer is used for detailed investigation and calibration of the engines. For being able to achieve best results, the eddy current dynamometer has to include different components to be coupled to the component/system under test.

The flanged coupling is used for better connection between the eddy current dynamometer and the component/system under test. It consists of a unit used for joining two different shafts within a component or a system, being able to meet stringent requirements for different testing procedures. Different coupling solutions are possible to be used, depending on the measurement precision, the tolerance to misalignment and the torsional rigidity, having different shapes and design. The disc couplings are able to transmit torque between the bolts through a series of a thin, stainless steel disc assembled in a pack. These bolts connect shaft hubs into one piece construction. Misalignments are absorbed through flexing a disc.

The objective of this paper is to highlight the calculus of a silent flange as major and mandatory component for the Schenk WT190 eddy current dynamometer. Its design was started and achieved

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during the preliminary development stage, while the structural analysis was performed before the flange final design. The final designed silent flange solution was needed in order to facilitate the mounting and unmounting operations needed for the machines under test to be coupled to the dynamometer. The silent flange represents the mandatory component needed on the test bench, without whom no tests were able to be performed while fast coupling any torque/power machine/system. This study case of the flanged coupling design used for the Schenck Eddy Current Dynamometer in order to carry the torque/the power from the torque/power machine/systems includes in deep research stages and structural analysis for the behavioral functioning under different loads. All the research stages were done in order to manufacture the flange and to make the test bench working for different types of torque/power machineries/systems, from internal combustion engines, to electric machines and to different transmission components that are able to deliver torque/power.

2. The Silent Flange Design

The flange is linking two different components in order to deliver torque/power at different rotational speeds. The disc coupling flange uses different connecting screw-bolts and fixing clamp, being a rigid coupling, mandatory to achieve no axial displacement between the coupling components. The main geometrical dimension for a coupling flange are presented in figure 1, being determined using the equations (1) to (7).

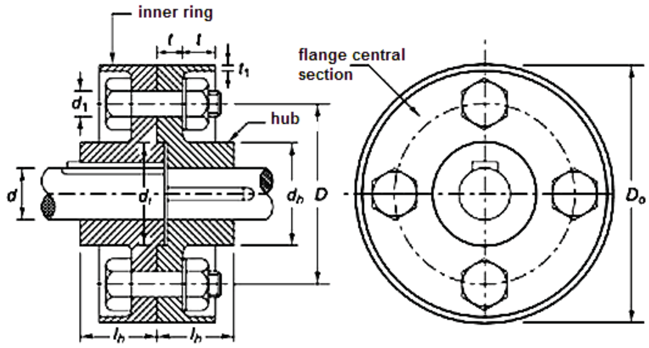

Figure 1. The main dimensions for the coupling flange [3]

$$d_b = 2 \cdot d \quad (1)$$
$$l_b = 1.5 \cdot d \quad (2)$$
$$D = 3 \cdot d \quad (3)$$
$$t = 0.5 \cdot d \quad (4)$$
$$t_1 = 0.25 \cdot d \quad (5)$$
$$d_r = 1.3 \cdot d \quad (6)$$
$$D_0 = (4 \cdot d + 2 \cdot t_1) \quad (7)$$

where d_b is the external diameter of the hub flange, l_b is the effective length of the hub flange, D is the screw-bolt diameter evenly distributed, t is the flange thickness, t_1 is the safety thickness of the outer ring, d_r is the diameter radial bearing, D_0 is the flange external diameter,

The design of the rigid flange has to accomplish the requirements that include the maximum transmitted torque, of 600 Nm, and the maximum transmitted power, of 190 kW. For achieving a safer operation after the final design, the safety values will be 1.5 higher.

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The used materials will be selected in order to respect the tensile strength, apparent yield strength, elongation at rupture and get breaking constriction. The maximum allowable stress is determined based on the maximum rupture strength.
In order to determine the running shaft external diameter, the flange do not have a dedicated shaft coupling diameter.
Analysis coupling stiffness and torque transmission can be carried out in two distinct situations. Forces acting circular individually on the screw-bolts are described in figure 2.

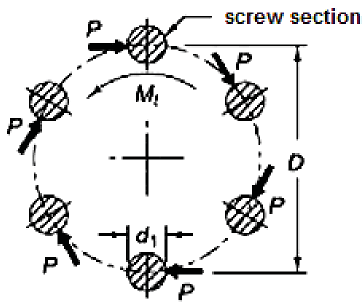

Figure 2. Screw-bolts shear strength [3]

The screw-bolts can be fastened tight and the torque transmission is made only by surface mounting flange coupling bolts. The screw-bolts are subjected to direct shear stress due to the resultant force P and are not subject to torsional stress. The resultant force is the result of shear stress. The transmitted torque is being determined using equation (8).

$$M_{tot} = P \cdot \frac{D}{2} \cdot N_b \tag{8}$$

where: M_{tot} is the transmited torque using the flange, P is the maximum power to be transmited, D is the screw-bolt circle diameter settlement, N_b is the screw-bolts total number.
If the The screws are not mounted fixed and tighted, the previous analysis can not be applied. In this case, the screws are tight enough with pre-tightening and the torque transmission is accomplished by friction. For a uniform distribution of friction, the radius of friction is given by equation (9):

$$R_f = \frac{2}{3} \cdot \frac{(R_0^3 - R_i^3)}{(R_0^2 - R_i^2)} \tag{9}$$

where R_0 is the flange external radius ($D_0/2$), R_i is the shaft diameter ($d/2$), R_f is the friciton radius.
The friction force and friction torque are determined using equations (10) and (11):

$$F_f = \mu \cdot P_i \cdot N_b \tag{10}$$
$$M_f = F_f \cdot R_f = \mu \cdot P_i \cdot N_b \cdot R_f \tag{11}$$

where: P_i is the initial tensile strnght for each screw-bolt, μ is the friction coefficient between flanges.
The studied eddy current dynamometer has its own primary flange from which the macine/system under test can be coupled. The design of this flange is made by the WT190 Schenck dynamometer manufacturer, as presented in figure 3.a. The new designed silent flange is custom made, allowing the

easier coupling for different systems to be under tests, depending on the application, as presented in figure 3.b.

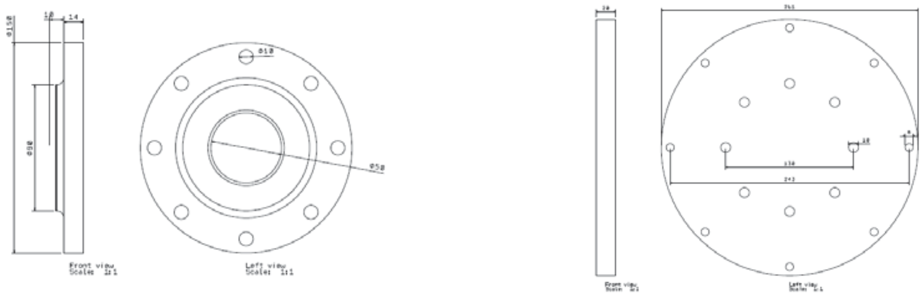

Figure 3. The ***a***-conventional flange and the ***b***-silent flange

The silent flange was designed using dedicated software for numerical analysis. Due to the complexity of the model, to the geometrical constrains and to the limitation of the post-processing levels, the designed model was simplified.

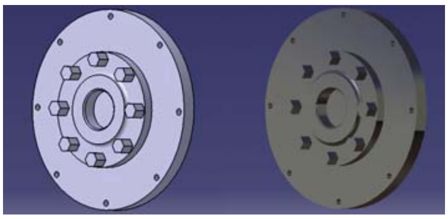

Figure 4. The silent flange sketch

3. The Silent Flange Structural Analysis

The silent flange is the component designed to provide the connection between the torque/power machine/system and the electric machine, part of the test bench. The main purpose of the flange coupling is to transfer the torque from a shaft to another shaft, having the same rotary velocity.

Critical stresses are involved while designing and testing the flange, in adition to its operation near the limit. The analysis can be performed either by finite element method (FEM) either by using empirical formulas. Some assumptions can be made if they are necessary. The chosen method for designing the flange was based on finite element modeling and analysis, using both ANSYS and CATIA. Following the needs of an elaborate study, the method is using more than one level of design. Also, different types of loading were taken into account.

The stress state of the flange, at different levels of loadings require FEM analysis and design (based on modeling and simulation), including setting the dimension and the shape, using different materials for manufacturing, while different selection criteria are taken into account, in addition to the stress evaluation stages of the silent flange virtually manufactured using the chosen materials.

The silent flange destination is defining the analysis type, including an endurance test using a buckling test rig. Using an accurate method during the silent flange stress evaluation, the areas with higher stresses are hightligted. The design requirements are helping to conclude to the silent flange stress behavior on differet stresses situations, including the distributed effort to the screw-bolt holes areas. The deformations that are present in the areas of interest are obtained.

The chosen material for manufacturing the silent flange helps achieving the needed accuracy. The used material properties are presented in Table 1.

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Table 1. The material properties

Density	CTE*	Specific Heat	Thermal Conductivity	Resistivity
7850 kg m^-3	1.2e-005 C^-1	434 J kg^-1 C^-1	60.5 W m^-1 C^-1	1.7e-007 Ohm m

*CTE= Coefficient of Thermal Expansion

Table 2. The modeling properties

Model Geometry Parts	
Object Name	*PartBody*
State	Meshed
Graphics Properties	
Visible	Yes
Transparency	1
Definition	
Suppressed	No
Stiffness Behavior	Flexible
Coordinate System	Default Coordinate System
Reference Temperature	By Environment
Material	
Assignment	Structural Steel
Nonlinear Effects	Yes
Thermal Strain Effects	Yes
Bounding Box	
Length X	1.e-002 m
Length Y	0.261 m
Length Z	0.261 m
Properties	
Volume	5.2471e-004 m³
Mass	4.119 kg
Centroid X	5.e-003 m
Centroid Y	-6.0449e-011 m
Centroid Z	-1.9521e-007 m
Moment of Inertia Ip1	3.494e-002 kg·m²
Moment of Inertia Ip2	1.7502e-002 kg·m²
Moment of Inertia Ip3	1.7507e-002 kg·m²
Statistics	
Nodes	30786
Elements	5422
Mesh Metric	None

Table 3. The silent flange analysis: loads distribution

Model (A4) > Static Structural (A5) > Loads Distribution									
Object Name	*Fixed Support*	*Force*	*Force 2*	*Force 3*	*Force 4*	*Force 5*	*Force 6*	*Force 7*	*Force 8*
State	Fully Defined								
Scope									
Scoping Method	Geometry Selection								
Geometry	16 Faces	2 Faces							
Definition									
Type	Fixed Support	Force							
Suppressed	No								
Define By		Components							

The flange analysis includes the mechanical deformation analysis using static structural, being able to provide the stresses and the deformations results.

4. Results

The geometrical model of the flange was developed. Several analysis were performed on the model. Important steps have been completed in designing the flange: setting the dimensions and the material, model development, simulation and validation, structural analysis.

Due to the small deformations and the determined equivalent stress, the designed coupling flange is suitable for being used on the Schenk Dynamometer WT 190. More of that, the zero backlash and the fact that there is no motion between the shafts make the designed coupling flange suitable for high positioning accuracy applications.
The maximum stress reaches the value of around 2.9 MPa which enables to conclude that the silent flange can be fairly used in the application of the engine, electric machine and transmissions testing. New solutions are taken into account for future research and manufacturing possibilities, including composite materials (hard special purpose plastics) in addition to iron.

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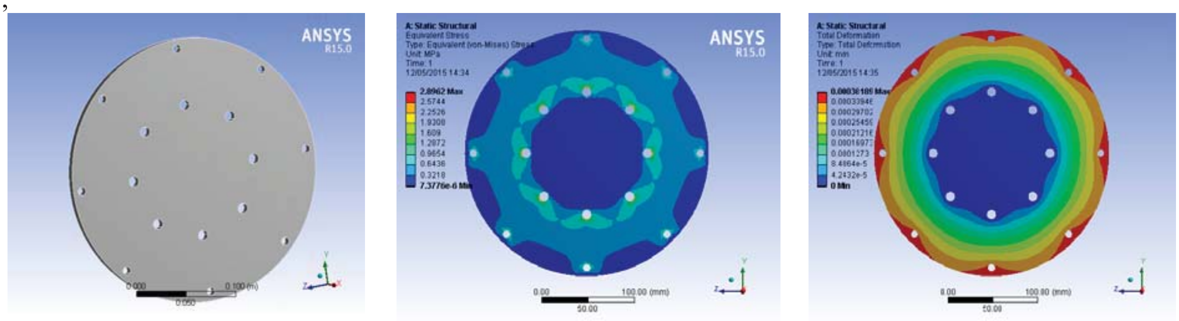

a-sketch b-equivalent stress c-maximum deformation

Figure 5. The silent flange analysis

Table 4. Results

Results		
Object Name	*Equivalent Stress*	*Total Deformation*
Minimum	7.3776 Pa	0. m
Maximum	2.8962e+006 Pa	3.8189e-007 m
Minimum Value Over Time		
Minimum	7.3776 Pa	0. m
Maximum	7.3776 Pa	0. m
Maximum Value Over Time		
Minimum	2.8962e+006 Pa	3.8189e-007 m
Maximum	2.8962e+006 Pa	3.8189e-007 m

5. Conculsions

The above presented approaches contained the stresses evaluation for the silent flange to be used on a Schenk WT 190 dynamometer. Based on its design, the silent flange dimensions were established. In addition, the material to be used for manufacturing was established.
The silent flange study is needed in order to facilitate mounting for different machines/systems under testing procedures. The allignment between the shafts is mandatory to perform safe and secure testing. The stresses behavior and deformations evolution during operation confirm the possibility of using the silent flange following its objectives.

6. References

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