E-ISSN: 2321–9637
Volume 2, Issue 1, January 2014
International Journal of Research in Advent Technology
Available Online at: http://www.ijrat.org

EXPERIMENTAL ANALYSIS OF TURBULENT FLOW HEAT TRANSFER IN A RECTANGULAR DUCT WITH AND WITHOUT CONTINUOUS AND DISCRETE V-SHAPED INTERNAL RIBS

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ABSTARCT:
In this present work, Experiments were performed to collect heat transfer and friction data for forced convection flow of air in rectangular duct with and without internal ribs. The analysis was conducted within the range of Reynolds number from 3000 to 18000. The horizontal rectangular duct was subjected to constant and uniform heat flux. Experimental results for this configuration are reported for three different channels viz., smooth rectangular duct, rectangular duct with continuous V- shaped ribs and rectangular duct with discrete V- shaped ribs. The effects of internal ribs on the heat transfer coefficient and friction factor are compared with the result of smooth duct under similar flow conditions. Experimental results show that the local Nusselt number distribution is strongly depended on the position, orientation, and geometry of the ribs. The friction factor ratio goes up with an increase in the Reynolds number, but its value depends on the arrangement of ribs. The results also show that the discrete V- shaped ribs produce overall less heat transfer enhancement than the continuous V- shaped ribs. However, the increased heat transfer enhancement in the continuous V- shaped ribs came at the cost of an increased pressure penalty.

Keywords: *Heat transfer augmentation; Forced convection heat transfer; Internal ribs; Rectangular duct.*

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1. INTRODUCTION

Heat exchangers have been widely employed in several industrial and engineering applications. The Techniques for enhancing heat transfer are relevant to several engineering applications. In recent years, the high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment. The heat transfer rate can be improved by introducing a disturbance in the fluid flow (breaking the viscous and thermal boundary layers), but in the process, pumping power may increase significantly and ultimately the pumping cost becomes high. Therefore, to achieve a desired heat transfer rate in an existing heat exchanger at an economic pumping power, several techniques have been proposed in recent years. The great attempt on utilizing different methods is to increase the heat transfer rate through the compulsory force convection. The meanwhile, it is found that this way can reduce the sizes of the heat exchanger device and save up the energy. M. Sozen and T.M. Kuzay [1] numerically studied the enhanced heat transfer in round tubes filled with rolled copper mesh at Reynolds number range of 5,000-19,000. With water as the energy transport fluid and the tube being subjected to uniform heat flux, they reported up to ten fold increase in heat transfer coefficient with brazed porous inserts relative to plain tube at the expense of highly increased pressure drop. Q. Liao and M.D. Xin [2] carried out experiments to study the heat transfer and friction characteristics for water, ethylene glycol and ISOVG46 turbine oil flowing inside four tubes with three dimensional internal extended surfaces and copper continuous or segmented twisted tape inserts within Prandtl number range from 5.5 to 590 and Reynolds numbers from 80 to 50,000. They found that for laminar flow of VG46 turbine oil, the average

Stanton number could be enhanced up to 5.8 times with friction factor increase of 6.5 fold compared to plain tube. D. Angirasa [3] performed experiments that proved augmentation of heat transfer by using metallic fibrous materials with two different porosities namely 97% and 93%. The experiments were carried out for different Reynolds numbers (17,000-29,000) and power inputs (3.7 and 9.2 W). The improvement in the average Nusselt number was about 3-6 times in comparison with the case when no porous material was used. Fu et al. [4] experimentally demonstrated that a channel filled with high conductivity porous material subjected to oscillating flow is a new and effective method of cooling electronic devices. The experimental investigations of Hsieh and Liu [5] reported that Nusselt numbers were between four and two times the bare values at low Re and high Re respectively. Bogdan and Abdulmajeed et al. [6] numerically investigated the effect of metallic porous materials, inserted in a pipe, on the rate of heat transfer. The pipe was subjected to a constant and uniform heat flux. The effects of porosity, porous material diameter and thermal conductivity as well as Reynolds number on the heat transfer rate and pressure drop were investigated. The results were compared with the clear flow case where no porous material was used. The results obtained lead to the conclusion that higher heat transfer rates can be achieved using porous inserts at the expense of a reasonable pressure drop. Smith et. al. [7] investigated the heat transfer enhancement and pressure loss by insertion of single twisted tape, full length dual and regularly spaced dual twisted tapes as swirl generators in round tube under axially uniform wall heat flux conditions. Chinaruk Thianpong et.al. [8] Experimentally investigated the friction and compound heat transfer behavior in dimpled tube fitted with twisted tape swirl generator for a fully developed flow for Reynolds number in the range of 12000 to 44000. Whitham [9] studied heat transfer enhancement by means of a twisted tape insert way back at the end of the nineteenth century. Date and Singham [10] numerically investigated heat transfer enhancement in laminar, viscous liquid flows in a tube with a uniform heat flux boundary condition. They idealized the flow conditions by assuming zero tape thickness, but the twist and fin effects of the twisted tape were included in their analysis. Saha et al. [11] have shown that, for a constant heat flux boundary condition, regularly spaced twisted tape elements do not perform better than full-length twisted tape because the swirl breaks down in-between the spacing of a regularly twisted tape. Rao and Sastri [12], while working with a rotating tube with a twisted tape insert, observed that the enhancement of heat transfer offsets the rise in the friction factor owing to rotation. Sivashanmugam and Sundaram [13] and Agarwal and Rao [14] studied the thermohydraulic characteristics of tape-generated swirl flow. Peterson et al. [15] experimented with high-pressure (8–16 MPa) water as the test liquid in turbulent flow with low heat fluxes and low wall–fluid temperature differences typical of a liquid–liquid heat exchanger.

The present experimental study investigates the increase in the heat transfer rate in a rectangular duct heated with a constant uniform heat flux with air flowing inside it using continuous and discrete V-shaped internal ribs of same geometrical configuration. The present work has been carried out with turbulent flow (Re number range of 3,000-18,000) as most of the flow problems in industrial heat exchangers involve turbulent flow region.

2. EXPERIMENTAL WORK

2.1. *Experimental Setup*

The apparatus consists of a blower unit fitted with a pipe, which is connected to the test section located in horizontal orientation. Nichrome bend heater encloses the test section to a length of 50 cm. Four thermocouples T2, T3 and T4 at a distance of 11 cm, 22 cm, 33 cm and 44 cm from the origin of the heating zone are embedded on the walls of the tube and two thermocouples are placed in the air stream, one at the entrance (T1) and the other at the exit (T5) of the test section to measure the temperature of flowing air as shown in Fig. 1.

The pipe system consists of a valve, which controls the airflow rate through it and an orifice meter to find the volume flow rate of air through the system. The diameter of the orifice is 1.4 cm and coefficient of discharge is 0.64. The two pressure tapings of the orifice meter are connected to a water U-tube manometer to indicate the pressure difference between them. Input to heater is given through dimmer stat. Display unit consists of

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voltmeter, ammeter and temperature indicator. The circuit was designed for a load voltage of 0-220 V; with a maximum current of 10 A. Difference in the levels of manometer fluid represents the variations in the flow rate of air. The velocity of airflow in the tube is measured with the help of orifice plate and the water manometer fitted on board.

2.2 ***Procedure***

Air was made to flow though the test duct by means of blower motor. A heat input of 60 W was given to the nichrome heating wire wound on the test duct by adjusting the dimmer stat. The test duct was insulated in order to avoid the loss of heat energy to the surrounding. Thermocouples 2 to 4 were fixed on the test surface and thermocouples 1 and 5 were fixed inside the pipe. The readings of the thermocouples were observed every 5 minutes until the steady state condition was achieved. Under steady state condition, the readings of all the five thermocouples were recorded. The experiments were repeated for different channels viz., rectangular duct with continuous V- shaped ribs and rectangular duct with discrete V- shaped ribs. The fluid properties were calculated as the average between the inlet and the outlet bulk temperature. Experiments were carried out at constant heat input and constant mass flow rate, for all the three test ducts.

Fig. 1 Experimental Set-up

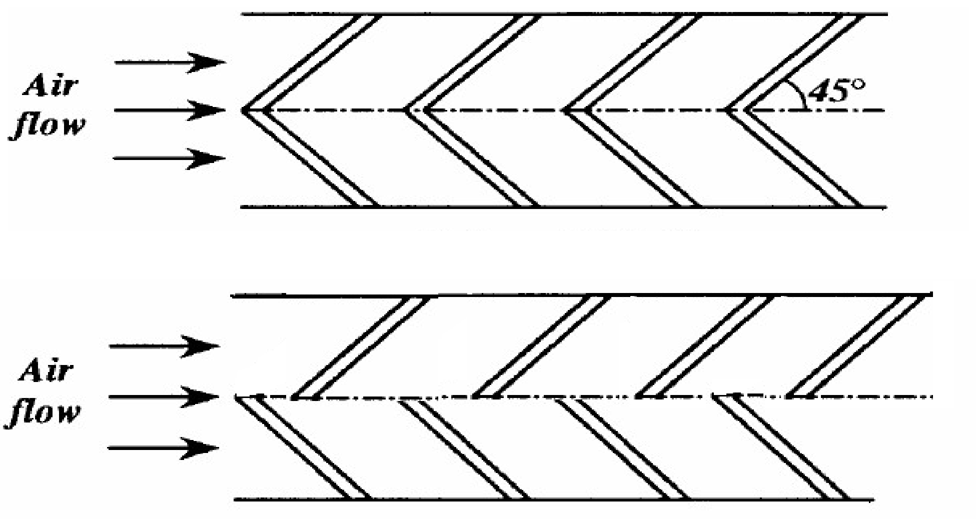

Fig.2 Continuous and Discrete internal V-shaped ribs Configurations on upper and lower surface of rectangular duct

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3. DATA REDUCTION

The data reduction of the measured results is summarized in the following procedures:

The heat transfer coefficient for the "i" wall segment was calculated using the net heat input to the wall; the wall temperature and the bulk mean temperature as

$$h_i = q_i / A_{s,i} (T_{w,i} - T_{b,i}) \tag{1}$$

The actual heat going into the duct is given by the amount of power supplied which is the product of voltage applied and the current flowing in the circuit minus the amount of heat loss from each plate. The flexible heaters were assumed to supply uniform heat to all the test ducts. The heat loss from the test section to the surrounding was calculated experimentally by conducting a no-flow condition. The heat loss to the surrounding was found to vary between 5-15% for various Reynolds number. The major heat loss was found to occur through the thick wired copper thermocouples installed in each wall segment.

The Nusselt number was normalized using the Nusselt number correlation for a fully developed flow in a smooth duct. The relative uncertainty on Nusselt number was found to be 5%.

$$Nu / Nu_o = (hD_h / K) [0.023 \, Re^{0.8} \, Pr^{0.4}] \tag{2}$$

All the temperatures were measured using the thermocouples. The bulk mean air temperature at the inlet and the exit was measured using the thermocouples and the bulk mean temperature at the end of each test duct was calculated using the formula given by

$$T_{b,out} = T_{b,in} + q / (m \, C_p) \tag{3}$$

The pressure drop measured with a micro-manometer across the test section, was used to calculate the friction factor. The relation between pressure drop and friction factor is given by

$$\Delta P_{Friction} = \frac{f . L_{Pipe}}{d_{Pipe}} \frac{\rho . u^2}{2} \tag{4}$$

The fiction factor is normalized by Gnielinski equation

$$f_o = [0.79 \ln (Re_{Dh}) - 1.64]^{-2} \tag{5}$$

The correlation for thermal performance is given by

$$TP = (Nu / Nu_o) / (f/f_o)^{1/3} \tag{6}$$

4. RESULT AND DISCUSSION

Heat Transfer Distribution for smooth rectangular duct, rectangular duct with continuous V- shaped ribs and rectangular duct with discrete V- shaped ribs:

The Nusselt number was found to increase with increasing Reynolds number. The reason attributed to this behavior is as Reynolds number increases the turbulent mixing is enhanced in the channel, which leads to effective removal of heat. However, the Nusselt number ratio, which is obtained by normalizing the average Nusselt number with the fully developed turbulent flow in smooth test tube correlated by Dittus Boelter, was found to decrease as shown in Fig 3.

In the case of smooth test duct the variation of Nusselt number ratio with increasing Reynolds number is very small as there is no flow separation and reattachments in this case and so Reynolds number does not have any effect on the heat transfer augmentation. Therefore, the Nusselt number ratio for all four Reynolds number follow a similar trend. The Nusselt number ratio was found to be highest for the lowest Reynolds number and lowest for the highest Reynolds number. It was found to vary between 2.15 and 1.1 along the axial direction.

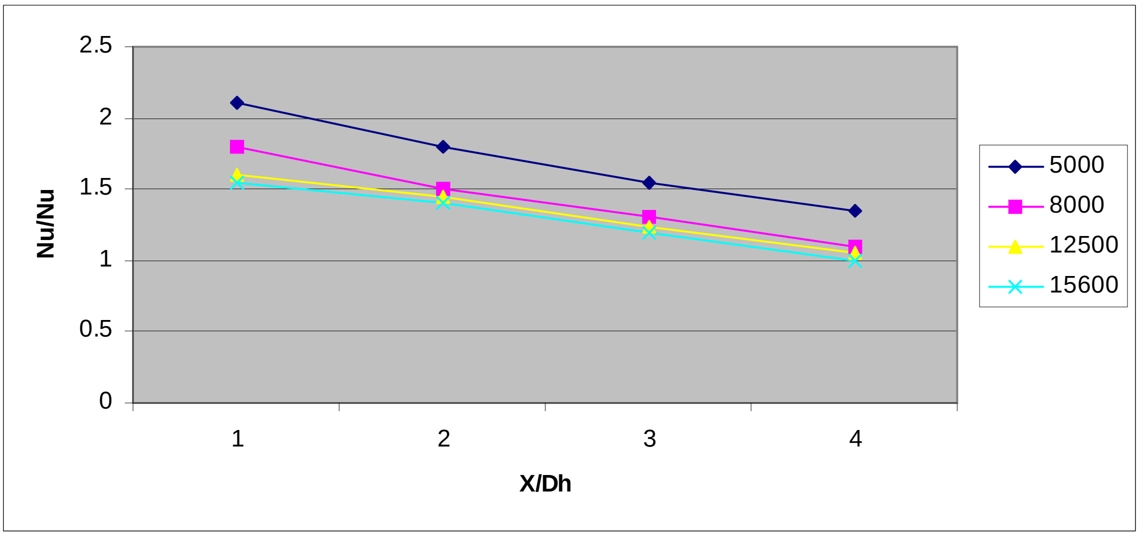

Fig.3 Effect of Reynolds Number on Heat Transfer Distribution for Smooth Rectangular duct

The heat transfer augmentation for rectangular duct with continuous V- shaped ribs was found to increase by factor of 1.7 to 2.1 in comparison to the smooth rectangular duct and by a factor of 1.06 to 1.20 in comparison to the rectangular duct with discrete V- shaped ribs. The increase in heat transfer coefficient is attributed to the induction of cross-stream secondary flow, which results in better turbulent mixing.

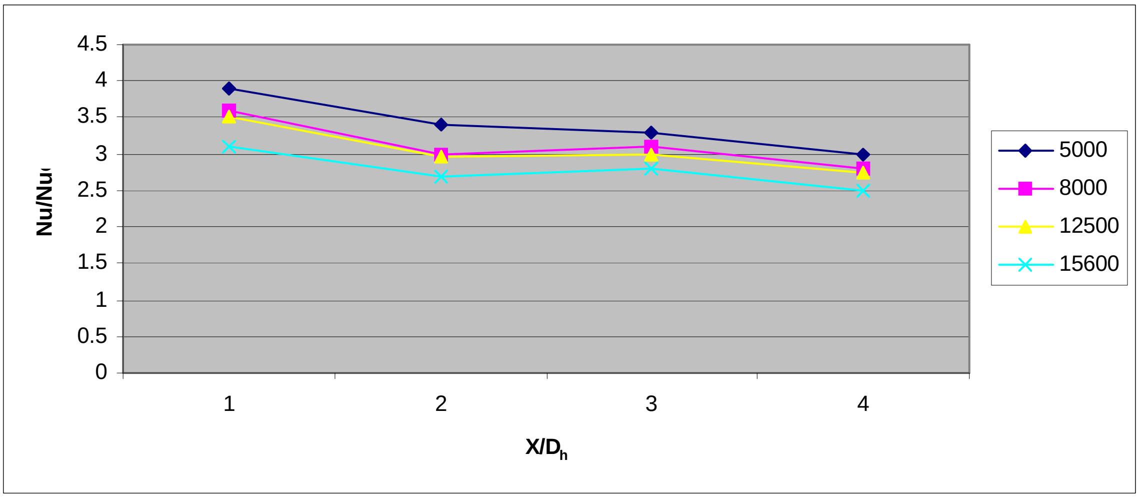

Fig.4 Effect of Reynolds Number on Heat Transfer Distribution for Rectangular duct with continuous V- shaped ribs

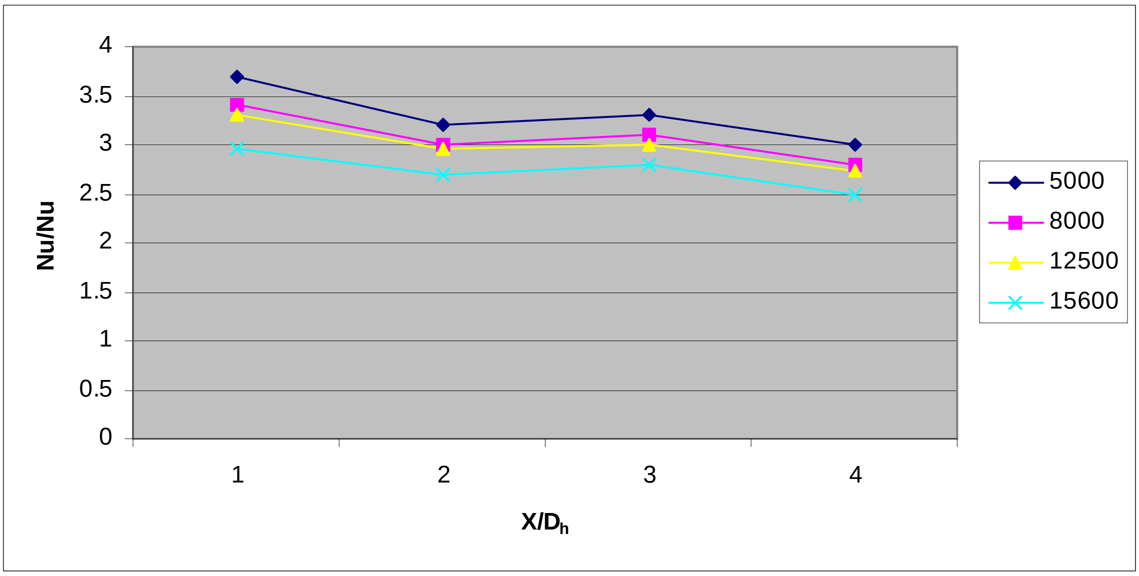

Fig.5 Effect of Reynolds Number on Heat Transfer Distribution for Rectangular duct with discrete V- shaped ribs

Friction Factor Characteristics for smooth rectangular duct, rectangular duct with continuous V- shaped ribs and rectangular duct with discrete V- shaped ribs:

The friction factor ratio was found to increase with increase in Reynolds number due to the resistance offered to the flow of fluid. The heat transverse augmentation was found to vary between 1.5-2 for the rectangular duct with discrete V- shaped ribs and 2-2.5 for the rectangular duct with continuous V- shaped ribs. This enhancement in both the cases was accompanied by pressure drop penalty of 1.5 to 3 on the rectangular duct with discrete V- shaped ribs and 3 to 4 on the rectangular duct with continuous V- shaped ribs. The rectangular duct with continuous V- shaped ribs produced 20 % to 40% higher pressure drop when compared to the rectangular duct with discrete V- shaped ribs for the complete range of Reynolds number investigated.

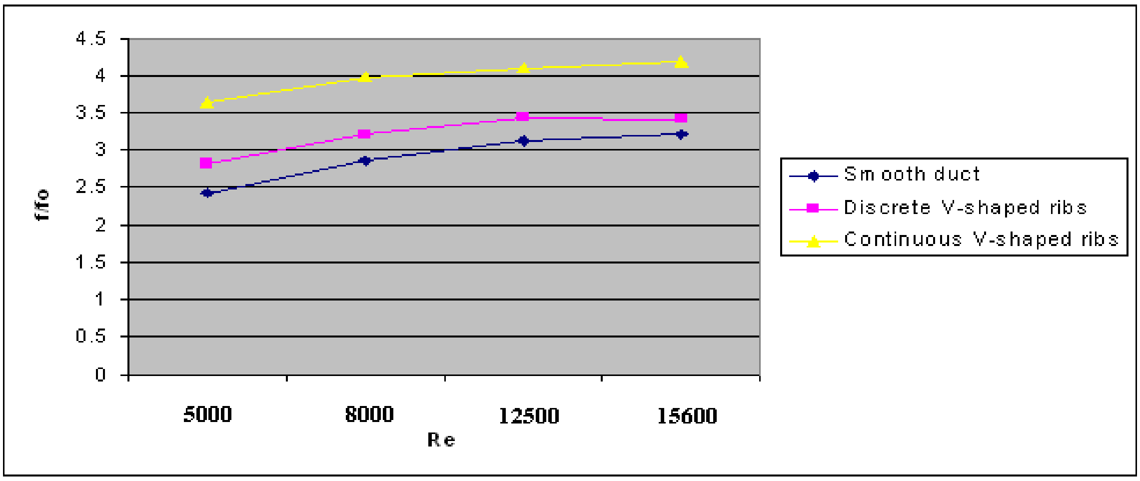

Fig. 6 Average Friction Factor Ratio versus Reynolds Number for Various Configurations

Thermal Performance:

Thermal performance for the plate with internal ribs is drawn as below in fig. 7, shows that thermal performance is increasing with increase in Reynolds number. But the thermal performance for rectangular duct with discrete V- shaped ribs is poor as compared to the rectangular duct with continuous V- shaped ribs. This is due more turbulence and strong vortex formation in rectangular duct with continuous V- shaped ribs. Also it reflects that thermal performance of rectangular duct with continuous V- shaped ribs is more than one which means applying continuous V- shaped ribs is beneficial to increase heat transfer enhancement.

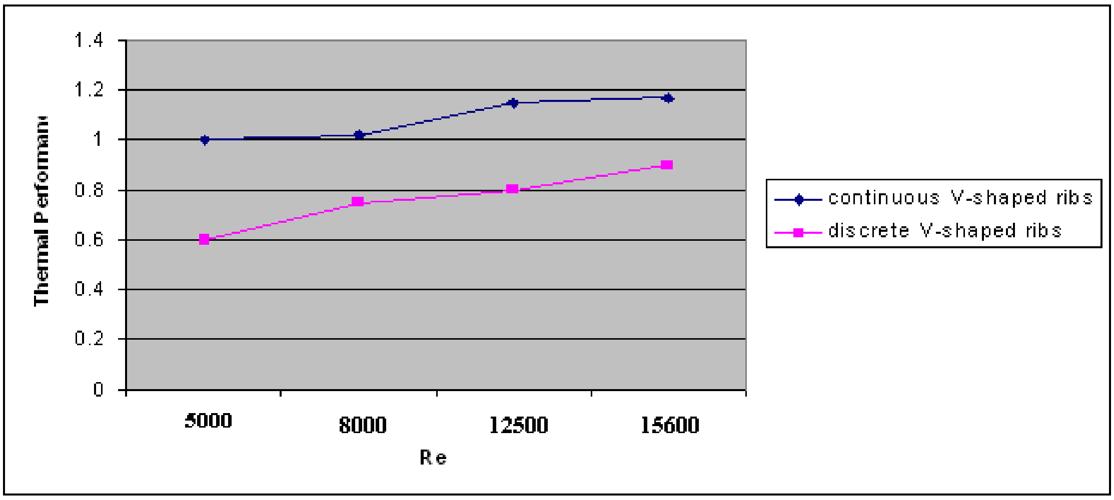

Fig. 7 Variation of Thermal Performance with Reynolds Number for Various Configurations

Acknowledgments

We are thankful to the Hon. Principal Dr. A.W. Kolhatkar and Head of Department Dr. A. B. Borade, Jawaharalal Darda Institute of Engineering and Technology, Yavatmal, Maharashtra, India for providing Mechanical Engineering Research Laboratory for Experimentation study and their moral support.

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