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NUMERICAL ANALYSIS PROCEDURE FOR PREDICTING TEMPERATURE FIELD IN DESIGN OF AUTOMOTIVE FRICTION CLUTCH

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ABSTRACT−In design of the friction clutches of automobiles, knowledge on the thermo-elasticity a priori is very informative in the initial design stage. Especially, the precise prediction technique of maximum temperature and stress should be requested in design of mechanical clutches for their durability and compactness. In this study, an efficient and reliable analysis technique for the design of the mechanical clutches by using computer modeling and numerical method was developed. A commercial software STAR-CD™ was used to find the convective heat-transfer coefficients. MSC/NASTRAN™ software was followed to predict the temperature of clutch with utilization of estimated coefficients. Some experiments were also performed with a dynamometer to verify the procedure and calibrate the thermal load. As a conclusion, a design procedure, including numerical steps and experimental techniques for calibration, was proposed.

KEY WORDS : Finite element analysis, CFD analysis, Automotive friction clutch, Transient analysis, Pressure plate, Flywheel, Navier-Stokes equations

1. INTRODUCTION

A clutch is a mechanical device for quickly and easily connecting or disconnecting a pair of rotating coaxial shafts. It is usually placed between the driving motor and the input shaft to a machine, permitting the engine to be started in an unloaded state. Among many types of clutches, a clutch in popular use is the single plate, dry disc type as shown in Figure 1.

It basically consists of six major parts: flywheel, clutch disc, pressure plate, diaphragm spring, clutch cover and the linkage necessary to operate the clutch (Nunney, 1998; Orthwein, 1986). The friction clutch of automotive generates heat from the rubbed surface due to the power dissipation by the frictional slip between flywheel and clutch disc and between clutch disc and pressure plate. The generated heat has a considerable influence on the efficiency of clutch so it should be stressed to keep the clutch under some critical temperature ranges (Murali, 1998). Otherwise, the overheated clutch could fail or work at a very low efficiency. Therefore, thermal capacity of the clutch should be assessed in the initial stage of the clutch design. However, it is impractical to make complete experiments because of various and complicated driving conditions. Moreover, the deficiency of heat flux data inside the clutch adds difficulty in the precise analysis.

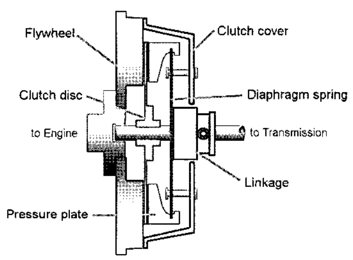

Figure 1. A schematic of a diaphragm spring clutch.

Numerous researches have been done to solve dry friction contact problem with various methods (Shin *et al.*, 2004). In order to solve this problem exactly, heat transfer, structural and contact analysis must be considered simultaneously, which requires huge computational cost and time. Even though many researches trying to solve this coupled problem directly have been done, they were limited for either simple models or steady-state operating condition analyses (Zagrodzki and Truncone, 2003; Zagrodzki and Wagner, 2002). Due to that, for a complex geometry system and for transient analysis, the de-coupled method is still widely used because it can be possible to analyze transient behavior and has excellent

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efficiency (Choi *et al.*, 2002; Kang *et al.*, 2003).

In this study, an experiment was traced by the finite element analysis to examine the thermal capacity of a clutch. Through this, attention was paid to developing a thermal analysis procedure for a friction clutch. The potential merit of study was checked whether the thermal capacity of the clutch could be considered by FE analysis with assigning the appropriate load and boundary conditions. In a clutch FE model, the most important thing is to have the appropriate boundary conditions that correspond to the real situation. However, in order to derive the appropriate load and boundary conditions, it is necessary to understand the friction works between clutch disc and flywheel and between pressure plate and clutch disc, and convective heat transfer of the environment during the clutch operation. For this end, a steady state conjugate heat transfer analysis was first performed by using CFD (computational fluid dynamics) to determine the convective heat-transfer coefficients at the solid-fluid interface. Then, the estimated convective heat-transfer coefficients were employed in heat convection boundary conditions to the axisymmetric model, and FEA (finite element analysis) was used to analyze transient temperature in the pressure plate and flywheel.

Axisymmetric analyses for the pressure plate and flywheel were done first to see the rough conditions of friction heat load conditions which could simulate the experiment situation with the initial assumption that the proportion of heat flux to the flywheel and pressure plate was 0.9. Changing the proportion value, many iterative analyses were performed comparing the analysis result with experiment one. During this process, the rough values of load conditions were adjusted to get the approximate analysis result to the experimental one, and finally the appropriate conditions were obtained.

2. EXPERIMENT

2.1. Experimental Description and Procedure

Figure 2 shows clutch assembly installed at dynamometer for experiment. The driven shaft was thus connected to a hydraulic torque controlled motor used as a brake and controlled in order to simulate vehicle inertial load. The drive shaft was powered by a 35 kW hydraulic motor whose speed could cover up to 3450 rpm. A main motor was connected to a reducer and finally to a drive shaft. The clutch was connected to the shaft. The driven shaft was linked to a braking motor. The experiment dynamometer was equipped with sensors to measure the magnitudes that can define and influence the clutch transmissibility function. Sensors were arranged in order to measure transmitted torque, rotating speed, actual contact pressure between clutch disc and flywheel and temperature of the contact surfaces. Transmitted torque, contact pressure and rotating speeds of shafts were measured by dynamometer itself. Flywheel and pressure plate temperatures were measured by chromel-alumel (K type) thermocouples placed at a depth of 1.5 mm from each mean radius points of friction surface with clutch disc. A thermocouple was placed inside of the clutch housing to measure ambient temperature (Ercole *et al.*, 2000; Velardocchia *et al.*, 1999).

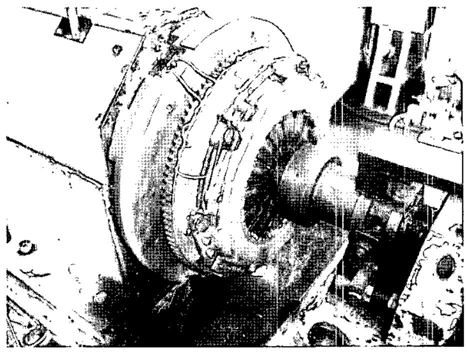


Figure 2. A clutch assembly installed at dynamometer for experiment.

Table 1. Experimental conditions.

Time (min)	20
Number of engagement	42
Input speed (rpm)	2000
Torque (N·m)	280
Friction surface outer diameter (m)	0.215
Friction surface inner diameter (m)	0.145

The experiment was designed so that the diaphragm spring clutch can be tested while simulating one of the most common severe operating conditions, i.e., frequent standing starts. Experiment conditions are summarized in Table 1.

2.2. Experimental Results

Figure 3 shows the transmitted torque history during the whole experiment period. When each rotational angular velocity of engine and clutch is regarded as input and output angular velocity, if the input and output angular velocities and torque are known, heat generated on the friction surface could be found without any limitation of the friction feature of clutch. To obtain heat flux that is applied as the heat load condition of FE model, each variation of input and output angular velocity was measured whenever clutch was engaged during experiment.

Figure 4 shows the relative angular velocity calculated by the difference of input and output angular velocities from start to finish of an engagement. Load condition of

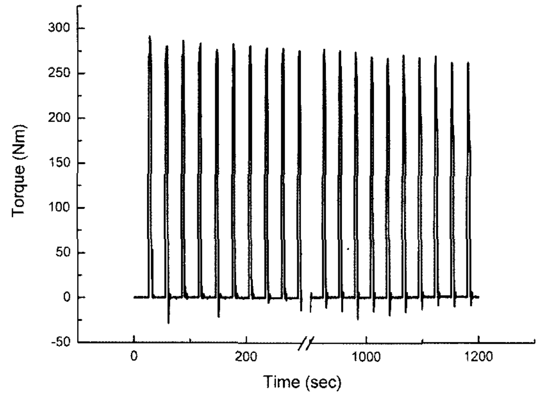


Figure 3. Transmitted torque history.

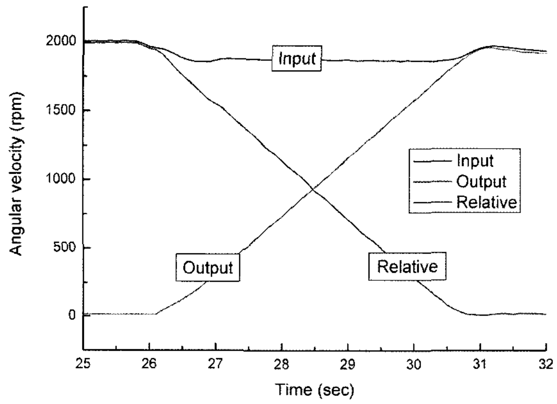


Figure 4. Relationship among measured input angular velocity and output angular velocity and calculated relative angular velocity during an engagement.

FE model was determined through the torque and relative angular velocity.

For the thermal analysis of the clutch application, a uniform heat flux is applied over the two friction surfaces (between flywheel and clutch disc and between clutch disc and pressure plate) of the clutch parts using a following formula:

$$q = \frac{\pi(\omega_i + -\omega_o) \cdot T \cdot p}{30A} \tag{1}$$

where

q : heat flux (W/m^2)

ω_i : input rotational speed (rpm*)

ω_o : output rotational speed (rpm)

T : torque (N·m)

p : proportion of heat flux to flywheel and pressure plate (initial value : p_0 = 0.9 each) (Beretta and Malfa, 2003)

A : total friction surface area of clutch (0.0396 m^2)

*1 rpm (revolution per minute) = π/30 rad/s

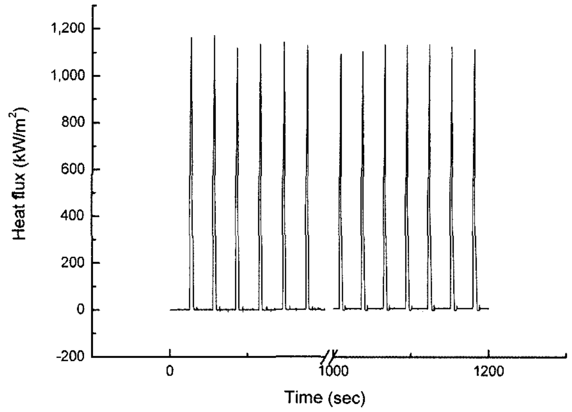


Figure 5. Heat flux input history.

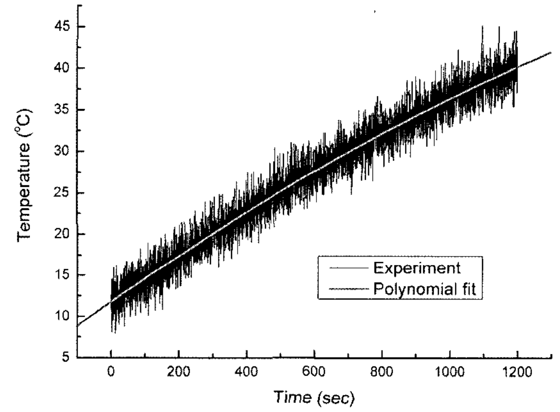


Figure 6. Atmosphere temperature history.

Figure 5 shows the calculated heat flux history with initial value of p_0. Figure 6 shows the measured atmosphere temperature along with the result of polynomial curve fitting used as the boundary condition for the FE analysis.

3. COMPUTATIONAL PROCEDURE

Computational flow chart for this analysis is summarized in Figure 7. Heat transfer including a rotating body is of major importance in the analysis and design of machinery with rotating parts (Oehlbeck and Erian, 1979; Jeng *et al.*, 1979; Sim and Yang, 1984; Arora and Stokes, 1972; Grieve *et al.*, 1998). In this study, the procedure was composed of two parts, steady state conjugate heat transfer CFD analysis and transient FE analysis. The CFD analysis was first performed to determine the convective heat-transfer coefficient field, and the result was employed in heat convection boundary condition to the transient FE analysis.

The heat flux load condition applied to the FE analysis was calculated by Equation (1), and was adjusted with

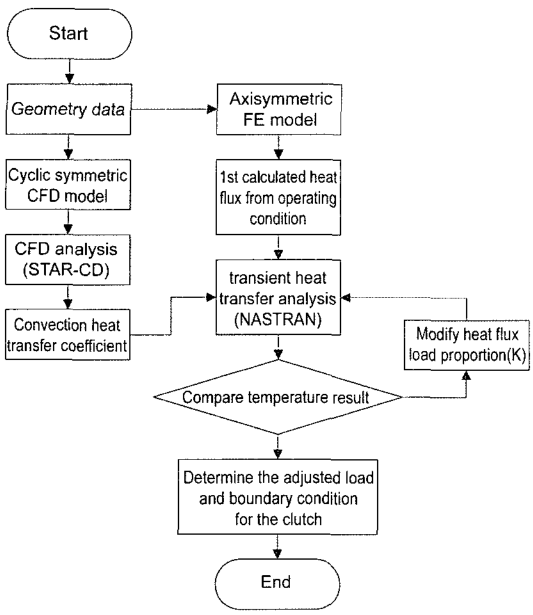


Figure 7. Computational flow chart.

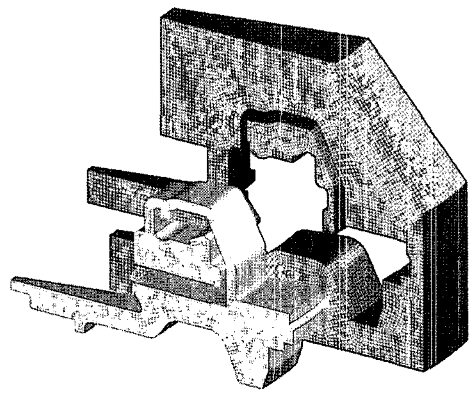


Figure 8. 3D mesh model of solid region and fluid region.

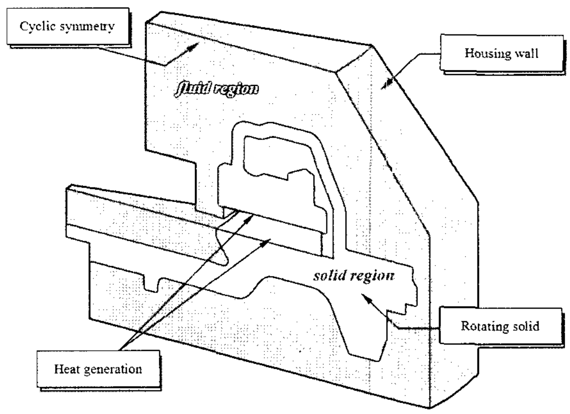


Figure 9. Computational domain and boundary conditions for CFD analysis.

changing the proportion value [p in Equation (1)] until the temperature results of FE analysis approximated to the experimental result. Through this iterative procedure, the adjusted load and boundary condition for the clutch were determined.

Based on the geometry, cyclic symmetric models were built for CFD analysis to consider the air flow in the model including rotating solid, and axisymmetric models were built for FE analysis.

3.1. CFD Analysis

Meshes for both solid (clutch parts) and fluid (air) region were created. The 3D mesh models are shown in Figure 8. Steady state conjugate heat transfer analysis was performed to find flow velocity and temperature distribution in both the solid and fluid regions. In order to enhance analysis efficiency, the parts which were considered as not affecting the result of analysis such as a small hole, fillet on the corner and joint parts, were simplified in the model.

STAR-CD™ was used for the conjugate heat transfer CFD analysis. STAR-CD™ numerically solves the generalized Navier-Stokes equations that describe most of the heat transfer and fluid flow phenomenon. The velocity field can be obtained simultaneously solving temperature and pressure fields, the continuity, momentum, energy and turbulence equations. The governing equations were solved with a rotating frame of reference. The angular velocity of clutch assembly was set constant. The steady state airflow was assumed. The generated friction heat is uniformly distributed over the whole rubbing surface area. Incompressible flow was also enforced for simplicity. In order to represent turbulence, k-ε model was employed (CD Adapco Group, 2003). The heat loss caused by convection only was considered.

Figure 9 shows the boundary conditions for the faces of the computational domain. The clutch assembly (clutch cover, pressure plate and flywheel) was rotating at 2,000 rpm (209.4 rad/s). No slip condition was used at the surface of clutch assembly and housing. Cyclic boundary conditions were used at either side of the wedge of the solid and fluid computational domain.

In the steady state conjugate heat transfer analysis, the fixed temperature boundary conditions were imposed on friction surfaces. The fixed temperatures were 170°C for flywheel and 260°C for pressure plate. These values are the average values of temperature history of each part. The exact value may not be essential because the purpose of this analysis was to estimate the heat transfer coefficient distribution on the interface of solid and fluid region. It was also assumed that the heat transfer coefficient should not change with time and depend on the flow field only.

3.2. FE Analysis

An axisymmetric element model can be applied if the shape, load and boundary condition are symmetrical with respect to an axis. However, the shape of pressure plate and flywheel under this study are not exactly symmetric, so that the analysis of a simplified model using axisymmetric elements might not reflect properly the one of the real conditions. Nevertheless, using axisymmetric element provides some advantages of easy modeling and saving the computation time and resources, compared with that using full solid elements (Chainkey, 1994; Reymond and Miller, 1994). In accordance to those reasons, it is very effective to use axisymmetric elements FE model for the complex clutch analysis, for finding the heat flow and examining the appropriateness of the boundary conditions and applied load, in spite of the some error (3% or less) due to its not perfectly exact axisymmetric geometry (Heo *et al.*, 2002). In this study, the simplified axisymmetric FE model was built, and repeated analyses were performed to determine the applied thermal load that had been used for experiment. The heat transfer boundary conditions were adopted from the CFD analysis. Because the CFD model was a cyclic symmetric model, and the FE model was an axisymmetric model, the CFD result should be modified. As results, the approximate heat flux proportions (p) for pressure plate and flywheel were determined (Cho and Ahn, 1999; 2001; Kennedy, 1981; Bathe, 1996; Dow and Burton, 1997; Lee *et al.*, 2000; Park and Park, 2000; Fukano and Matsui, 1986). Table 2 is the temperature-dependent material properties of cast iron of flywheel and pressure plate. Figure 10 shows the built FE model.

Table 2. Thermal properties of cast iron.

Temperature (°C)	0	100	300	400
Density (kg/m^3)	7196	7196	7196	7196
Specific heat (J/kg·°C)	443	468	580	636
Conductivity (W/m·°C)	53.8	50.2	43.1	39.9

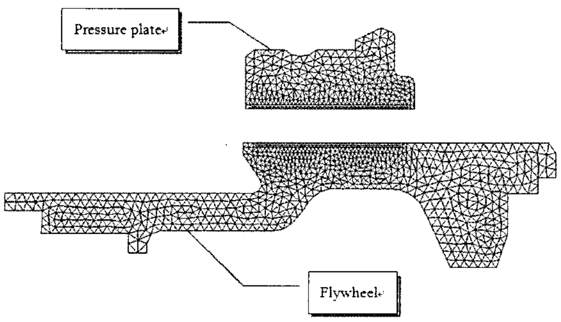


Figure 10. Axisymmetric FE clutch model.

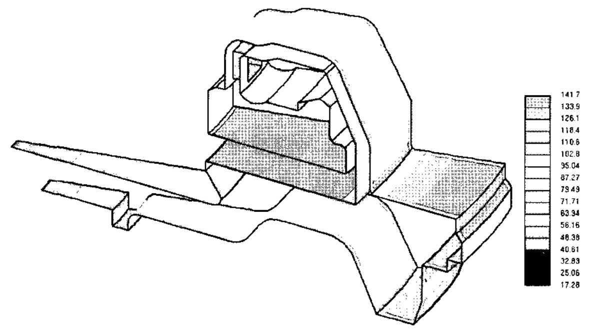


Figure 11. Convective heat-transfer coefficient contour (W/m^2·°C).

3.3. Computational Result and Discussion

Figure 11 shows the steady state CFD result of convective heat-transfer coefficient on the clutch assembly. The values of friction surfaces of pressure plate and flywheel should be ignored in this result because they are not part of interface of solid and fluid.

Because the linear velocity at outer radius region is faster than that at inner radius area, higher convective heat-transfer coefficient field (up to 141.7 W/m^2·°C) was shown at outer region.

The approximate convective heat-transfer coefficient in the case of airflow at 35 m/s over 0.75 m square plate is around 75 W/m^2·°C (Holman, 2001), and the average value of the result is around 100 W/m^2·°C.

Because the calculated convective heat-transfer coefficient field was the result of the average temperature of clutch parts, another field condition to consider the temperature effect was necessary.

Limpert (1998) suggested many empirical equations for calculating the convective heat-transfer coefficient of rotating device such as brake and clutch. For rotating solid disc the convective heat-transfer coefficient with turbulent flow may be approximated by

$$h = 0.04\left(\frac{k_a}{D}\right)\mathrm{Re}^{0.8} \tag{2}$$

$$\mathrm{Re} = \frac{VL_C}{\nu_a} \tag{3}$$

where

h : convective heat-transfer coefficient (W/m^2·°C)
k_a : conductivity of air (W/m·°C)
D : outer diameter (m)
Re : Reynolds number
V : linear speed (m/s)
L_C : characteristic length, outer diameter (m)
ν_a : kinetic viscosity of air (m^2/s)

Even though the clutch assembly is not disc, the adoption of this equation can give the reasonable values of convective heat-transfer coefficient and help to consider the temperature dependent heat-transfer characteri-

Table 3. Temperature-dependent properties of air and convection coefficient.

Temperature (°C)	0	100	300	400
Conductivity of air (W/m·°C)	0.024	0.032	0.045	0.051
Kinetic viscosity of air (m^2/s)*10^{-5}	1.37	2.33	4.74	6.17
Convection coefficient (W/m^2·°C)	130.8	112.3	90.4	82.8

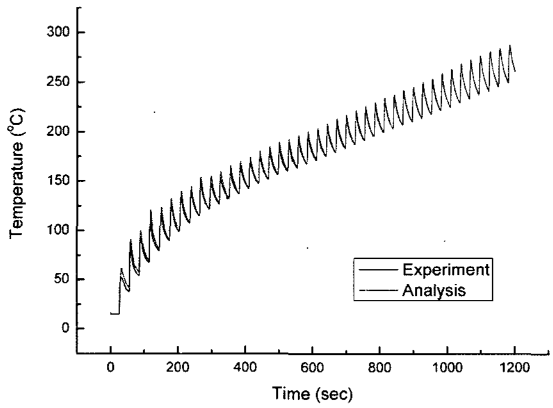
stic.

Considering the temperature-dependent material properties of air (conductivity and viscosity), the convective heat-transfer coefficient of rotating solid disc which is 0.240 m diameter and rotates at 2,000 rpm and applied temperature-dependent properties of air are shown in Table 3. As the temperature of air increases, the convective heat-transfer coefficient value decreases.

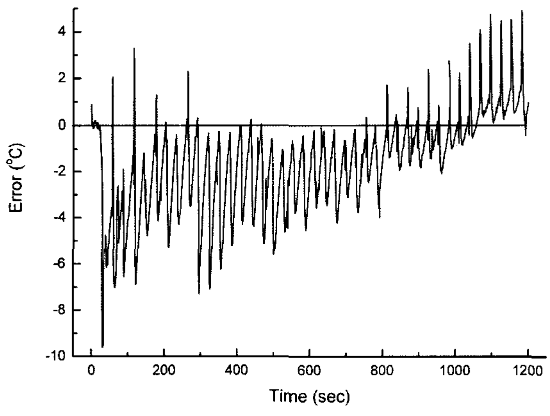
Based on these two conditions (spatial dependence and temperature dependence), convective heat transfer boundary conditions were applied in FE analysis.

For the flywheel analysis, the appropriate heat flux proportion p was estimated at 85%, and the rest 15% was regarded as the loss from the conduction to the clutch disc and in some other forms of energy. The proportion value p was determined when the temperature error at the elevated temperature range was less than 5°C. Figure 12 shows the comparison between the experimental and simulation result at a depth of 1.5 mm from mean radius point of friction surface.

As shown in Figure 12(b), calculated temperature values are lower than experiment result at the beginning of history and are higher at the ending of history. That means that the temperature-dependent heat-transfer

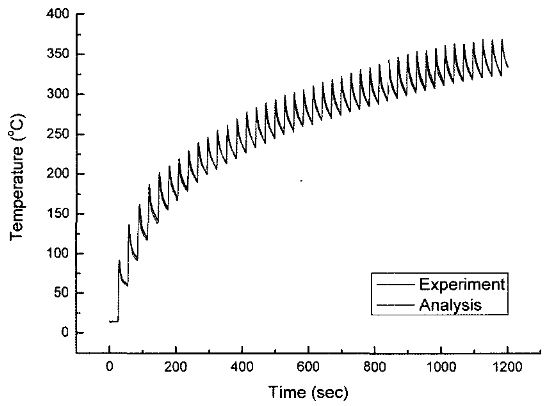


(a)

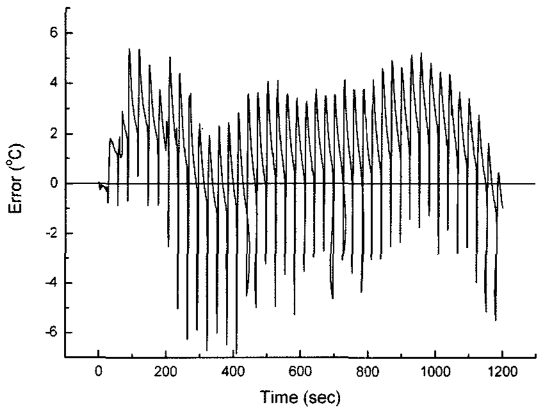


(b)

Figure 12. Temperature history of the flywheel for applied temperature-dependent convective heat-transfer coefficient field: (a) comparison between experiment and analysis; (b) estimation temperature error (analysis - experiment).



(a)



(b)

Figure 13. Temperature history of the pressure plate for applied temperature-dependent convective heat-transfer coefficient field: (a) comparison between experiment and analysis; (b) estimation temperature error (analysis - experiment).

coefficient is over-estimated in low temperature region and under estimated at high temperature, but the error range at elevated temperature region (over 100°C) is in around ± 5°C.

For the pressure plate analysis, the heat flux proportion was also estimated at 84% and the rest 16% was regarded as the loss from the conduction to the clutch disc and in some other forms of energy. The proportion value p was determined when the temperature error at the elevated temperature range was less than 5°C. Figure 13 shows the comparison between the experiment and simulation result at a depth of 1.5 mm from mean radius point of friction surface.

As shown in Figure 13(b), differently from the flywheel case, temperature error tends to neither increase nor decrease. This means the temperature dependent convective heat-transfer coefficient condition is well adopted for high temperature application. The error range at entire region is in around ± 5°C.

4. CONCLUSION

In this study, the thermal flow of flywheel and pressure plate using CFD and FEM was considered. In order to estimate the appropriate environmental conditions during the clutch operating period, the experiment was performed, and analysis results from using axi-symmetric FE model was compared with the experimental results. To simulate the experiment using CAE, exact load and boundary conditions should be applied so that the convective heat-transfer coefficient field was extracted from CFD analysis and heat flux proportion factor were deduced via such repeated comparisons. The results of the analysis applied for the finally adjusted conditions were very close to the experimental results. From these results, the suggested FE analysis procedure provides very good quality of result for understanding the thermal flow characteristics of the automotive clutch that operates intermittently.

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REFERENCES

Arora, R. C. and Stokes, V. K. (1972). On the heat transfer between two rotating disks. *Int. J. Heat Mass Transfer*, **15**, 2119–2132.

Bathe, K. J. (1996). *Finite Element Procedures*. Prentice-Hall, Inc.. New Jersey.

Beretta, G. P. and Malfa, E. (2003). Flow and heat transfer in cavities between rotor and stator disks. *Int. J. Heat Mass Transfer*, **46**, 2715–2726.

CD Adapco Group. (2003). *Methodology STAR-CD Ver.3.15*. CD Adapco Group. London, UK.

Chainky, M. (1994). *MSC/NASTRAN Thermal Analysis User's Guide V 68*. The Macneal-Schwendler Co. Santa Ana, CA, USA.

Cho, C. and Ahn, S. (1999). Periodic hot spots simulation during intermittent contact on a disc brake rotor. *Proc. 32th Int. Symp. Automotive Technology & Automation* **8, 2**, 185–194.

Cho, C. and Ahn, S. (2001). Thermo-elastic analysis for chattering phenomenon of automotive disc brake. *Int. J. Korean Society of Mechanical Engineers* **5, 5**, 569–579.

Choi, J., Kim, D., Lee, I., Cha, B. and Kang, M. (2002) Transient thermoelastic analysis of disk brakes using finite element method. *Trans. Korean Society of Automotive Engineers* **10, 5**, 160–167.

Dow, T. A. and Burton, R. A. (1997). Thermoelastic instability of sliding contact in the absence of wear. *Wear*, **19**, 315–328.

Ercole, G., Mattiazzo, G., Mauro, S., Velardocchia, M., Amisano, F. and Serra, G. (2000). Experimental methodologies to determine diaphragm spring clutch characteristics. *SAE Paper No*. 2000-01-1151.

Fukano, A. and Matsui, A. (1986). Development of disc-brake design method using computer simulation of heat phenomena. *SAE Paper No*. 860634.

Grieve, D. G, Barton, D. C., Crolla, D. A. and Buckingham, J. T. (1998). Design of a lightweight automotive brake disc using finite element and Taguchi techniques. *Proc. Instn. Mech. Engrs. Part D*, **212**, **4**, 245–254.

Heo, J., Cho, C., Cho, H., Kim, D. and Han, K. (2002). Numerical analysis procedures for reliable design of automotive mechanical clutches. *SAE Paper No*. 2002-01-0762.

Holman, J. P. (2001). *Heat Transfer*. 9th Edn. McGraw-Hill. New York.

Jeng, D. R., DeWitt, K. J. and Lee, M. H. (1979). Forced convection over rotating bodies with non-uniform surface temperature. *Int. J. Heat Mass Transfer*, **22**, 89–98.

Kang, S., Kim, C., Lee, D. and Kim, H. (2003) The thermal analysis of brake disc using the solid model and 2D coupled model. *Trans. Korean Society of Automotive Engineers* **11, 6**, 93–100.

Kennedy, Jr F. E. (1981). Surface temperature in sliding systems-A finite element analysis. *ASME J. Lubrication Technology*, **103**, 90–96.

Lee, C. Y., Chai, Y. S., Kwon, J. D., Nam, W. H. and Kim, T. H. (2000). Finite element analysis and optimal design of automobile clutch diaphragm spring. *J. Korean Society of Mechanical Engineers* **24, 6**, 1616–1623.

Limpert, R. (1998). *Brake Design and Safety*. Society of Automotive Engineers, Inc.

Murali, M. R. (1998). Failure analysis of center plate of a clutch and brake combination using finite elements. *SAE Paper No*. 982799.

Nunney, M. J. (1998). *Automotive Technology*. Society of Automotive Engineers, Inc. Warrendale. PA.

Oehlbeck, D. L. and Erian, F. F. (1979). Heat transfer from axisymmetric sources at the surface of a rotating disk. *Int. J. Heat Mass Transfer*, **22**, 601–610.

Orthwein, W. C. (1986). *Clutches and Brakes*. Marcell Dekker, Inc.. New York. U.S.A.

Park, Y. C. and Park, D. S. (2000). Thermal stress analysis of brake drum by using finite element analysis. *J. Korean Society of Precision Engineering* **8**, **3**, 77–84.

Reymond, M. and Miller, M. (1994). *MSC/NASTRAN Quick Reference Guide V 68*. The Macneal-Schwendler Co.

Shin, K., Brennan, M. J., Joe, Y-G. and Oh, J-E. (2004). Simple models to investigate the effect of velocity dependent friction on the disc brake squeal noise. *Int. J. Automotive Technology* **5**, **1**, 61–67.

Sim, Y. S. and Yang, W. J. (1984). Numerical study on heat transfer in laminar flow through co-rotating parallel disks. *Int. J. Heat Mass Transfer* **27**, **11**, 1963–1970.

Velardocchia, M., Ercole, G., Mattiazzo, G., Mauro, S. and Amisona, F. (1999). Diaphragm spring clutch dynamic characteristic test bench. *SAE Paper No*. 1999-01-0737.

Zagrodzki, P. and Truncone, S. (2003). Generation of hot spots in a wet multidisk clutch during short-term engagement. *Wear* **254**, **5**, 474–491.

Zagrodzki, P. and Wagoner, P. (2002). Thermo-mechanical effects in a single-sided multidisk clutch/brake design. *SAE Paper No*. 2002-01-1439.