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SOME ASPECTS CONCERNING THE THERMAL BEHAVIOUR OF A VISCOUS COUPLING

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Abstract
The wide use of viscous couplings in the field of automotive transmissions requires optimization methods for their operation. A key element in order to insure a proper functioning is the thermal regime. The present work deals with the computation of the temperature field in couplings in the axial direction, using the Finite Volume Method.

KEY WORDS: *coupling, silicon oil, viscosity, temperature*

1. INTRODUCTION

The structure of a viscous coupling is similar to that of a multiple-disk clutch. The gap between the disks is filled with a fluid, which transmits the torque without the disks being in contact with one another [1]. Torque transmissions in the viscous coupling is based on the transmissions of shearing forces in fluids. Although they present some doubtless advantages, the multidisk viscous couplings also present a very difficult problem, namely the uncertainty in the calculus of the transmitted torque. In fact, the torque depends on the oil viscosity, which in turn depends on the temperature field in the coupling.
In the present work, we intend to clarify some aspects regarding the quick computation of maximal temperature in this kind of couplings.

2. NOMENCLATURE

c	specific heat, [J/kg·K]
h	film thickness, [mm]
k	thermal conductivity, [W/m·K]
n	relative rotational speed, [rpm]
q	unit heat flux, [W/m^2]
u_r, u_θ, u_z	velocity components, [m/s]
A	friction area
R_i	inner radius, [mm]
R_O	outer radius, [mm]
Q	heat flux, [W]
T	temperature, [°C]
T_0	initial temperature, [°C]
α	disk frontal surface/ambient heat transfer coefficient, [W/m^2·K]
δ	disk thickness, [mm]
μ	dynamic viscosity, [Pa·s]
μ_0	dynamic viscosity at 300K, [Pa·s]
ρ	density, [kg/m^3]
ω_1	driving shaft angular velocity, [rad/s]
ω_2	driven shaft angular velocity, [rad/s]
ω	relative angular velocity, $\omega=\omega_1-\omega_2$ [rad/s]

Subscripts and superscripts

F	fluid
S	solid

3. THEORETICAL MODEL

A multidisk viscous coupling is composed, basically, of a pack of disks, separated by thin oil films. Some disks are connected to the driving shaft, while the others are linked to the driven part of the coupling. The fluid friction which occur at the film-disk interface helps transmit the driving torque. A general schema of the coupling is presented in Fig.1.

3.1 Assumptions

The complex phenomena which takes place in a multi-disk viscous coupling render its analysis to be exceedingly difficult. In order to obtain a feasible theoretical model, some simplifying assumptions are necessary. These assumptions are the following:

- the oil is a Newtonian fluid and the flow is laminar;
– the disks are perfectly parallel and equidistant during the running;
– the temperature field in a viscous coupling presents axial symmetry;
– the influence of centrifugal forces are considered;
– the flow due to pressure gradients (Poisseuille flow) is considered.

3.2 Governing equations and boundary conditions

The thermal problem of a viscous coupling present axial symmetry. Of all the corresponding speeds, u_r, u_θ, u_z, only u_θ is not zero, which is calculated at the mean radius. The temperature field in film and disks is governed by the unidimensional energy equation:

$$\rho_F c_F \frac{\partial T}{\partial t} = k_F \frac{\partial^2 T}{\partial z^2} + \mu(T)\left(\frac{\partial u_\theta}{\partial z}\right)^2 \tag{1}$$

The viscosity-temperature law is described by the Slotte relationship:

$$\mu = \mu_0 \left(T_0/T\right)^m \tag{2}$$

The speed is calculated at the mean radius, $(R_i+R_o)/2$:

$$u_\theta(r) = \omega \frac{R_i + R_o}{2} \frac{z}{h} \tag{3}$$

The temperature field in the disks is described the heat equation:

$$\rho_S c_S \frac{\partial T}{\partial t} = k_S \frac{\partial^2 T}{\partial z^2} \tag{4}$$

Realistic boundary conditions were considered for the energy equation in the oil film as well as in the disks (Fig.2), so that the heat flux continuity is maintained.

The heat flux continuity condition at film-disk interface is:

$$k_F \frac{\partial T}{\partial z} = k_S \frac{\partial T}{\partial z} \tag{5}$$

Convection boundary conditions occur at frontal surface of the terminal disks:

$$k_S \left.\frac{\partial T}{\partial z}\right|_{z=0} = \alpha \left(T\big|_{z=0} - T_0\right) \tag{6}$$

The fluid friction torque, which is calculated using the following equation:

$$M_f = \int_A \mu \frac{\partial u_\theta}{\partial z} r \, dA \tag{7}$$

4. NUMERICAL PROCEDURES

The Finite Volume Method was chosen in order to solve the model, due to some advantages as compared to the Finite Element Method (ability to handle different materials, possibility of using variable steps, stability in transient problems, etc).

Using a commercially available computer program (Fluent, Fluent Inc.), a series of numerical simulations was performed based on the input data listed in Table 2.

5. RESULTS AND DISCUSSIONS

In Fig. 3 the temperature gradients in film and disks are presented for an oil with constant viscosity. In Fig. 4, 5, 6, 7, the temperature distribution in both, film and disks is presented at inner and outer radii for two oils and two angular velocities ω.

Fig. 8 shows the temperature distribution versus radius at the middle zone of the coupling. The temperature variation in the fluid film is higher than the temperature variation in the disks. The temperature values proportionally increase with angular velocity. The maximum temperature is located in the central disk at outer radius, due to the symmetry of thermal conditions.

The best agreement between the theoretical and experimental results could be observed for the Fig. 4, 5, 6, 7, [2]. There are some differences between the experimental and theoretical results [2], due to the values of the environement temperature which can be considered constant for the theoretical models.

Because the fluid friction strongly depends on viscosity, which at its turn depends on temperature, the temperature variation inside the coupling produced a decrease in the fluid friction torque.

The determination of the temperature allows the establishment of some important functioning parameters of the viscous coupling (mainly used in the interaxle version of the vehicle integral transmissions) such as: the loading capacity (i.e., the number of the active surfaces), the dynamic viscosity of the working fluid, the materials of which the active components could be made of and the constructive dimensions of the transmission.

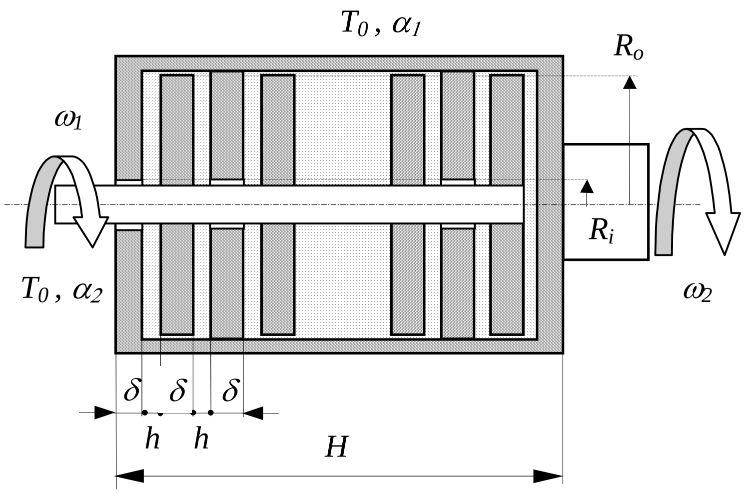

Figure 1: General scheme of a multi-disk viscous coupling

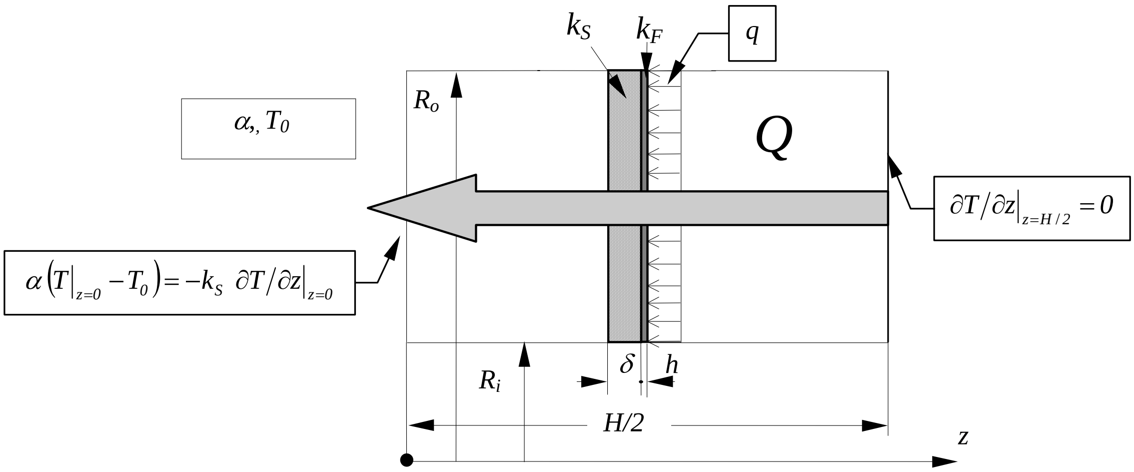

Figure 2: *Scheme of thermal boundary conditions*

Table 2: Data used in numerical simulations

Disk inner and outer radii	R_i, R_O	[mm]	30, 55
Disk thickness	δ	[mm]	2
Oil film thickness	h	[mm]	0.2
Number of friction surfaces	z	[—]	10
Relative angular speed	$\omega=\omega_1-\omega_2$	[rad/s]	10,15
Thermal conductivity	k_S, k_F	[W/mK]	40, 0.14
Density	ρ_S , ρ_F	[kg/m^3]	7840, 930
Specific heat	c_S , c_F	[J/kgK]	465, 1880
Reference temperature	T_0	[K]	300
Dynamic viscosity at T_0 for oils 1, 2	μ_{01}, μ_{02}	[Pa·s]	0.12, 0.582
Exponent in Slotte equation	m	[—]	2
Convection coefficients	α_1, α_2	[W/m^{2}K]	50, 50

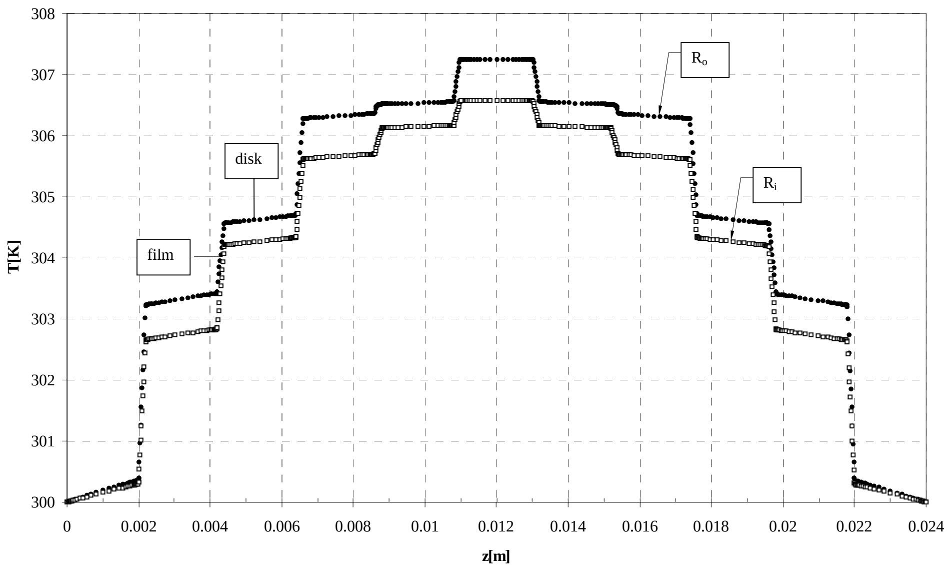

Figure 3: *Axial temperature distribution with constant viscosity, μ_{01}=0.12 at ω=10 rad·sec^{-1}*

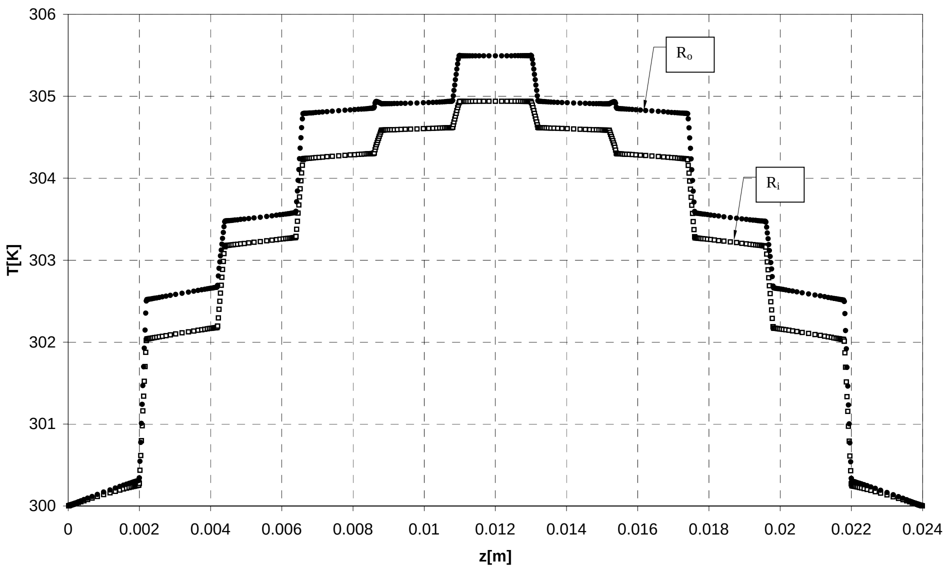

Figure 4: *Axial temperature distribution with variable viscosity, μ_{01}=0.12 at ω=10 rad·sec^{-1}*

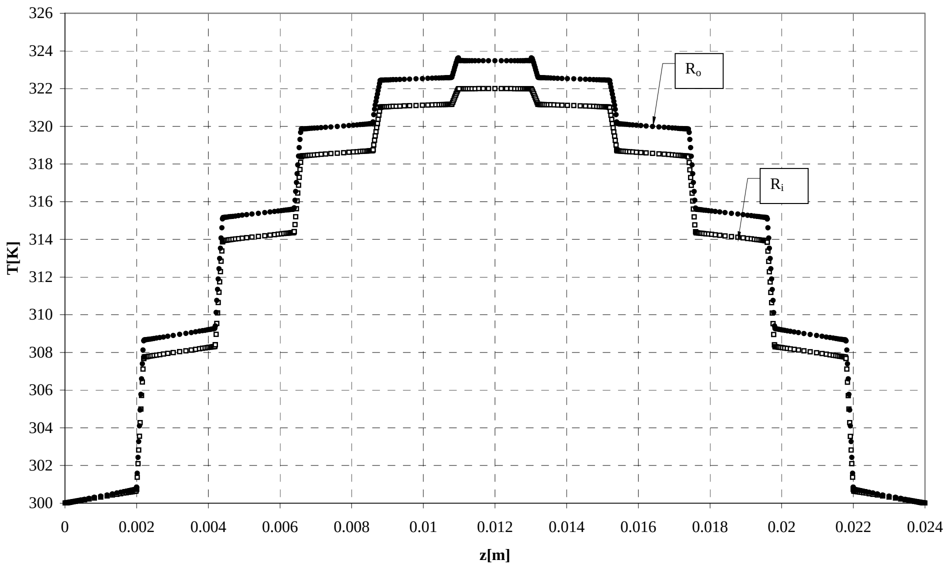

Figure 5: *Axial temperature distribution with variable viscosity, μ_{02}=0.582 at ω=10 rad$\cdot$sec^{-1}*

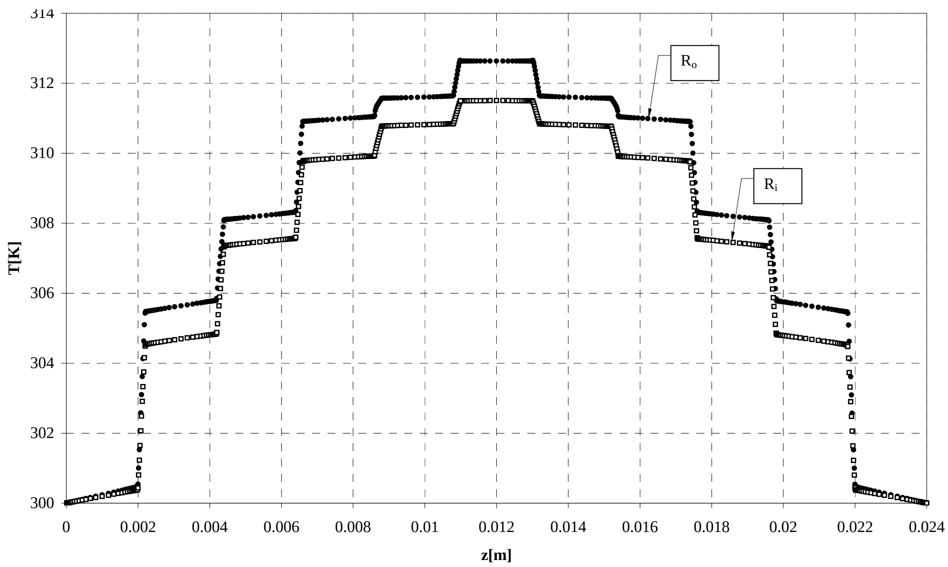

Figure 6: *Axial temperature distribution with variable viscosity, μ_{01}=0.12 at ω=15 rad$\cdot$sec^{-1}*

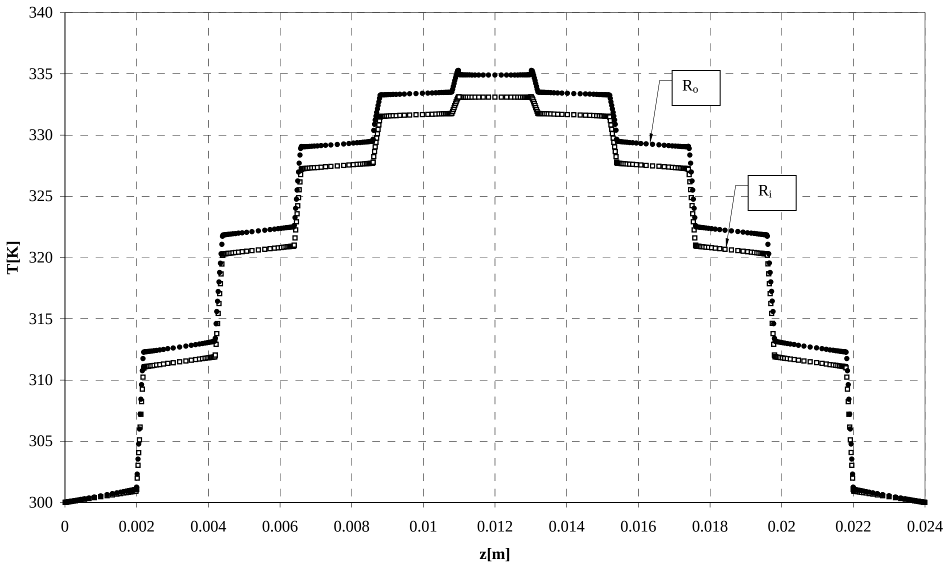

Figure 7: *Axial temperature distribution with variable viscosity, μ_{02}=0.582 at ω=15 rad$\cdot$sec^{-1}*

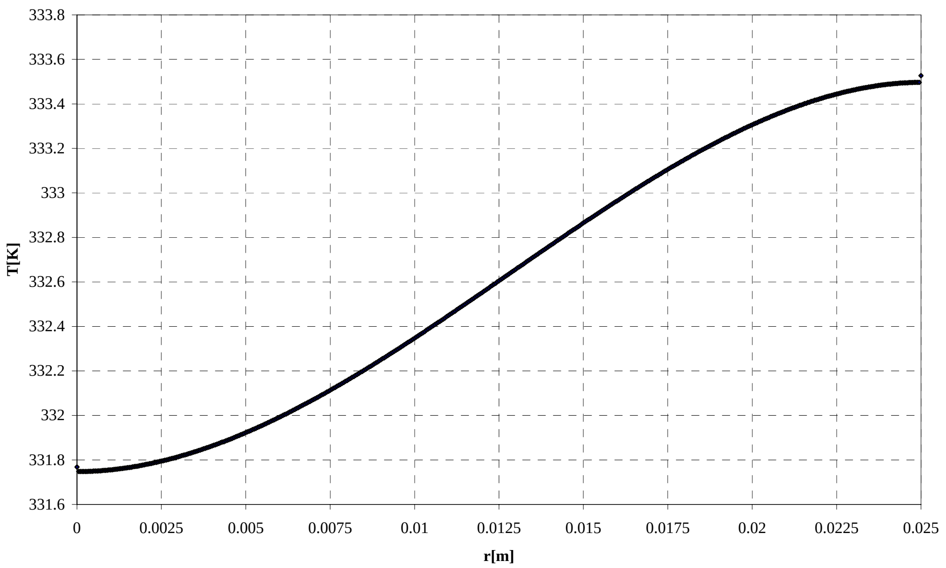

Figure 8: *Radial temperature distribution with variable viscosity, μ_{02}=0.582 at ω=15 rad$\cdot$sec^{-1}*

6. REFERENCES

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