ANALYSIS OF FORCED CONVECTION IN A DUCT WITH OBSTACLES

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Abstract. *The purpose of this work is the study of the numerical and experimental viscous incompressible steady state flow with heat transfer into a narrow channel containing rectangular array of obstacles. The experimental approach involves the determination of the heat transfer coefficient through the temperature measurement in thermal wake and the results were compared with numerical results. For the numerical analysis it was used the computational program of fluid mechanics and heat transfer named ICEPAK. It was confirmed that in the last lines of the array the heat transfer was more pronounced due to forced convection. It was determined the average heat transfer coefficients (with uncertainties between 6% and 15%) and it was observed that the effect of the position decreases as flow speed increases. It was also concluded that both turbulence models used were unable to predict the effect of the position properly. However, the k epsilon RNG model showed better behavior resulting in temperature intermediaries values compared with experimental results.*

Keywords*: Average heat transfer coefficient, Average Nusselt number, forced convection, turbulence models*

1. INTRODUCTION

The great development of the electronics equipment in recent decades caused a significant increase in processing capacity and also marked a decrease in the size of electronic components and processors. This aspect technology is known as an increase in electronic packaging. In contrast to this development it was occurred the reduction of areas available for heat dissipation.

When it was considered the capacity to solve problems through the fluids mechanics, heat transfer and computational concepts, the analysis electronic cooling systems should not be so challenging. However, to predict if an electronic component will operate below of the operating temperature maximum, it is necessary to know the thermal power dissipated. The knowledge of this power is a barrier, because the manufacturers generally know only the power maximum of each component and or the total electrical power of the plate.

The maximum power dissipated by components led oversized projects that require cooling process impracticable for air forced convection, due to small area for heat transfer. Besides, it is necessary to maintain the air flow at speeds within the limits imposed by the electronics in order to avoid both electrostatic and acoustic problems.

The determination of the dissipated power in each component could be made by cooling Newton´s law. Experiments measure the components and air temperatures and then raise correlations to determine the heat transfer coefficients.

Cabinets in data center and telecommunications companies contain several drawers with dozens of electronic board juxtaposed. In these systems the forced convection cooling is still the best cost benefit relation. The design of a cooling system relies on numerical simulations due to difficult in experimental analysis in development stage. However, to perform the numerical simulations is still necessary to know the dissipated power by the components. Dissipated power experimental measures are necessary to feed the numerical simulation.

Thus the study of air flow in narrow channels is of great interest to electronics industry, especially by the fact that forced convection is still the most widely used.

The aim of this study is to analyze numerically and experimentally the forced convection into a narrow channel contained obstacles with dissipated power.

The heat transfer convection coefficient can serve as reference in the determination of the dissipated power by electronic components.

2. EXPERIMENTAL METODOLOGY

Figure 1 shows a schematic of experimental apparatus. The test section is 508 mm long by 254 mm wide with height channel of 12.7 mm. Air enter in the test section, passes through a plenum chamber and then is measured through one Pitot tube used in determining the overall flow rate. This is followed by a flow control valve and a centrifugal blower which then exhausted flow from the laboratory.

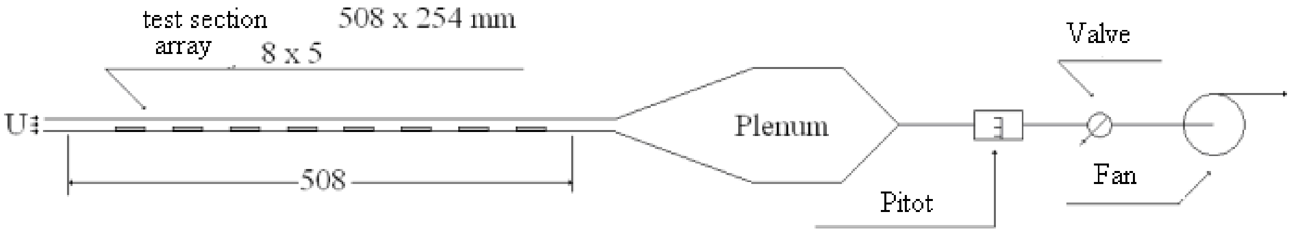

Figure 1. Experimental Apparatus

The figure 2 shows the channel floor and the upper wall consisted of a 7.8 mm commercial plexiglass.

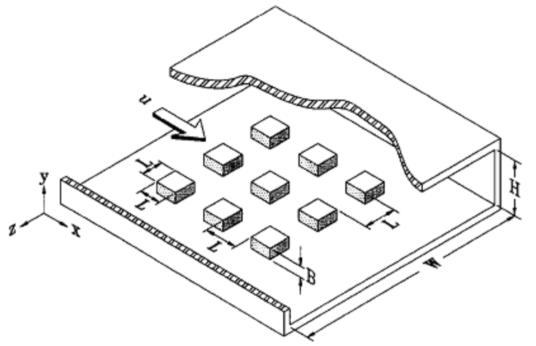

Figure 2. Nomenclature of Experimental Apparatus

In the figure 3 it is possible to see the top view of the test section. The component array is mounted in the bottom of the channel. The material of the components is pure aluminum, which has 25.4 mm length (L), 25.4 mm width (L) and 6.35 mm high (B).

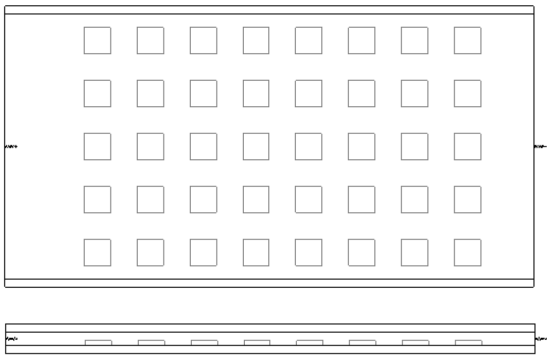

Figure 3. Views of top and side of the test section

One obstacle in the center column was instrumented with a resistance heating element (active obstacle). This component was also instrumented with four calibrated thermocouples which were located at the top in the corners. All the downstream components in this center column were instrumented with calibrated thermocouples in order to measure the thermal wake effect of the dissipation from the heated component.

We put the active obstacle at the first line in the center column; then we connected it with power supply through bottom wall of the channel; the thermocouples also came in by this the same wall. We turned on the exhaust and adjusted the valve to obtain an average speed of 1.97 m/s at test section. Then we adjusted the power supply to 3.68 W. It was monitored the temperature for 40 minutes and so we recorded them. This procedure was repeated twice more for this position of the active obstacle.

The whole procedure of the previous paragraph was repeated by changing the speed to 2.99 m/s and 4.04 m/s. After this step we put the active component in the second row and we restarted de procedure; so we proceeded until the active obstacle occupied the penultimate line of the arrangement in the center column.

3. MATHEMATICAL MODELING

The Joule´s law relates the electric current flowing through a conductor with the dissipated thermal energy due to resistance as:

$$P = I^2 R \tag{1}$$

The equation (1) gives the heat rate that is dissipated through the filament. One part is transmitted to aluminum block and the remainder is transmitted to terminals of DC source. The energy transmitted to the terminal is calculated considering the cables as fins endless. Considering an energy balance for a differential element of this system, cross section constant and integrating, dissipated power obtained for two cables is:

$$q_d = 2\sqrt{h_n P k_c A_c \left(T_c - T_\infty\right)} \tag{2}$$

Where hn is the free convection coefficient (W/m2K), P is the perimeter (m), kc is the thermal conductivity (W/Mk) of the copper, Ac is cross seccion area of cable (m2), Tc is the cable temperature and Tinf is the ambient temperature.
Thus, the dissipated power is calculated as:

$$q = P - q_d \tag{3}$$

The overall analysis of energy equation of active element in steady state subject to three process of heat transfer (convection, radiation and conduction) leads to the following equation:

$$q = q_{cond} + q_{conv} + q_{rad} \tag{4}$$

The rate of heat transfer by convection can be easily determined rearranging the equation (4):

$$q_{conv} = q - q_{cond} - \varepsilon\sigma A_{rad}\left(T_s^{\,4} - T_\infty^{\,4}\right) \tag{5}$$

The average convection heat transfer coefficient is the calculated as follow:

$$h = \frac{q_{conv}}{A_{conv}\left(T_s - T_\infty\right)} \tag{6}$$

Where A_{conv} is the area of convection heat transfer and T_s is surface temperature and T_∞ is fluid temperature.

3. RESULTS

In this section it will be presented the experimental results for the test section, including the temperature distribution, the heat transfer coefficients, Nussel number and the results of uncertainties analysis for all quantities analyzed.
Arvizu and Moffat (1982) showed that the temperature rise of components downstream of a heat dissipating component could be correlated in terms of flow conditions and component location relative to heated component. They then were able to suggest a calculation procedure based on superposition concepts whereby the individual temperatures of all components in an array could be estimated.
The figure 4 shows the channel arrangement and the position of the obstacles. Only components of the central column were numbered and their temperatures were monitored during each experiment when only one component dissipated thermal energy. The objective is to understand the influence of the position within the arrangement in the determination of cooling and heating of the obstacles that are downstream of active component.

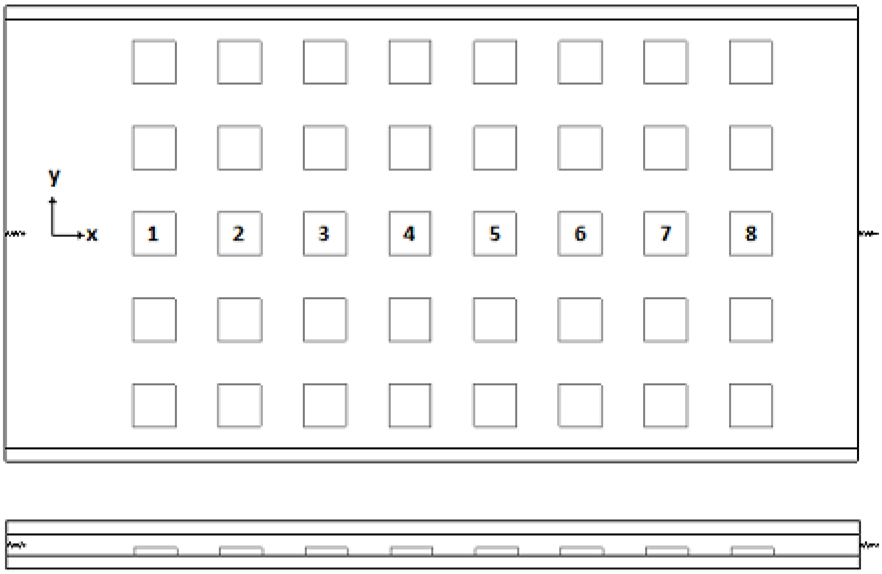

Figure 4. The obstacles arrangement

It was imposed three velocities at the channel entrance. Reynolds number of 3150 at 1.97 m/s; Reynolds number of 4780 at 2.99 m/s and Reynolds number of 6460 at 4.07 m/s.

In the figure 4 it was observed that the active component occupied seven positions, each position in abscissa corresponds to row that the active component occupied. The active component dissipated 3.68 W (calculated from equation 5). Each point on the curve of the figure 4 represents the result of a single test, because there was only one active component per test. Thus, the figure 5 shows the average temperature of the active component according to its position in the array for each velocity tested.

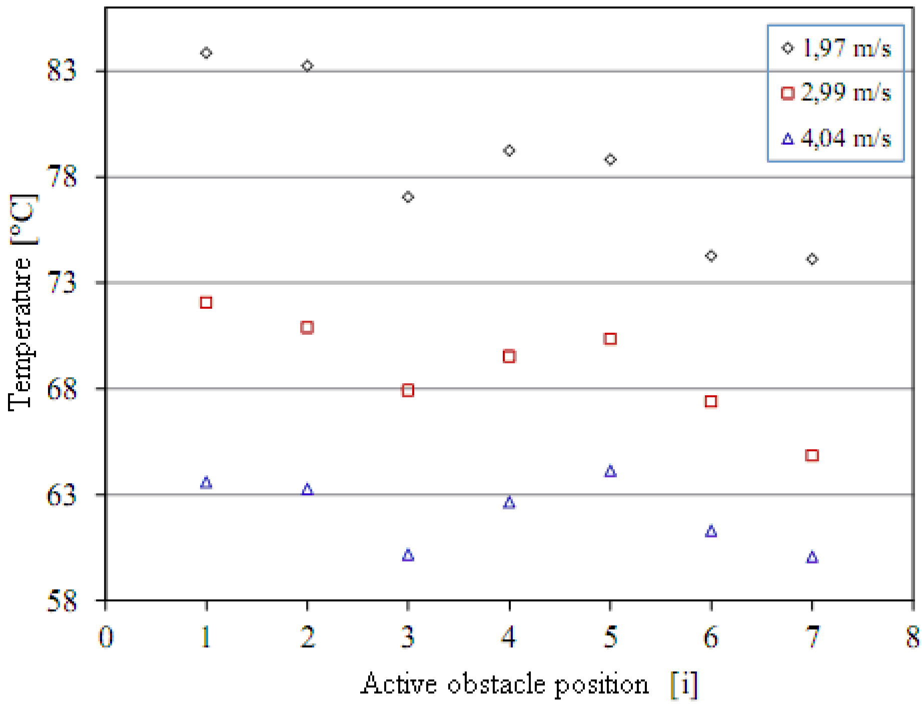

Figure 5. Average temperature of the active component as a function of position

When the active component occupies a new position in central column the active component temperature decreases due to the increase of turbulence effects, although only two regions of the faces of each obstacle were subjected to recirculation effects.

The figure 6 is a comparison between the results of this work and the developed by Wirtz and Dykshoorn (1984), when the active obstacle was in second row. In general the two results were similar; the active obstacles had an increase similar of temperature with variation between 0 and 1.5 C. It believes its variation is due to time established to reach the steady state.

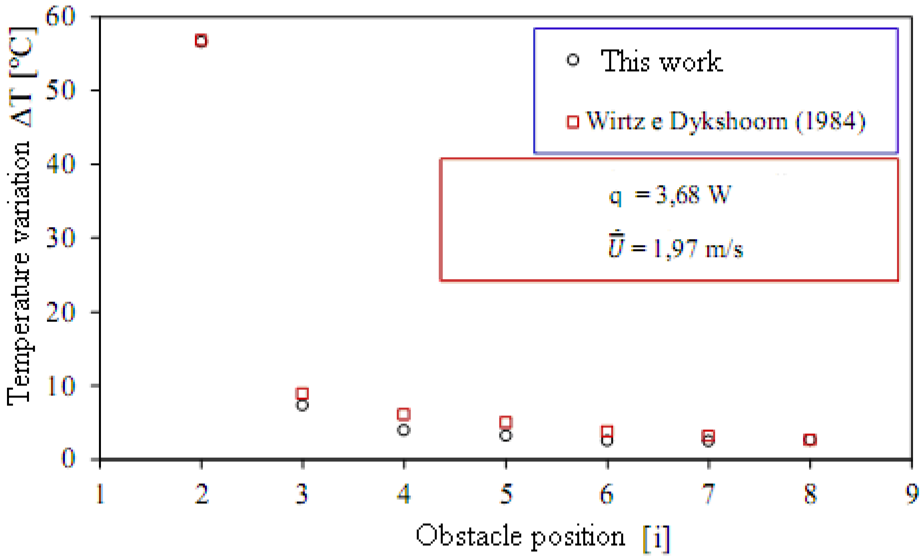

Figure 6. Comparison of results

In their studies of convection from cubicle elements, Arvizu and Moffat (1982) found that the component Nusselt number could be correlated in terms of Reynolds number based on array velocity, where the array velocity is the velocity which is characteristic of the flow in the space between components. The array velocity may be determined by running experiments with the channel wall resting on top of the components so that all coolant flow is through the array. The measured array velocity may then be correlated in terms of pressure drop across one or several rows of components. This allow us to compute a Reynolds number based on array velocity and correlate the dimensionless heat transfer coefficient in terms of this Reynolds number.

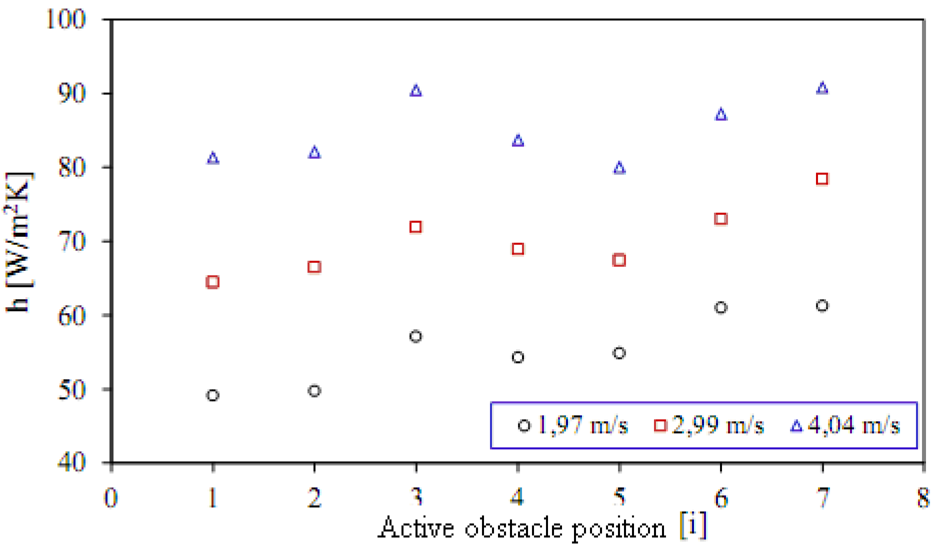

Figure 7. Convective heat transfer coefficients for each component

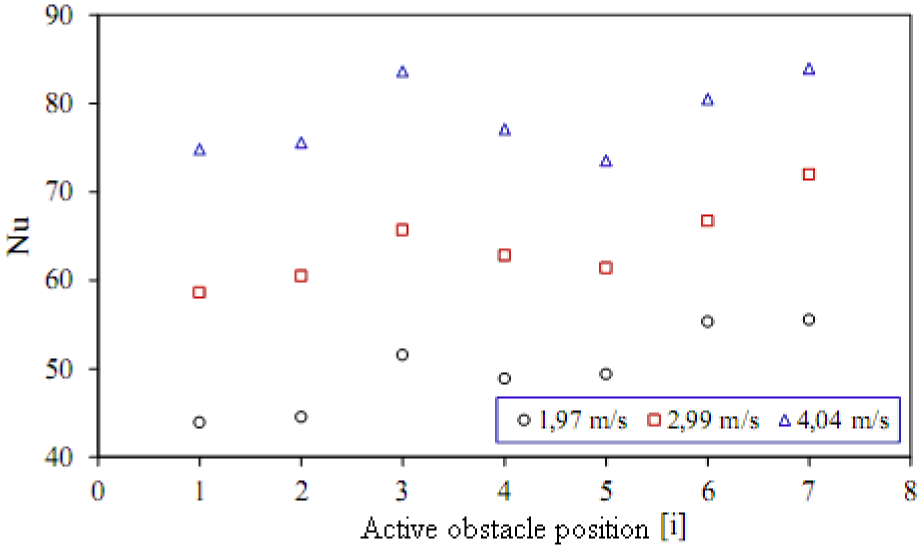

Figure 8. Nussel number for each component

The characteristic length used to determine the average Nussel number from convective heat transfer coefficient was the length of the sides of obstacles (L). The figure 9 and figure 10 show the uncertainties of the average convective heat transfer coefficient and the uncertainties of the average Nusselt number, respectively. The greatest uncertainties for velocity of 1.97 m/s were approximately 14% for Nusselt number and for 2.99 m/s and 4 m/s the uncertainties were around 11 and 15% for Nusselt number, respectively.

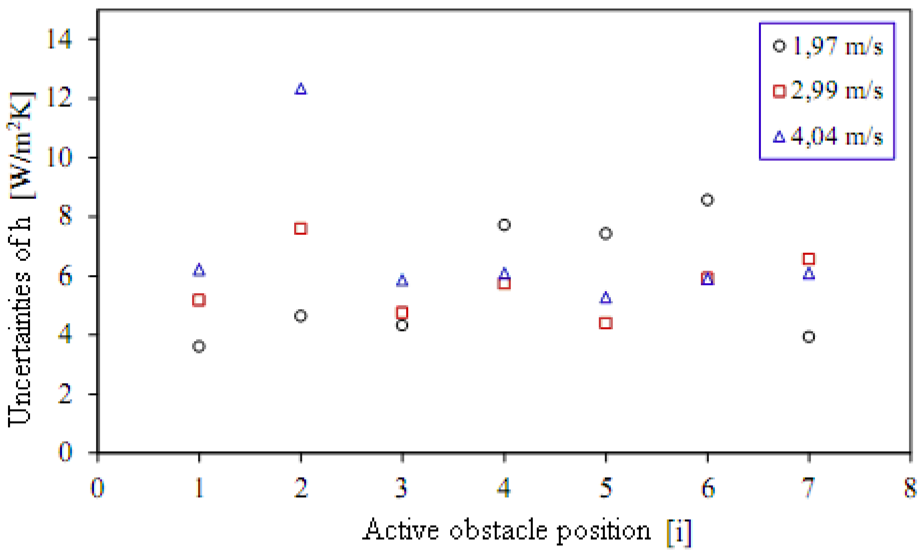

Figure 9. Uncertainties for Nussel number for each component

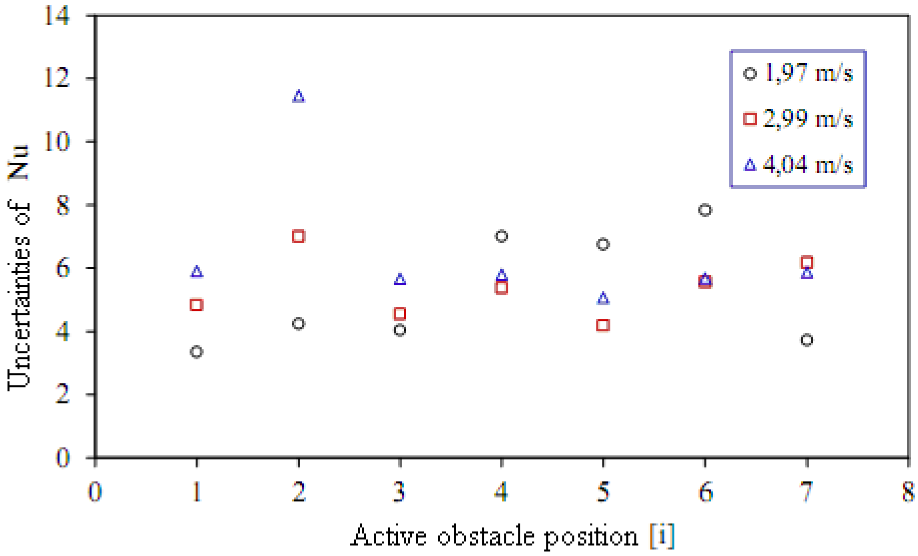

Figure 10. Uncertainties for Nussel number for each component

The equations 7, 8, 9, 10, 11, 12 and 13 show average Nusselt number correlations obtained for active obstacles in the positions 1, 2, 3, 4, 5, 6 and 7, respectively. The uncertainties these correlations are between 6% and 15%. The correlations are applicable for Reynolds number between 3000 and 6500. The Reynolds number is based on characteristic length L.

$$Nu_L = 14.4 + \frac{9.3\,\mathrm{Re}_L}{1000} \tag{7}$$

$$Nu_L = 15.3 + \frac{9.4\,\mathrm{Re}_L}{1000} \tag{8}$$

$$Nu_L = 20.5 + \frac{9.7\,\mathrm{Re}_L}{1000} \tag{9}$$

$$Nu_L = 22.1 + \frac{8.5\,\mathrm{Re}_L}{1000} \tag{10}$$

$$Nu_L = 26.4 + \frac{7.3\,\mathrm{Re}_L}{1000} \tag{11}$$

$$Nu_L = 31 + \frac{7.6\,\mathrm{Re}_L}{1000} \tag{12}$$

$$Nu_L = 29.3 + \frac{8.6\,\mathrm{Re}_L}{1000} \tag{13}$$

The correlations obtained for all active obstacles are linear equations and the Reynolds number are based on average velocities in the channel inlet.

3. CONCLUSIONS

It was studied the effect of position on the temperature and heat transfer due to turbulence in an array of obstacles within a channel subject to an arrow air flow. The channel entrance was long enough to assure developed turbulent flow.

It was determined experimentally the temperature distribution into the arrangement of obstacles. Besides, it was compared the temperature distribution results with the literature there was good agreement between results.

It was proved experimentally that there is a tendency of a higher heat transfer by forced convection for active components more inside of the arrangement. Despite, high levels of uncertainties for convective heat transfer coefficients (between 6 and 15%) show that the effect of the position decreases as velocities increases.

4. ACKNOWLEDGEMENTS

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5. REFERENCES

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