

# An Experimentally Validated FE Analysis for Transient Thermal Behavior of the Rolling Tire

M. Namjoo<sup>1</sup>, H. Golbakhshi<sup>2\*</sup>, H. Momeni-Khabisi<sup>3</sup>

1: Instructor, Department of Mechanical Engineering of Biosystems, Faculty of Agricultural, University of Jiroft, 78671-61167, Jiroft, Iran. 2, 3: Instructor, Department of Mechanical Engineering, University of Jiroft, 78671-61167, Jiroft, Iran.

## Abstract

Evaluating the thermal effects and variations in temperature of rolling pneumatic tires, is a very important factor in safe performance of the vehicles. Normally, the transient thermal investigation of rolling tires is performed by tire test rigs. However, experimental analysis is a very time and cost consuming process and because of technical limitations, the tests cannot be carried out in most severe conditions. In this work, a validated finite element model is proposed for transient thermal investigation of rolling pneumatic tires. Compared with the experimental tests, the current study gives satisfactory results for temperature distribution of the tire.

**Keywords:** Finite element method, rolling tire, hysteresis loss, transient thermal analysis.

## Introduction

The rolling pneumatic tire endures alternatively compression and expansion during its motion. Due to hysteresis property of rubber, less energy is given back during unloading than it was received during loading. Inevitably, when energy dissipation takes place, temperature develops in the tire [1]. It is noted that the rubber has low thermal diffusivity. So, dangerously high local temperatures is created inside the tire domain [2]. High temperatures can strongly accelerate chemical alternation processes which lead to deteriorating the rubber network structure [3]. So, reliable prediction distribution due to hysteresis loss of the rubber is crucial for accurate estimation of safe service life and even, optimizing the fuel consumption [4].

The analysis of tire is a challenging task. The rubber shows non-linear elastic and inelastic with large deformations when subjected to external loads; while steel belts exhibit completely different material properties in anisotropic structure of the tire [1, 3]. Furthermore, the contact to the road and the rim demands for a contact formulation [5]. Due to the complexity of the problem and limitation in analytical approaches, the experimental methods required to determine the tire temperature field. For this purpose, some sophisticated systems with modern temperature and pressure monitoring

components are used for evaluating of tires under dynamic conditions [6].

In recent years, the finite element method has turned out to be a suitable substitution for extremely expensive and time consuming field tests. The thermal effects which are developed in the tires have been numerically evaluated in many published papers. The work presented by Sokolov considers internal friction and internal heat generation relative to be the mechanical energy losses [2]. Rao et al. calculated the energy loss form strain cycles in the 3D deformation analysis and the viscoelastic properties of rubber [7, 8].

In the work done by Yavari the frictional effects were also included in order to determine the temperature distribution [9]. Sarkar estimated cyclic change in stresses and strains for a 2D tire geometry to create a pseudo steady state temperature profile [10]. Luchini developed directional incremental hysteresis theory as a strain based model for considering non-sinusoidal cycles of a rolling tire [11]. McAllen determined the heat generation in a transient conduction analysis, by using a phase lay between the stress and strain fields [12]. Clark and Dogde analyzed the heat generation of aircrafts tires which undergo large deformations due to high vertical loads in combination with high velocities [13]. Srirangam et al. used an innovative pavement interaction modal to iteratively determine the effect of tire operating

temperatures on hysteretic friction on tire/road interaction surface [1]. Konde et al. used temperature- dependent friction coefficient for thermo-mechanical analysis of aircraft tires [14].

In recent investigations, the transient dynamic algorithms have been used for combination of motion and stress analyses of rolling tires [15]. Goreishy proposed a mix Lagrangian/ Eulerian method for analysis of pneumatic rolling tires [16]. However, including rolling effect in simulation requires long CPU time and obtained results for transient strains have considerable oscillations which affects the accuracy of the analysis. So, in some works the centrifugal forces are added to static interaction of tire with the road for facilitating the analysis [17].

Despite the drawbacks of experimental methods, they provide very useful information for demonstrating the of numerical investigation results. The aim of current study is evaluating the temperature distribution of a 205/55R16 radial tire via a transient thermo mechanical analysis. Comparing the obtained results with the experimental measurements of Haung et al. for the same tire proves the considerable validity and accuracy of the simulation procedure [18].

### Material and methods

The rubber as main constructing part of the tire body, shows highly nonlinear stress-strain relationship and small elasticity theories using E and G moduli are not suitable for describing the response of the tires to the external loading. The models based on stored mechanical energy provides more useful means for measuring the responses [19]. In tire modeling studies, the Mooney-Rivlin model [20] is commonly used for describing the mechanical behavior of rubber structures. According to this model, the strain energy function U is expressed as:

$$U = C_{10}(\bar{I}_1 - 3) + C_{01}(\bar{I}_2 - 3) + \frac{1}{D}(J_e - 1)^2 \quad (1)$$

Where  $C_{10}$  and  $C_{01}$  are material constants, U is strain energy density,  $J_e$  is the elastic volume ratio and D introduces the compressibility.  $\bar{I}_{1,2}$  stand for first two deviatoric strain invariants defined as:

$$\begin{aligned} \bar{I}_1 &= \lambda_1^2 + \lambda_2^2 + \lambda_3^2 \\ \bar{I}_2 &= \lambda_1^2 \lambda_2^2 + \lambda_1^2 \lambda_3^2 + \lambda_2^2 \lambda_3^2 \end{aligned} \quad (2)$$

Where  $\lambda_i$  are principal stresses.

For an incompressible material such rubber, D equals to infinity [14]. The nonlinear stress-strain relationship of Mooney- Rivlin type can be expressed as:

$$\sigma = 2 \left( \lambda - \frac{1}{\lambda^2} \right) \left( C_{10} + \frac{C_{01}}{\lambda} \right) \quad (3)$$

Where  $\sigma$  is stress,  $\lambda$  is expansion ratio,  $\varepsilon$  is the strain. Based on test data, the material properties of the rubber can be determined by curve fitting [21, 22].

For the vast range of vehicles velocity, the hysteresis loss and dissipated energy is considered as the main cause of temperature rise in a tire [8]. Thus, the dissipation due to the friction will be neglected in this study. It is noted that for evaluating the temperature distribution of tires, the total stored strain energy and power loss should be computed from a nonlinear elastic finite element analysis of the inflated, loaded, rotating tire.

The tire/ground interaction analysis needs a sophisticated finite element model since contact and friction at contact surface are highly non-linear complicated problems [14]. However, according to the experimental data, for vast range of vehicles speed, the friction resulted from relative slippage has less than 5% effect on mechanical behavior of a rolling tire and can be discarded for simplification [23]. Through static analysis, the inflation pressure and axial loading are considered for obtaining the stresses and strains of the tire. The load is applied at the center of the tire and inflation pressure is then applied at the inner surface. The contact between the tire and the rim is simplified by assuming that the tire adheres to the rim. Instead of prohibitively expensive and cumbersome Lagrangian method for discretizing the motion of tire [14, 24], an approximate approach adopted in this work for including the centrifugal forces resulted from the rotation into the static analysis of tire/road interaction.

In current study, the rubber is assumed to be nonlinear isotropic and tire reinforcements as elastic orthotropic material. The numerical values of thermo- mechanical properties of the materials used in the finite element model are based on and are presented in Table 1 [25]. It is noted that for tire compounds, over the normal operating temperatures the thermal conductivity variations show a little difference [26]. So, the

thermo-mechanical properties of the tire are considered to be temperature independent.

**Table 1.** Material properties used in tire FE model [27].

	Tread	Belt	Carcass
Mooney-Rivlin constants ( $Mpa$ )			
$C_{01}$	2.0477	-	-
$C_{10}$	1.1859	-	-
Density, $\rho \left( \frac{kg}{m^3} \right)$	1140	7644	1390
Modulus of elasticity, $E \left( Gpa \right)$	-	55	0.794
Poisson ratio, $\nu$	-	0.3	0.45
Thermal conductivity, $\mu \left( \frac{W}{m^2 \cdot ^\circ C} \right)$	0.293	60.5	0.293
Hysteresis constant, $H$	0.1	-	-



**Fig1.** The radio transmitter system placed inside the tire which is presented by Huang et al.[18].



**Fig2.** The System Mounting inside the [18].

As mentioned earlier, hysteresis loss is assumed to completely contribute to internal heat generation. The work presented here uses a hysteresis value reported in Ref. [8]. Knowing the rotational speed of the tire, the heat generation rate can be calculated from the following equation:

$$\dot{q}_v = H \times U_{tot} \times \frac{V_L}{2\pi R_t} \quad (4)$$

Where  $H$  is hysteresis value,  $U_{tot}$  is total stored strain energy density,  $V_L$  is the speed and  $R_t$  is the effective radius of the rolling tire. The general governing equation of heat transfer in polar coordinates, is expressed as:

$$\frac{1}{r} \frac{\partial}{\partial r} \left( kr \frac{\partial T}{\partial r} \right) + \frac{1}{r^2} \frac{\partial}{\partial \theta} \left( kr \frac{\partial T}{\partial \theta} \right) + \frac{\partial}{\partial z} \left( k \frac{\partial T}{\partial z} \right) + \dot{q}_v = \rho C \frac{\partial T}{\partial t} \quad (5)$$

Where  $T$  is temperature,  $k$  is thermal conductivity and  $\dot{q}_v$  is the heat generation per unit volume resulted from internal hysteresis. Solving this equation one can easily found the time variation of temperature inside the tire domain.

The validity of obtained results is validated by wireless measurement system which is presented by Huang et al. [18]. A set of transmitter and receiver system (shown in Fig. 1) is also used to overcome the difficulty of data transmission in a rolling tire. The type of the transmitter & receiver unit is Freescale MC13213. The thermistor links to the transmitter by wires, its resistance variations are converted to the digital voltage signals by the circuit and A/D converter on the transmitter board, and then the signals are communicated from the transmitter to the receiver through radio at 2.4 GHz. The whole measurement system was calibrated with a Laboratory Thermostatic Water Bath.

As illustrated in Fig. 2, six thermistors are fixed at one side of the inner wall by using the tire repair glue. In this system, negative temperature coefficient thermistor of MF52 with a bead diameter of 2 mm is selected as temperature sensor. The low size and cost of these sensors makes them affordable for experimental analysis. In order to solve the unexpected sensor failure problem, two groups of thermistors, as substitutes for one another based on the hypothesis of uniform circumferential temperature, were placed at the same positions on two tire cross sections. The temperature distributions of both the inner and the outer wall in the rolling tire can be measured. As long as each one of the two transmitters is

functional throughout the testing, the target temperature can be surely obtained.

## Results and discussions

In this work, the results of a 3D finite element for interaction of 205/55R16 steel belted radial tire are used for obtaining the strain energy loss and resultant heat generation rate inside the tire body. However, as the first step for the thermal investigation, it is assumed that the whole tire is made by the rubber and the thermal properties such as conductivity and specific heat have minimal variation within operating conditions.

The flow of air around the outer surfaces of the tire causes forced heat transfer convection which its coefficient  $h_c$  can be found according to the following empirical equation [26]:

$$h_c = 5.9 + 3.7V \quad (6)$$

Where,  $V$  is the velocity of the air. The airflow speed has a single value over the tread blocks of the tire. However, for the sidewalls, this speed is a function of linear distance from the hub center. For these surfaces, the average velocity is substituted in equation. The convection coefficient at the inner surfaces is

taken to be  $5.9 \frac{W}{m^2 \cdot ^\circ C}$  due to negligible relative velocity of the inside inflated air [25].

For five time steps, the temperature gradients in tire domain are represented in Fig. 3. The ambient temperature and translational velocity of the rolling tire are selected as 29°C and 60 km/hr. It can be observed that lower temperature fields occur at the bead and exposed regions and hottest points inside the tire body are located on the inner surface. For all the cases shown in Fig. 3, axisymmetrical distribution can be observed for transient temperature gradient. According to the obtained results, after 40 min of revolution, the temperature gradient reaches to its steady state value and after that no considerable variation occurs unless the operating conditions change.

For 50 min transient heat transfer, the distributions of the temperature on outer surfaces of tread blocks are shown in Fig 4. The considered points are along curved line B-B shown in Fig. 6. It is evident that in early stages of the analysis, the temperature gradient has a very fast increase rate and after that converges to its steady state values.

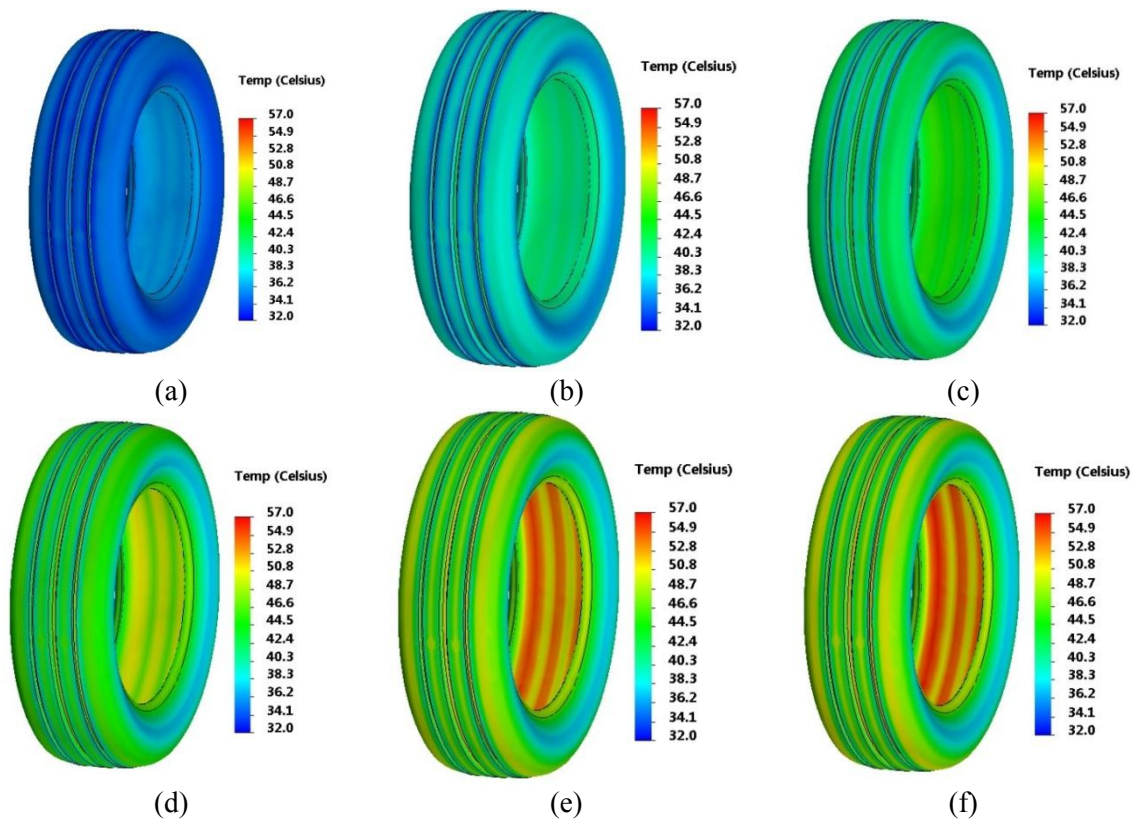


Fig3. The temperature gradient of tire after rolling motion of: a) 2 min, b) 4 min, c) 10 min, d) 20 min, e) 30 min and f) 40 min

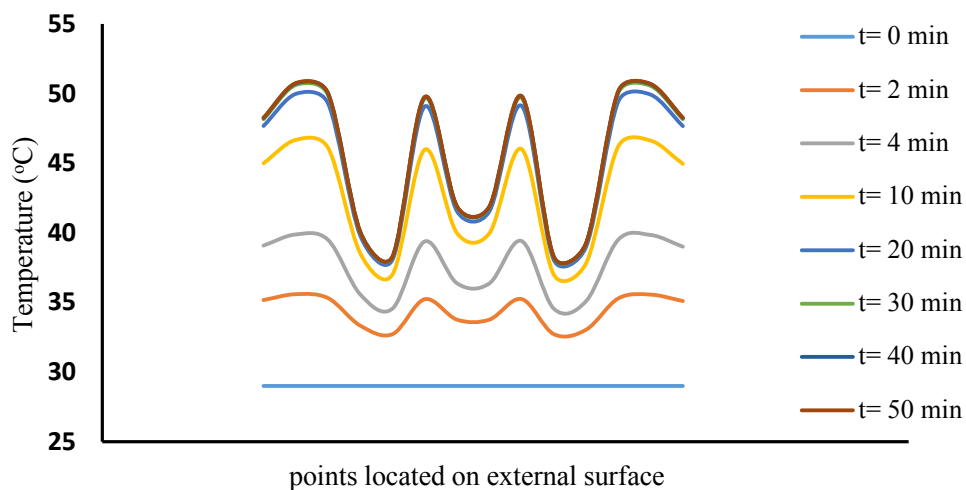


Fig4. Variation of temperature outer wall and tread blocks along the curve line B-B in the rolling tire

Fig. 5 shows the time variation of temperature distribution for inside surface of the tire. Due to the poor thermal convection, the points located on inner side have relatively

higher temperature rather than the points located on the tread side. Furthermore, the analysis shows that the temperatures on the middle of both inner and outer surfaces are higher than the

corners. This can be explained by the fact that the points located on the corner edges have two sides with the surrounding air. So, more effective heat exchange occurs in this region. For estimating the accuracy and acceptability of the results, they are compared with the experimental data obtained from the sensors attached to the same rolling pneumatic tire (see Fig. 6.). The experimental conditions are as follows:

Initial inflation pressure: 250 kPa, Load: 5000 N, Speed: 60 km/h, Ambient temperature: 302 K.

For the specified points located on inside surface of tire, the results of numerical analysis are compared with the empirical data of the sensors. The locations of three considered

sensors  $S_1$ ,  $S_2$ , and  $S_3$  are schematically shown on Fig. 6. For the tread side points, the numerical and experimental results are again compared in Figs. 7 for three specified sensors. A great agreement can be easily observed which proves the accuracy of the finite element analysis. It can be seen that the finite element analysis forecast accurately the steady state temperature of the specified point. However, the numerical analysis has a relatively slight error in prediction of temperature increase rate during the transient part heat transfer. This can be explained by the fact that the value of transient and steady state heat transfer coefficients are considered to be the same in this study. This assumption leads to some inaccuracy in obtained results.

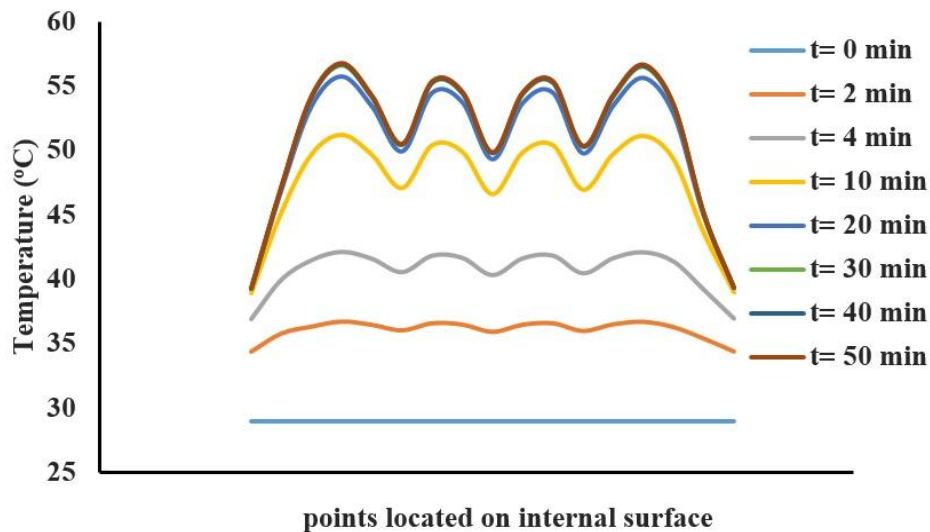


Fig5. Variation of temperature for inner wall along the curve line A-A in the rolling tire

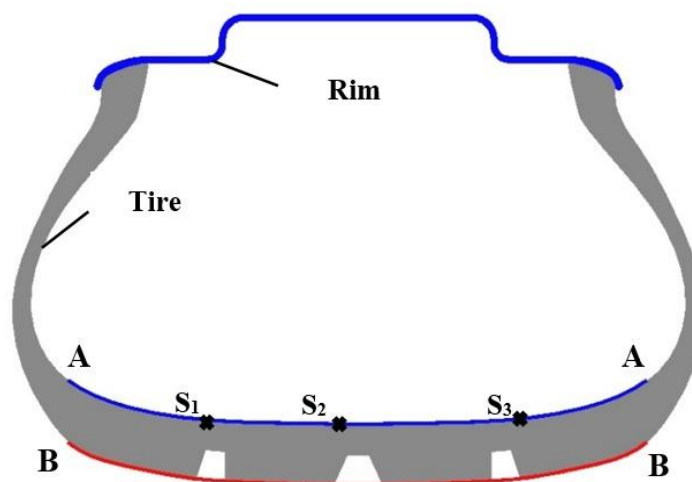
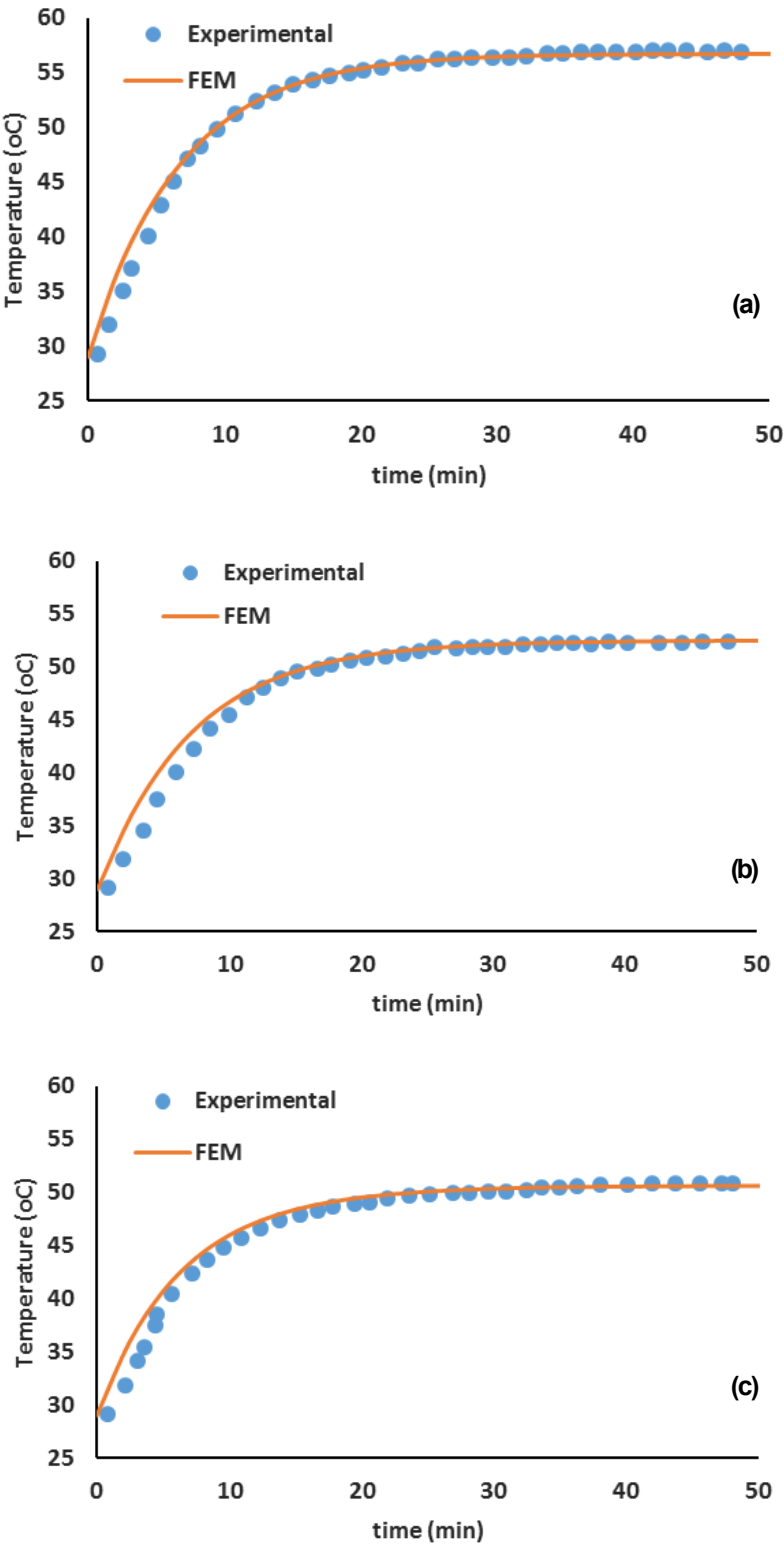


Fig6. The location of sensors and the curved lines A-A and B-B considered for the tire.



**Fig7.**Comparing the results of transient FE analysis and the sensors placed on inner wall of the rolling tire: a) for sensor  $S_1$ , b) for sensor  $S_2$ , c) for sensor  $S_3$ , respectively

## Conclusion

In presented study, an efficient numerical analysis based on finite element method is conducted for evaluating the transient temperature increase in a rolling tire. Instead of costly Lagrangian scheme, the dissipated strain energy and heat generation rate of the rolling tire is evaluated by enforcing the centrifugal forces on static analysis of tire/road interaction. It is noted that at each time step of the analysis, the temperature increases axisymmetrically inside the tire body. For translational velocity of 60 km/hr, about 40 min is needed to reach the steady state thermal conditions for the tire. Because of low thermal conductivity and poor heat exchange rate, the inner parts of the tire become hotter regions than the exposed areas. It is noted from the results that for considered operating conditions, the maximum increase developed inside the tire is found to be 20oC after reaching the temperature gradient into its steady state condition. For estimating the accuracy and acceptability of the results, they are compared with the experimental data obtained for the same pneumatic tire. According to the results, the proposed analysis has a considerable consistency and accuracy.

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