

AN INVESTIGATION INTO MECHANICAL AND THERMAL CHARACTERISTICS OF SYNTHETIC JET ACTUATORS WITH COMPACT PACKAGING

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AN INVESTIGATION INTO MECHANICAL AND THERMAL CHARACTERISTICS OF SYNTHETIC JET ACTUATORS WITH COMPACT PACKAGING

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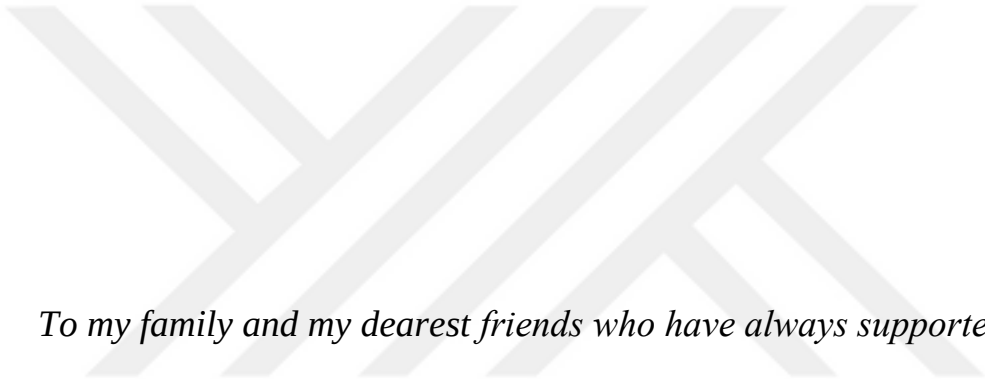
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To my family and my dearest friends who have always supported me...

ABSTRACT

Cooling systems have been one of the most important elements in packaged electronics due to their inseparable relation with performance. Processors, light emitting diodes (LED), servers, advanced communication systems, power electronics and many other electronic components require cooling to function properly and perform better. Therefore, thermal management plays a substantial role in many systems. That said, it must be acknowledged that the electronics aim to have the smallest composition possible, which is why their integrated cooling system must also have small volumes. Cooling mechanism must be designed such that the heat is removed from the cooling target with the highest efficiency, while having the smallest package size. These mechanisms must also provide a low-cost solution, hence the problems in electronics cooling emerge.

When it comes to the densely packaged systems, the “Tendon of Achilles” has always been the operating temperature of electronics. In most of the applications, especially in the printed circuit boards (PCB), novel and smart technologies are required to achieve the best performance while preserving small volume. From end-user to original equipment manufacturer (OEM), numerous applications use electronics packaged with cooling systems.

This thesis studies one of the novel technologies in active cooling in ambient temperature operating conditions, synthetic jet actuators. Synthetic jet actuators play an important role in passive-to-active cooling with condensed volumes, and long life-cycle capabilities. Composed of piezo actuated disks and an orifice, synthetic jet actuators excite the inner air volume by agitating piezo disks and creating an air flow.

Given their high advantage in cooling performance, low cost of material and production brings synthetic jet to the more feasible alternative in electronics cooling. To achieve the best cooling performance with the lowest power consumption possible, optimization of synthetic jet actuators, by a design of jet manufacturing setup and experimental validation and qualification has been studied.

To achieve precision and consistency in mechanical aspects of synthetic jet actuators, a manufacturing rig combined with a robotic dispenser and pneumatic pick and place system has been designed and produced. Visual microscope examination for boundary condition validation has been conducted to minimize the uncertainties resulting from mechanical inconsistencies.

In-house manufactured synthetic jet actuators have been tested to measure their mechanical and thermal performance, using laser vibrometer and PID controlled heat transfer test system.

Orifice shape and design has been studied by experimentation and analytical derivations in this thesis. In-house manufactured synthetic jet actuators are also compared with their alternatives, axial fans, where natural convection, forced convection by axial fan and synthetic jet impingement cases are observed. Later, as a benchmark study, a third-party synthetic jet product has been analyzed and thermally compared with the jets manufactured in this thesis.

Finally, a novel packaging study has been performed to create a product with validated driving frequency at the optimum resonant mode shape, and highest cooling performance, using finite element analysis (FEA) for validation and optimization of mechanical aspect and package design of a final product.

ÖZET

Soğutma sistemleri, elektronik cihazlarda, sistemin performansını direkt olarak etkileyen en önemli unsurlardan biridir. İşlemciler, ıřık yayan diyotlar (LED), sunucular, güç elektronięi ve dięer birok elektronik bileřen, verimli řekilde alıřabilmeleri ve beklenen performansı gsterebilmeleri iin uygun řekilde tasarlanmıř bir soęutma sistemine ihtiya duyarlar. Bu nedenle, ısıl ynetim temellerinden biri olan soęutma sistemleri elektrikte olduka nemli bir rol oynar. Genel kural olarak, bir sistemde elektronik komponentlerin mmkn olan en kk hacme sahip olması gerektięi dřnldęnde, entegre soęutma sistemlerinin de olabilecek en kk hacimde ve mmkn olan en verimli řekilde planlanması gerekmektedir. Ancak ısının soęutma hedefinden en yksek verimle uzaklařtırılması saęlanırken, aynı zamanda dřk maliyetli bir zm sunulmalıdır. Bu da elektronik soęutma sistemlerinde en sık karřılařılan sorun ortaya ıkabilmektedir.

Kk paketli elektronik sistemler sz konusu olduęunda, "Ařıl Tendonu" her zaman elektroniklerin alıřma sıcaklıęı olmuřtur. Gnmzde elektronik uygulamaları ile ilgili yapılan alıřmaların byk bir kısmı, zellikle baskılı devre kartlarında (PCB), soęutma sistemlerinin mmkn olan en kk hacimlerde tutularak, en iyi performansı elde edebilmesi amacıyla yeni ve akıllı teknolojiler geliřtirilmesi hedeflenmektedir. Son kullanıcıdan orijinal ekipman reticisine (OEM), ok sayıda uygulamada soęutma sistemleri geliřtirilmiř elektronik cihazlar veya sistemler kullanılmaktadır. Elektroniklerin termal ynetimi, soęutma kapasiteleri, toplam hacmi ve enerji verimlilięi aısından farklı alıřmalar ve zmler gerektirmektedir. En uygun soęutma sisteminin seimi, tasarım mhendislerinin en nemli

sorumluluklarından biridir.

Bu çalışmada, ortam sıcaklığı çalışma koşullarında aktif soğutmadaki en yeni teknolojilerden biri olan sentetik jetler incelenmiştir. Sentetik jetler, küçük hacimleri ve uzun kullanım ömrü özellikleri ile pasif-aktif soğutma geçişlerinde önemli bir rol oynamaktadır. Sentetik jetler, piezo tahrikli diskler ve bir açıklıktan (delik veya boşluk) oluşmaktadır. Piezo disklerin hareketi ile iç hava hacmi harekete geçmekte ve bu hareket bir iç hava akışı oluşturmaktadır. Soğutma performansındaki birçok avantajın yanında düşük malzeme ve üretim maliyetleri göz önünde bulundurulduğunda sentetik jetler elektronik soğutmada son derece uygun bir alternatif olarak belirlenmektedir. Genellikle bir piezo disk ve küçük bir diyaframdan oluşan sentetik jetler, pasif soğutmadan aktif soğutmaya geçiş ihtiyacı olduğunda önemli bir bileşen olarak kabul edilmektedirler.

Bu çalışmada sentetik jetlerin en yüksek soğutma performansını en düşük enerji tüketimi ile elde edebilmek amacıyla, sentetik jetlerin optimizasyonu, jetlerin üretim düzeneklerinin tasarımı ve deneysel çalışmaların validasyonu üzerinde durulmuştur. Sentetik jetlerin üretiminde mekanik olarak hassasiyet ve tutarlılık elde edilebilmesi oldukça önemlidir. Bu nedenle bu çalışmada robotik bir dağıtıcı ve pnömatik alma ve yerleştirme sistemi ile birleştirilmiş bir üretim düzeneği tasarlanmış ve üretilmiştir. Geliştirilen düzenek aracılığı ile üretilen sentetik jetler, mekanik ve termal performansları lazer vibrometre ve PID kontrollü ısı transfer test sistemi kullanılarak test edilmiştir.

Orifis şekli (açıklık veya boşluk), sentetik jetlerin termal performansını etkileyen en önemli parametrelerden biri olarak belirlenmiştir. Bu tezde, açıklığın ağız şekli ve tasarımı, deneysel ve analitik analizler ile incelenmiştir. Tasarlanan düzenek ile

retilen sentetik jetler, soğutmada kullanılan bir alternatifleri olan eksenel fanlar ile doğal konveksiyon, eksenel fan ile zorlu konveksiyon ve sentetik jet arparak soğutma durumlarının gözlemlendiğı bir deneyde karşılaştırılmıştır. Daha sonra, bir kıyaslama alışması olarak, farklı bir üreticiye ait bir sentetik jet aynı şekilde analiz edilmiş ve bu tezde retilen jetlerle termal olarak karşılaştırılmıştır. Sentetik jetin alışma frekansı ve voltaj genliğini bulmak için osiloskop ile ölçümleri yapılmıştır. Son olarak, nihai rnn mekanik parametreleri ve paket tasarımının doğrulanması ve optimizasyonu için sonlu elemanlar analizi (FEA) kullanılarak, en düşük frekans ve en yüksek soğutma performansına sahip bir rn oluşturmak için bir paketleme alışması yapılmıştır.

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CHAPTER I

INTRODUCTION

This chapter provides an overview of electronics cooling and challenges faced in the field. Amongst the technological advancements, the relation of small footprint for electronics and their impact on the cooling mechanism design is highlighted. General importance of cooling requirement for the performance of electronics is explained. A novel technology product, synthetic jet actuators are introduced, along which the working principle is briefly explained. Later, the motivation of the thesis is presented, including an overview of the studies performed on the synthetic jet actuators and their thermal, mechanical and flow performances.

1.1 Importance of Volume and Footprint Area in Electronics

Since the use of electronics have been massively multiplied over the last decades, from end users to industries, every application requires smaller weight along with better computation speed and performance. Especially in the aerospace, defense and medical industries, a high demand of “technological shrinking” is present. The need for lower form factors requires numerous advancements in electronics. In aerospace applications, smaller packages of electronics result a vast cost reduction in terms of available weight that can be transported into space [1]. Main component of electronics is generally formed of transistors, small circuit pieces that contain and shuttle the information. As expected, shrinking need has also affected the manufacturing of transistors. Consumer electronics such as computers and smart phones have smaller electronics packaging as the developments have been made.

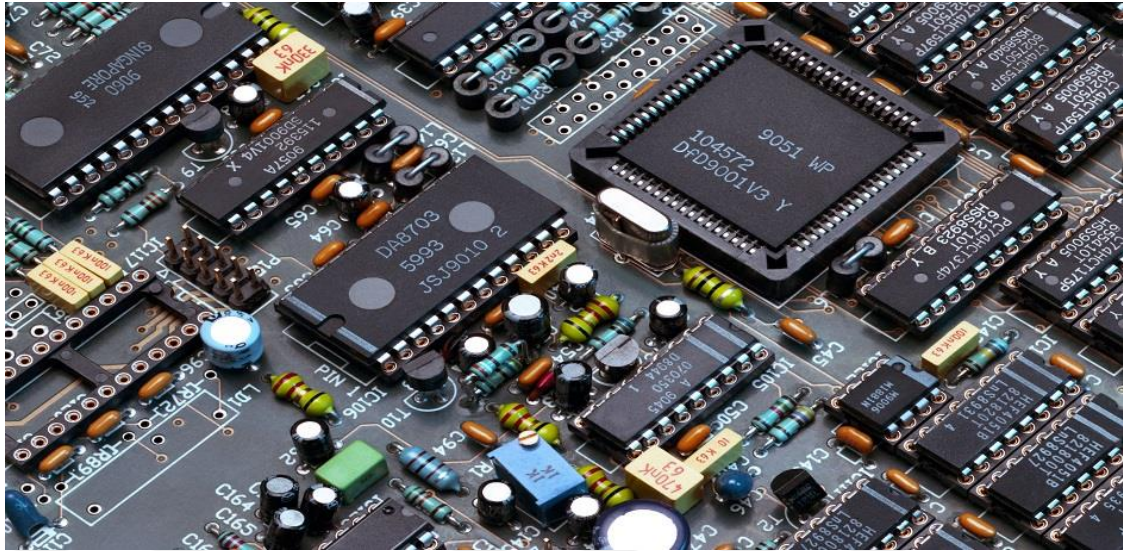


Figure 1: Embedded electronics [1]

Mobile phones in 90's have left their place to the slim and lightweight smart phones. Central processing units (CPUs) have smaller layer thicknesses coming from 90 nm to 14 nm, while having the number of transistors multiplied each year. This creates a field of massive possibilities in terms of computation and consumer electronics.

As the cutting-edge technologies forced the electronics packaging field to test its limits, physics came into play with the environmental boundaries, thermal management. As the information is shuttled through the electronics by electrons, densely packaged systems resulted high temperatures which decreased the performance of electronics, along with the electromagnetic disturbances and noises, resulting even malfunction of components. Later, natural need of dissipation of excessive heat has been emerged and has always been considered together with electronics today.

1.2 Electronics with Cooling Requirements

As the form factor of electronics gets smaller, a challenge of high heat fluxes in packages has emerged. The capability of transporting information with great speed in electronics gets affected as the hot spots have higher temperatures to operate. Most of the electronics suffer from thermal management problems, especially in confined spaces. Thermal management has become a substantial industry with a yearly market cap of over \$15 billion [2] investigating the best performance milieu for electronics while preserving a tight space for cooling equipment.

General cooling types, active and passive cooling are used in electronics cooling. Former is used in applications where moderate heat accumulation is present and temperature rise can be prevented with heat sinks, grills, and so on. For active cooling, most of the applications use air cooling due to their availability, low cost and less maintenance requirements. Axial fans are typical examples for air cooling of electronics. Air cooling has disadvantages because of their relatively large size, noise, and energy consumption because of the propeller sizes. This creates a need for an alternative having a smaller package size and low energy consumption.

For liquid cooling systems, numerous cooling alternatives can be used due to their higher heat removal capacities, especially in refrigeration. Heat transfer with lower-than-ambient temperatures usually used in computational applications where extreme temperatures of component are possible, such as central servers at +5°C or quantum computers at -270°C [4].

1.3 A Novel Cooling Technique – Synthetic jet actuators

As the need for high performance in cooling with small package size increases, novel technologies have been used to come up with feasible solutions. One of the examples serving the passive-to-active cooling transition, is synthetic jet actuators. Synthetic jet actuators are products of novel technologies such as piezo materials, providing a compact cooling package with a relatively lower energy consumption.

Synthetic jet actuators have been used frequently in flow control applications, due to their turbulent flow formation capacity, such as boundary layer interference to reduce flow separation in fuel consumption reduction applications [5]. They are also used as cooling components providing zero mass flow into the control volume of the application, i.e., conforming to zero-net mass flux principle. These jets agitate the air inside their cavity by transporting through an orifice or nozzle, which is used as both inlet and outlet for the cooling mechanism. The jet flow of air that is used from the surrounding creates a vortex-shaped turbulent flow, dissipating heat with increased heat rates from the target.

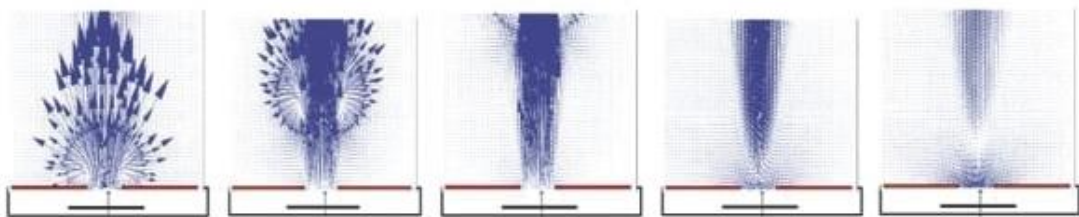


Figure 2: A typical flow scheme for synthetic jet actuators [6]

Synthetic jet actuators for cooling applications provide jet with cyclical manner. They usually work with an embedded wave form generator, providing a sinusoidal voltage wave with a frequency of usually between 100 and 10000 Hz. This frequency is

generally the resonant frequency of the mechanical model, creating higher displacement of diaphragm, thus creating higher flow of jet [7]. Synthetic jet actuators are used in electronics cooling where passive cooling is not sufficient and densely packaged cooling system is required. Especially in aerospace and defense applications [8, 9], their high reliability compared with axial fans comes in handy. Compared to axial fans, synthetic jet actuators have much smaller package size due to their simple and planar design. Apart from their sizes, synthetic jet actuators have longer life than axial fans, since they do not have rotating parts such as bearings, hence the reliability of synthetic jet actuators.

Synthetic jet actuators are used vastly for impingement cooling, where a target plate is aimed by an orthogonally directed air jet, and increasing local heat transfer by stagnation of turbulent flow. Synthetic jet actuators are substantially feasible alternatives compared with conventional impingement cooling mechanisms, due to their simple design.

Operating conditions of synthetic jet actuators vary their cooling performance, especially in mechanical design. A number of studies has been performed in synthetic jet cooling, shedding a light on mechanical parameters affecting thermal and flow behavior. Arik et.al [10] investigated high frequency synthetic jet actuators with infrared microscopic thermal camera, to observe the local and global heat transfer coefficients. Study has focused on a jet actuator with a square orifice shape, and operating frequency of 4500 Hz, which is close to the resonant frequency. They have observed that synthetic jet actuators used in the study proved to have a thermal performance several times higher than natural convection. They have also studied Helmholtz resonator type synthetic jet actuators [11], quantifying and measuring their

impingement heat transfer, with a driving frequency range of 3 kHz and 5 kHz. Hotwire measurements have also been conducted to characterize the velocity values at given operating conditions. In another study, Chen et al. [12] have studied the behavior of synthetic jet actuators and their study resulted a synthetic jet having an air jet velocity of 40 m/s. These jets have also been manufactured with accessible piezo materials.

Ikhlaq et al. [13] have studied hand-made synthetic jet actuators and their correlation between disk displacement and heat transfer rate, using computational analysis. They have proposed a relationship between displacement at driving frequency and cooling performance.

The key component of synthetic jet actuators, the piezo material, has been proved to be a right choice to drive the diaphragm of the jets. A study by Bryant et al. [14] tackled the analysis of feasibility of flow by synthetic jet actuators with piezoelectric diaphragm. They have observed the effect of flow generated by piezo-jets over bluff body surfaces, concluding that the orifice size is one of, if not the most important parameter in mechanical design of jets.

In a study by Arik and Icoz [15], a prediction has been performed for unsteady synthetic jet actuators. Relationship that is introduced in the study has been shown to estimate impingement heat transfer with an accuracy of 25%, given a large interval of operating conditions and geometric iterations. They have also studied [16] steady air impinging cases, along with the unsteady air impinging heat transfer in electronics.

Arik et al. [17] also focused on the power consumption stand point of novel alternative cooling options in electronics cooling, including synthetic jets. Three

types of cooling mechanisms, synthetic jet, piezo fan, and an axial fan with a small form factor. It is observed that the synthetic jet actuators resulted promising cooling performance with given energy consumption, which is to say that the synthetic jet actuators will take place in most of the electronics applications in the future.

Many research studies have been conducted to tackle the performance optimization of synthetic jets, though the field is observed to have more challenges and gaps on technical literature to improve. Recently, Arshad et al. [19] have critically reviewed the studies on the synthetic jets and they have pointed out research gaps and challenges such as phase and inclination angle, dimensionless stroke-length, effect of low Reynolds number, and so on.

In a study by Chaudari et al. [20], the effect of orifice shapes for the synthetic jets are experimentally investigated. Circular, square and rectangular orifice shapes have been introduced in the study, with varying hydraulic diameters and aspect ratios. They have reported that the synthetic jet with a rectangular orifice, having a high hydraulic diameter and smaller aspect ratio is more feasible for space-constrained applications, while square orifice jet showed more performance at higher axial distances.

Ikhlaq et al. [21] have performed an experimental and computational study on low, medium and high frequency synthetic jets, to propose a correlation of Nusselt number in terms of excitation frequency, target and orifice geometry, and length scale. They have proposed a correlation that estimates Nu values with an accuracy of 25%. They have also observed that high frequency synthetic jets consume more energy, due to their high frequency operating conditions and small geometry.

Garg et al. [22] have investigated the effect of synthetic jets with unsteady

impingements by experimentations, using jet velocity measurements and infrared camera temperature readings. They have used velocity measurements to determine the resonant frequency of the mechanical structure, along which the heat transfer coefficient resulted the highest. They have found that the performance of the created air jet depends critically on the driving frequency and on the distance between orifice and target. They also have observed a maximum heat transfer enhancement of 8.2 over natural convection, by the utilization of synthetic jets.

Kim and Gary [24], studied the effect of the orifice jet, with a motivation of optimization for the rectangular orifice synthetic jets. They introduced a modular synthetic jet generator design, along which variance of cavity dimensions and aspect ratio is investigated. They have observed that the aspect ratio of approximately 10 for rectangular orifice resulted the highest jet velocities at generator outlet.

Synthetic jet actuators are frequently used to improve the heat removal from heat sinks. Mahalingam and Gelezer [25] investigated the improvement of heat removal rates from heat sink, comparing continuous flow air blower and synthetic jet actuator. They have observed that synthetic jet actuators resulted approximately 40% higher removal rates than an air blower at 3 to 5 cfm. They have also found that the heat transfer coefficients between the fins have improved 2.5 times with synthetic jets.

Mangate and Chaudari [26] introduced the multiple orifice concept on the thermal behavior of the synthetic jets. They have designed a satellite orifice configuration in which they observed the effect of heat removal from heated surfaces. They have found that multiple orifice synthetic jets provide 12% more heat removal rates, while satellite orifices have negligible effects on cooling performance along the axial distances. They have concluded that central orifice is a critical component while

cooling with synthetic jet actuators.

Mangate et al. [27] supplemented their earlier study by investigating the multiple orifice synthetic jets with geometry shape variances. They have also observed the flow of jets with PIV measurements. Their findings showed that the central orifice plays a significant role, as its absence drastically drops the cooling performance. Furthermore, diamond shaped neighboring orifices resulted higher Nu values. Synthetic jets with multiple orifices also proved to remove heat with 75% more capacity, compared with single orifice synthetic jets.

Pavlova and Amitay [28] have looked into the thermal and flow behavior of both high and low frequency synthetic jets. They have experimentally compared the jets by varying the Re, St values, and length scale parameters. They have observed the cyclical synthetic jet cooling is 3 times more efficient than continuous jet cooling, which is corresponded to the high heat removal rates of coherent vortex rings that are generated with cyclical jet formation. Their results stated that the high frequency jets provided more cooling performance according to the given operating conditions.

Ramabhadran et al. [29] have studied the optimum electronic module design of synthetic jets, in order to achieve low distortion and high efficiency synthetic jet driver. They have determined a feasible digital-to-analog converter (DAC), focusing on the inherent flyback nonlinearity of the low harmonica signal. They have achieved a jet driver design that proved itself to be low power consuming and low distortion capabilities.

Seeley et al. [30] presented an analytical model of synthetic jets, comprising structural, acoustic and heat transfer parameters. Although the disk velocity and exit velocity resulted some inconsistencies, their model agreed well with the relationship

between heat transfer coefficient and sound intensity.

Silva-Llanca and Ortega [31] proposed a formulation for the airflow velocity of synthetic jets in terms of Reynolds number, taking into consideration the formation of secondary vortex dynamics. They have observed that the vortex pair that arrive to the target creates secondary pairs, creating a higher heat transfer enhancement, which is directly proportional to the Re value and inversely proportional to the driving frequency.

Song et al. [33] studied the prediction of life for LED systems that are cooled via synthetic jets. In their study, hierarchical reliability model has been presented to determine the lifetime of an LED light bulb, which is cooled by a synthetic jet. They have concluded that the decay of the synthetic jet was extremely low, while a 76% of lumen maintenance has been observed, after an operation duration of 50,000 hours.

Tan and Zhang [34] have introduced a geometric approach to synthetic jet performance investigation by presenting multiple orifice and single orifice jet design. They have created slot and circular shaped single orifice jets and same geometry with three-orifice jets, and observed their mechanical and thermal performance. They have found that orifice-to-target distance is significantly affecting the cooling performance with steep changes, especially for single orifice and hole orifice designs. They have also confirmed that the multiple orifice synthetic jets compete with single orifice jets only at optical mode, which is a critical point at operating conditions, and must be acknowledged when designing applications.

Valiorgue et al. [35] have proposed a power law relationship of synthetic jets their study where, phase-revolved flow field and local heat transfer data is presented. They have selected orifice-to-target distance ration as 2. PIV measurements and local heat

transfer readings have resulted a correlation between Nu and Re values.

Xu et al. [36] have investigated the impinging air jet from a synthetic jet generator on a target wall. They have observed using PIV measurements, that cyclical jet results a different vortex formation after third cycle of impingement at Re value of 126. As the impingement continues, vortex differences begin to stop and a cyclical vortex of similar shapes begin to occur. To supplement on this [37], they have also studied the effect of this flow behavior by varying the Re number and preserving other parameters. It is found that the flow behavior is significantly affected from Re number. At low Re values, it is observed that the vortex rings preserve coherence for a significant distance, while at high Re values, they lose their coherence quickly after impingement.

Yu et al. [38] have compared agitator and synthetic jets in heat sink cooling enhancement study, focusing on the heat transfer enhancement at each scenario. They have observed that agitators improve heat transfer capacity by 64%, while synthetic jets improve it by 31%. It is also observed that combined cooling improved heat removal capacity by 81%. Synthetic jets are found to be efficiently dissipating heat from the heat sinks due to the turbulent flow creating, especially at the near the tip of the fins.

Ikhlaq et al. [39] performed a computational study on the synthetic jet actuators, investigating the momentum and heat transfer distribution of low frequency applications. They have proposed a relatively more complex technique in 3D computational analysis, the moving mesh technique. In their study, instead of defining boundary conditions at the jet exit, they have introduced boundary conditions at the diaphragm of synthetic jet, and generate the solution by using a

moving mesh in the time domain. The CFD analysis resulted that at the phase angle of 135° , stagnation pressure is observed the highest. It is also observed that the jet flow pressure gradients dissipate at the length scale of 4 for all phase angles.

In one of the earlier studies, Uttukar et al. [41] have focused on the cavity design and its effect on the mechanical and flow behavior of the synthetic jet actuators. They have found that cavity variances in wide range result only a 7% change in momentum coefficient, which implies that the cavity parameters are not crucial for the performance of synthetic jets.

On their one of the earliest studies, Agrawal and Prasad [43] have investigated the properties of the vortices in the self-similar turbulent jets, in a fluid media measurement setup. They have taken PIV measurements and generated probability density distributions (PDF) for eddies of different size, namely small, intermediate and large. These fields are used for categorization of the vortices appearing according to their vorticity, energy, size and circulation. It is observed that radial distributions of vortices and their population can be characterized in terms of time averaged and non-monotonic trends of an axisymmetric jet.

Alimohammadi et al. [44] have generated a computational model in their study, predicting the heat transfer characteristics of a steady impinging air jet in transient flow regime. Their computational solution proved to have a maximum of 5% deviation, specifically in local Nu values. Validated with experimentations, the model can also capture the near-wall large temperature and momentum gradients, creating a field for detailed computational analysis on synthetic jet performance.

Bhapkar et al. [45] have studied the acoustic and thermal performance of synthetic jets having different orifice shapes, namely, circular, rectangular, square and elliptic

orifices. They have found a strong correlation between acoustic levels and heat transfer coefficients of the jets. It is observed that the resonance frequency of synthetic jets resulted the lowest sound intensities, while having the highest heat removal rates. From the orifice geometry stand point, elliptic orifice jet resulted the highest heat transfer coefficients at smaller axial distances, while highest performance at large axial distances is obtained with square and circular jets.

Chaudari et al. [50] have introduced the satellite orifice term into the jet actuator design, investigating the effect of the multiple orifice and their quantity on the cooling performance. Interestingly, multiple orifice jets have performed 30% higher than conventional single-orifice jets. Also, number of satellites has been found to be in correlation with axial distance, in terms of heat transfer coefficient. It is observed that 2 satellite orifice proved higher performance at lower axial distances, while 4 and 8 satellite orifice jets performed better at higher axial distances. Their study opened a concept of practical solution that can be used for mechanically constrained applications in electronics cooling.

In their later study, Chaudari et al. [51] also tackled the effect of heat sink cooling performance enhancement with synthetic jets, using different orifice shapes. They have also included a comparison of axial fan with synthetic jets for heat sink application. They have found that synthetic jets increased the heat removal rate of heat sink by approximately 4 times, comparing with the natural convection. However, the synthetic jets performed slightly worse than an axial fan. Shape of orifice with similar hydraulic diameter resulted negligible differences.

Fanning et al. [52] have investigated the effect of target-to-orifice distance, orifice-to-orifice distance of a pair of synthetic jets in an experimental study. They have

observed the behavior of the flow by PIV measurements and thermal performance by infrared thermography. It is observed that the pair synthetic jets with a driving phase difference between 60° and 120° have resulted the highest heat removal rates, at orifice-to-orifice and orifice size ratio of 3. They have also observed the phase difference between 135° and 180° proved better cooling performance for hot spot cooling.

Persoons and O'Donovan [54] have generated an analytical model that uses gas dynamics, to estimate the synthetic jet velocity and actuator deflection, using cavity pressure. This model is validated with axisymmetric jet experimentations, and does not pose any restrictions to the synthetic jet structural information, which is why it can be applied in most jet velocity measurement applications.

Persoon et al. [55] also investigated the performance of adjacent jet actuators and their performance comparison between single orifice synthetic jets. They have observed experimentally, that the pair synthetic jets prove to have 90% more heat removal rates. They have determined the optimum phase difference to be 120° , which is agreeing with the study of Fanning [52].

Persoon et al. [56] have also introduced a general correlation for the stagnation point Nu value for axisymmetric synthetic jets. The correlation is corresponding to Re number, target-to-orifice distance, and stroke length. It is observed to be matching with more than 85% of the experimental data.

Trávníček et al. [57] studied the effect of actuation on the synthetic jets, experimentally. They have compared the axisymmetric actuation with a reference single actuation. Helical and bifurcating actuation is introduced on the samples, in which PIV, flow visualization, hot-wire anemometry and naphthalene sublimation

techniques were used to observe the characteristics. For small orifice-to-target distances, a 40% increase in impingement cooling performance is observed.

In their recent study, Yadav and Agrawal [58] have studied the second peak point in heat transfer coefficient values for synthetic jets. They have characterized the vortex stagnation of impinging surface into two sections: inner layer and outer layer. They have concluded that the second peak in heat transfer is resulted from the separation/reattachment which is caused by the turbulent flow characteristic of the vortex, increasing normal and shear stresses. This effect summarizes the higher heat removal rates of cyclical synthetic jets, compared to continuous jets.

Arik and Uttukar [59] have performed a thorough 3D computational analysis on synthetic jet flow characteristics, and validating the solution with visual experimentations. Their study has resulted critical information on the distance between cooling target and orifice of the jet, as a result of the dipole shrinkage. It is also observed that the jet velocity is approximately 4 times higher than overall velocity of vortex dipole.

Given the examples above, it can be concluded that the mechanical parameters, thermal behavior and material analysis of synthetic jet actuators has been conducted densely in academic studies. However, further experimentation and application specific optimization is required, in order to bring the synthetic jet actuators to the point where they are more feasible and accessible. This thesis focuses on the optimization of synthetic jet actuators by a design of manufacturing station, along with their mechanical and thermal performance measurements. Both side orifice and central orifice jets have been produced with the designed manufacturing rig, detailed information can be found in Chapter II. The main motivation of this study is to obtain

a cooling mechanism with high thermal performance while preserving small form factor. In the light of the technical literature review, critical parameters such as orifice shape, orifice size, driving frequency, driving voltage amplitude and so on, are discussed in this study. Furthermore, comparison of in-house manufactured jets with commercial synthetic jets and an axial fan is performed.

Finally, a mechanical modeling is performed to acquire a direction for designing a novel packaging technique, which includes a metallic holder to provide a practical solution. According to the input obtained from the experimental and numerical studies, a novel design has been generated. Mode shapes and resonant frequencies are determined for the novel design, and manufacturing and validation is completed, comparing with the earlier studies.

CHAPTER II

SYNTHETIC JET MANUFACTURING

In this chapter, an introduction of a manufacturing station capable of producing synthetic jets is provided. Main components and their design criteria are explained in details, with respect to their impact on the structural integrity of synthetic jet actuators. Manufacturing flow chart and explanation of each process is also provided.

2.1 Synthetic Jet Manufacturing System

In order to achieve precise and repeatable synthetic jet actuators, a manufacturing rig has been designed. Using a robotic dispenser to apply the elastomeric wall, a pick and place mechanism has been designed for accurate and precise assembly of piezo disks.

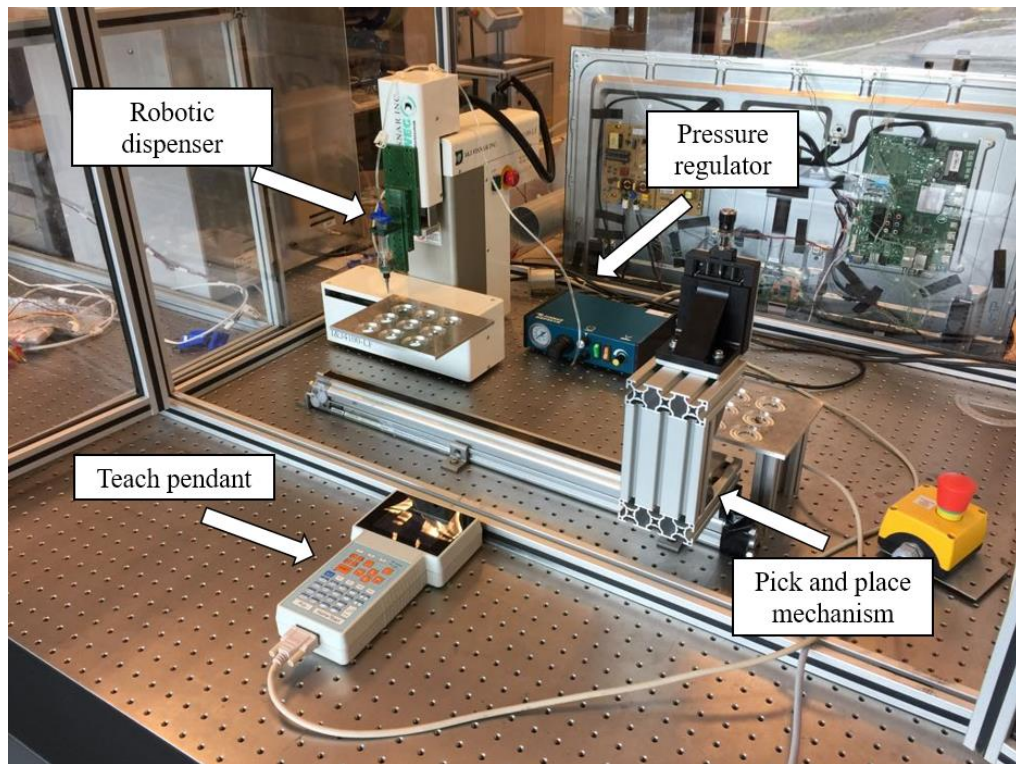


Figure 3: EVATEG synthetic jet manufacturing station

Synthetic jet actuators are affected by the misalignment of disks, i.e., piezo disk center and orifice plate. That said, distance between disks play a great role in cavity volume tolerance, which is why at the manufacturing procedure, parallel and co-centric placement of upper disk must be achieved. To overcome problems from misalignment, pick and place mechanism must have a rigid construction. Using aluminum structural sigma profiles, pick and place mechanism has been designed. To achieve repeatability, pneumatic cylinders have been selected for axial motion. Pick and place mechanism has been designed to have 2 axis motion, along with a pneumatic ejector set of nine pieces, to facilitate handling of the disks without damaging them. To adjust the distance between disks, a micrometer adjustable table has been introduced to the system. As a result, all of the dimension parameters can be adjusted with synthetic jet manufacturing station. All of the synthetic jets, along with the holders, have been produced using this station.

After production, synthetic jet actuators are left out to cure for about 24 hours. This procedure is critical because deformations and attachment problems can occur if the elastomeric wall is not fully cured. While being cured, piezo disks tend to migrate and change their position because of the non-uniform dispensing and curing of the wall. To prevent this, micrometer table is left positioned on the synthetic jet actuators, applying a constraint in movement. At each step of manufacturing, visual inspection is performed and faulty components are separated to the reject area, where recycling or disposal of the materials are done. Manufacturing flow chart is shown in Figure 4. Whole manufacturing rig is enclosed with a plexiglass chamber, in order to prevent unwanted temperature distribution gradients, causing non-uniform elastomer curing.

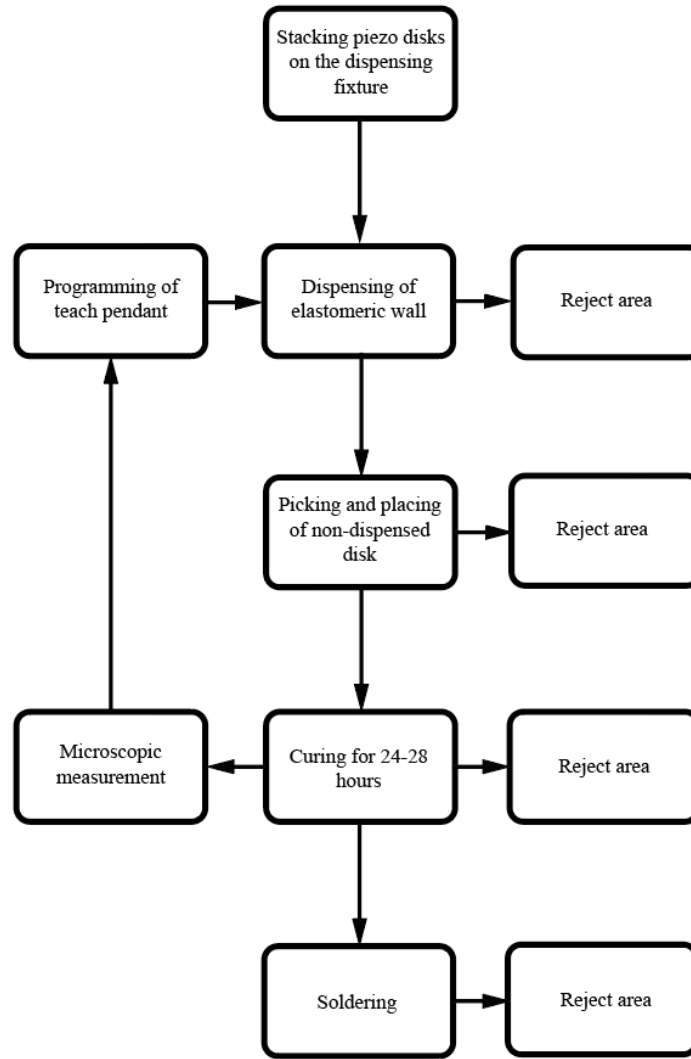


Figure 4: Manufacturing flow chart of synthetic jets

To validate and control the dimensional accuracy of synthetic jet actuators, microscope imaging is performed. Images are used in dispensed material of non-assembled disk, to measure the dispensed thickness of the elastomeric wall. A 3% accuracy for wall thickness is aimed in the produced jets. This is mostly dependent on the pressure applied to the robotic dispenser. To achieve precise wall thickness, dispensing pressure for the robot is reduced to 3 bar from the pressure regulator. Dispensing syringe is constantly monitored for hidden air volumes, which can affect

the dispensing force, and cause non-uniform lines having variable wall thickness.

In this thesis, each synthetic jet model is produced in sets of three, in order to reduce the uncertainty and error in experimentation. Further uncertainty analyses have been explained in the following chapter.

2.2 Robotic Dispenser System

Elastomeric wall is used as a structural boundary that allows a volume of air in between two shims, also preventing an electric conductivity. These walls are usually made out of dispensed silicone material, having an elastic structure. This elasticity creates a less stiff package of synthetic jet actuator, allowing the mechanical structure to have smaller resonant frequencies, and thus operating at lower sound intensity.



Figure 5: Robotic dispenser

Application of the elastomeric wall must be done delicately, since manually dispensed wall materials contain non-uniform shape and integrity, causing mechanical faults such as disk misalignment, small cavities that bypass the air flow instead of orifice, different resonant frequency and mode shapes, and so on. To prevent these imperfections, a robotic dispenser (Figure 5) has been used, creating a high accuracy and repeatability. Composed of 3-axis linear motion system, robotic dispenser provides a dispensing of silicone based liquid materials with high accuracy. All of the axes are driven with stepper motors using closed loop position control. The linear accuracy of all of the axes is ± 0.01 mm. Having a workspace are of 15 x 15 cm, dispensing fixture containing 9 slots for synthetic jets is designed, allowing 9 piezo disks to be prepared at once. Before dispensing, piezo disks are loaded on the dispensing fixture by hand, which is an aluminum plate containing 9 slots to align the disks. Fixture is later used as a curing chamber after the production.

The dispensing route is programmed by a handheld teach pendant, which contains parametric programming interface. Distance, dispensing geometry, speed, outflow percentage, retraction vacuuming, overflow compensation and many other parameters are adjusted in order to achieve uniform and steadily dispensed material. Dispensing is performed by a regulated pressurized air implemented on a liquid container, which is forced out of the nozzle. After manufacturing of the synthetic jets, iterations are performed in order to minimize the defects occurring from dispensing. Revision of the dispensing parameters are constantly done to increase the quality, according to the dimension controls from microscopic measurements.

2.3 Pick and Place Mechanism

Placing the upper disk of synthetic jets on the recently dispensed wall material is critically important regarding their mechanical behavior. Alignment, co-centricity, and distance between the disks are important parameters that affect the main operating conditions of synthetic jets, such as the driving frequency [47], and air flow [21]. To achieve accuracy and repeatability, a design of a pneumatically driven pick and place mechanism is introduced. System is composed of 2-axis linear motion system and a vacuum handling grippers, both operated by pressurized air, by which the robotic dispenser is also driven. Pick and place mechanism (Figure 6) is designed to pick 9 disks, upper disks of synthetic jets, and position them on top of the dispensed disks, which are located on the dispensing fixture, and delicately place the disks on the elastomeric walls. This procedure is aimed to be performed on 2-axis motion, vertical motion for picking and placing, and horizontal motion for moving between the stacked disk holders and dispensing fixture. Each motion is designed to have 2 end-positions, forward and backward, to facilitate the control of the system by eliminating any mid-positioning of axes, i.e., addition of a closed-loop control system.

Pneumatic control and command of the pick and place mechanism is performed by toggle-lever pneumatic valves. These valves are included with adjustable air exhaust fittings, enabling the adjustability of the motion speed for each axis. For motion axes and vacuum ejectors, a 5/2 and 3/2 lever toggled valves are used, respectively. Figure 7 depicts the pneumatic circuit diagram of the pick and place mechanism.

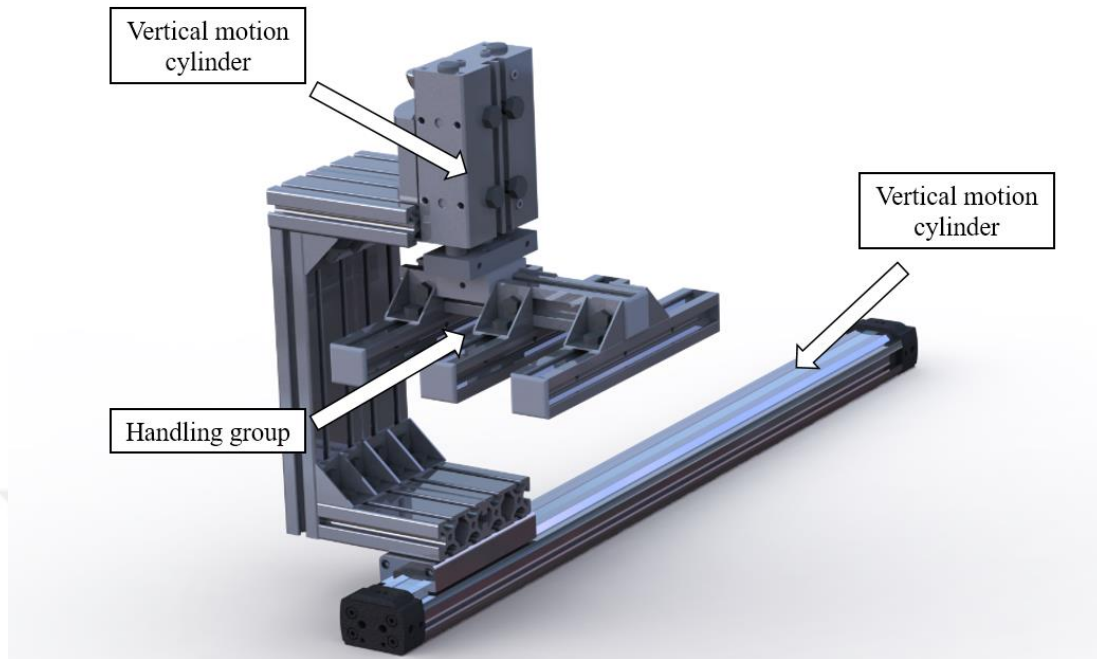


Figure 6: Pick and place mechanism

Handling of the upper disks is performed by a vacuum ejector and vacuum pad group, in which an active vacuuming is performed using the Bernoulli's principle, and holding the disks from the surface. This handling type facilitates the placement of the disk on top of the elastomeric wall, without any risk of deforming the wall or the risk of wall material contacting the handling system. Furthermore, to adjust the distance between the disks, a micrometer table is mounted on the handling guide. Micrometer table provides an adjustability of the end point position of the vertical axis, by an accuracy of 0.01 mm.

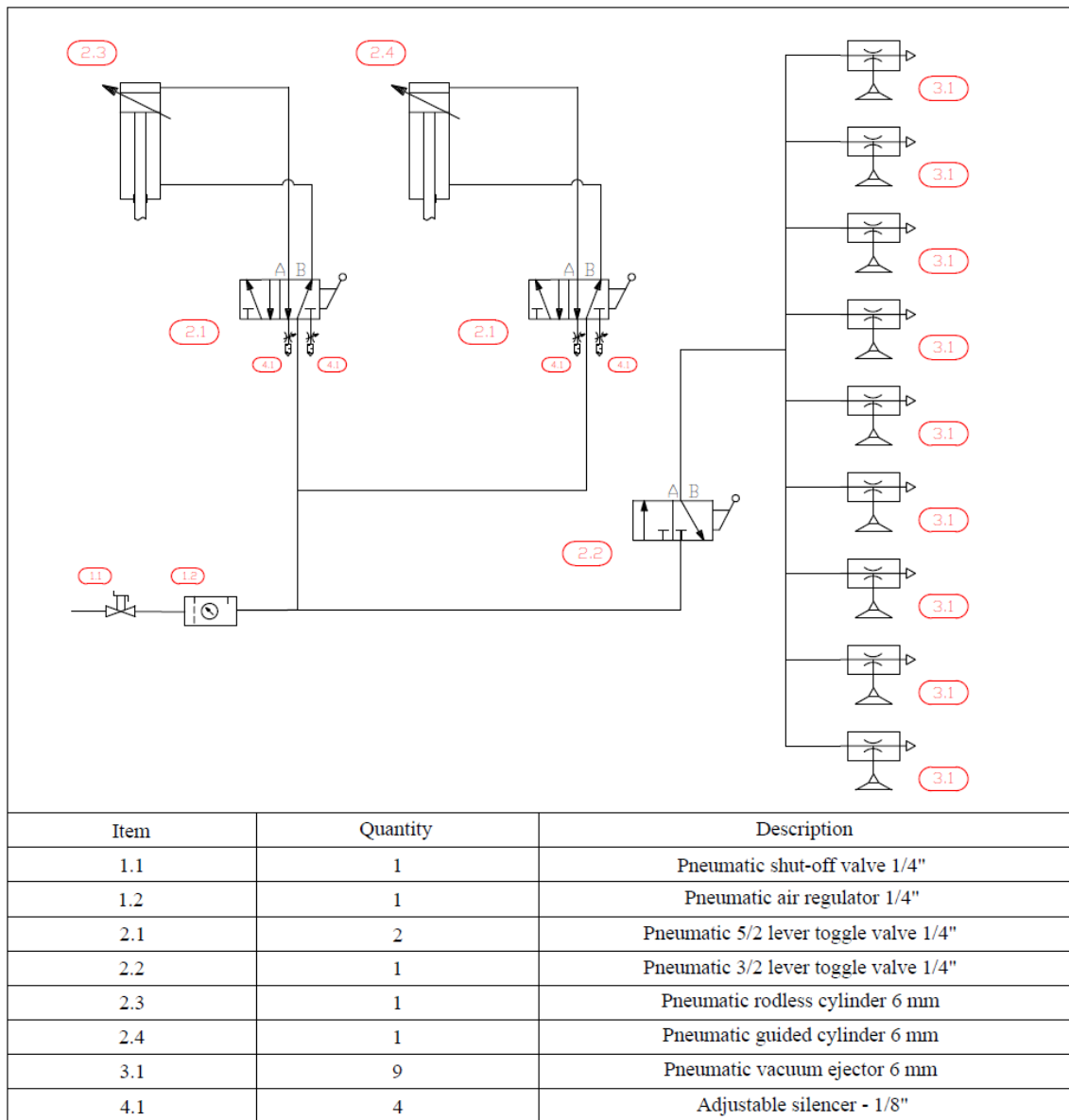


Figure 7: Pneumatic circuit diagram of pick and place system

2.4 Soldering

Excitation signal is implemented on the piezo disk, which is connected serially to the metal shim. To close the circuit, metal shim must also be connected to the signal circuit. That said, soldered cables and solder itself accumulate the mass of the mechanical structure, resulting shift in resonant frequency and also changing the mode shape at the resonant frequency. To get the repeatable results at every

manufactured synthetic jet, soldering points on the piezo disk and the cable thickness and length are determined to be same for each jet. After curing, synthetic jet actuators are soldered with cables for running. 2-core cables with 1.5mm^2 cross-section area thickness are soldered on the steel and piezo surface, respectively (Figure 8).

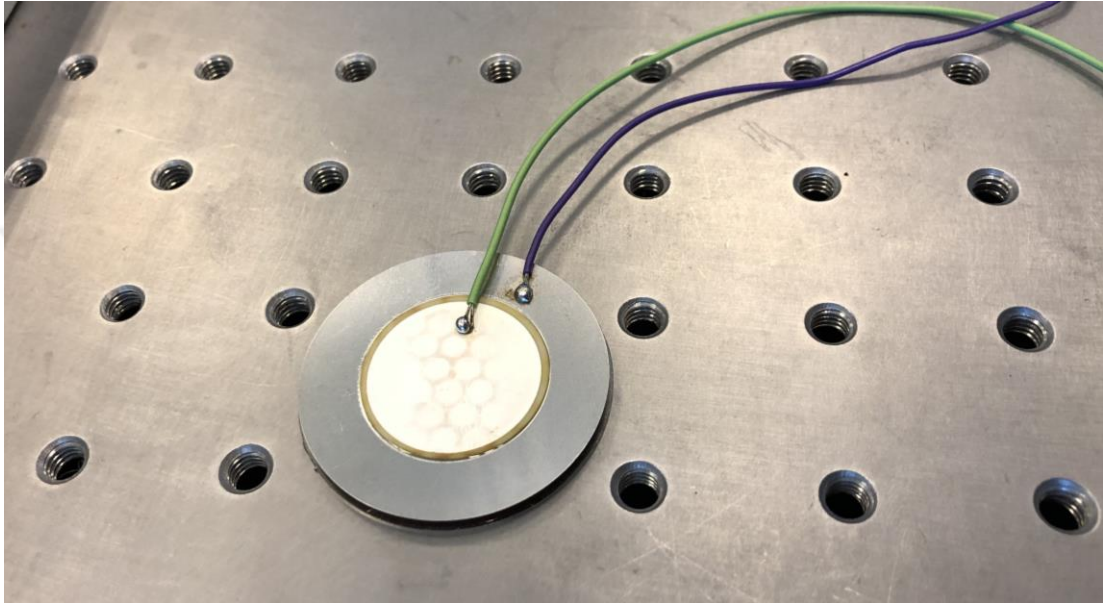


Figure 8: Synthetic jet actuator with soldered cables

Cables are introduced with same distance, to prevent deviation of resonant frequency in between jets. During testing, to prevent vibration related problems, strain relief mechanism is used. Strain relief mechanism is consisted of small fixtures that hold the cable with a small excess on the other end, preventing any tension to be applied at the piezo disk.

CHAPTER III

EXPERIMENTAL STUDY

To investigate and understand the mechanical and thermal behavior of synthetic jet actuators, experiments have been performed. This chapter provides information on the testing station, composed of two main test systems, namely laser vibrometer and heat transfer test system. Experimentation topology and improvements on the mechanical structure of the station is also provided. Later, two main experimental study on the synthetic jet actuators is presented, side orifice and central orifice jet actuators. Discussion of the experimental results is also provided in this chapter.

3.1 Test Setup

Mechanical and thermal behavior of synthetic jet actuators are measured in a testing system, composed of laser vibrometer and heat transfer modules, respectively. Testing station is designed to be used in the studies of Ikhlaq et al. ([13], [21]). Both systems are run by a computer with embedded controller, along with the data acquisition (DAQ) system. In both test types, synthetic jet actuators are driven by a wave form generator and the output signal from the wave generator is amplified by piezo amplifier. Figure 9 shows the topology of the experimental setup, along with the layout schematic of system components.

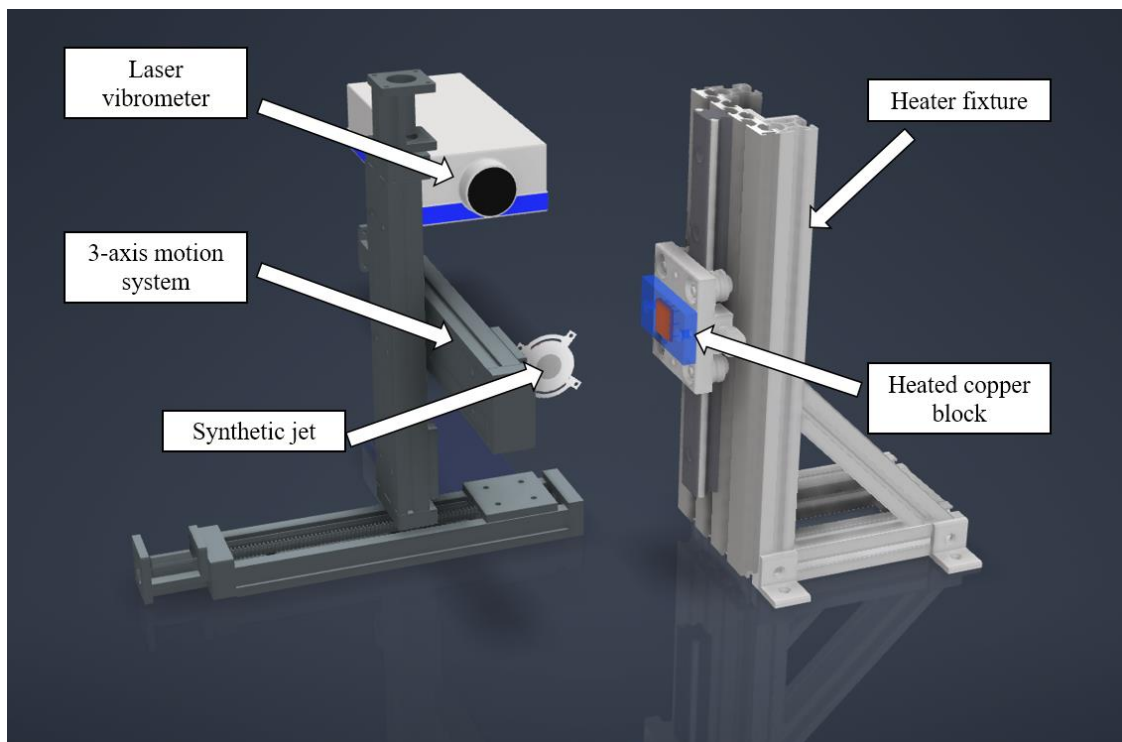
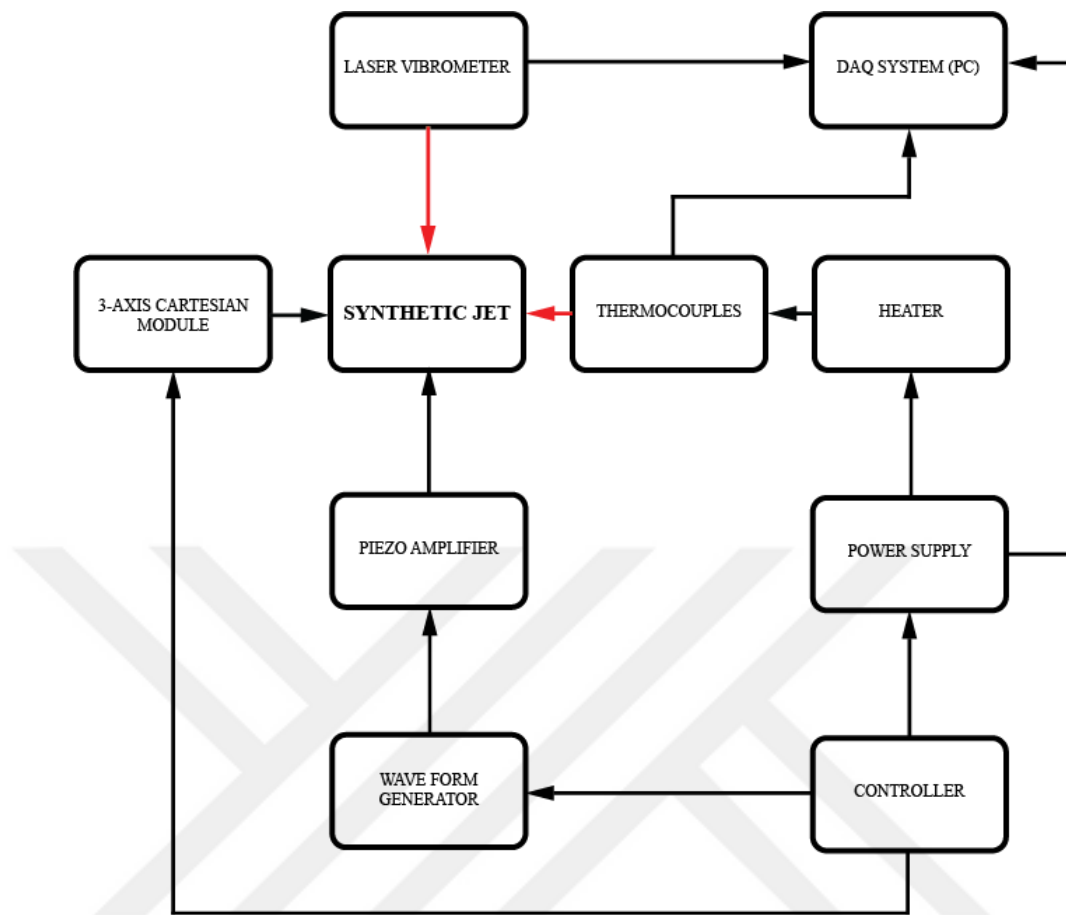


Figure 9: Synthetic jet testing system topology and layout

3.1.1 Laser Vibrometer Test System

For mechanical measurements, a laser vibrometer system is used. The system is composed of a laser head, a DAQ module, and a 3-axis cartesian linear system. Main principle of the laser vibrometer is to measure relative position of a point on the synthetic jet's piezo disk. Due to the data acquisition speed of up to 10 kHz, the system can measure displacement of vibrating objects with a frequency range of 0 to 1 kHz with great resolution. The objective of the thesis is to measure the displacement and velocity at which the disk displaces, from the center of the synthetic jet. To align the laser head to the center of the synthetic jet, a 3-axis motion system is used. Same device is also used for locating the focal point of the optical sensor of the laser head.

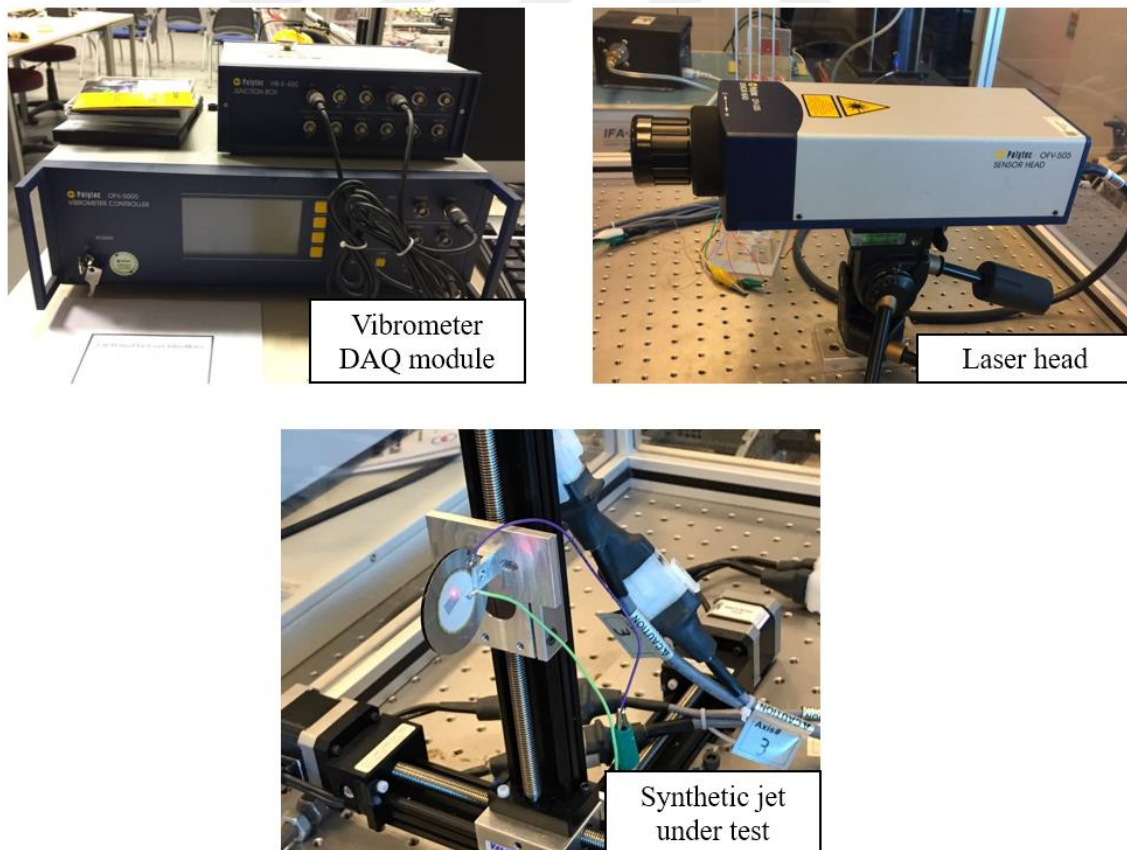


Figure 10: Laser vibrometer test setup with specimen under test

Readings from laser vibrometer tests are observed for determining the resonant frequency of the synthetic jet actuators, which is the point where the deflection of the piezo disk is the highest [13]. Later, mechanical model of the synthetic jet actuators are used for validation with the laser vibrometer readings. In this thesis, driving frequency range and thus the measured frequencies are between 100 – 1000 Hz.

3.1.2 Heat Transfer Test System

For measuring thermal performance, heat transfer test system is used. System is composed of a 1" by 1" (25 mm x 25 mm) Kapton heater mounted on a copper slab with same surface area, with a thickness of 1.5 mm (Figure 12). Whole configuration is mounted on a plexiglass substrate bulk material, having only one surface exposed to the atmosphere. The reason for this design is to achieve a thermally lumped system, having a uniformly distributed temperature. Heated copper is mounted with 4 pieces of T type thermocouple for validating the temperature distribution along the surface. Same design is used and validated by Ihklaq et al. [13].

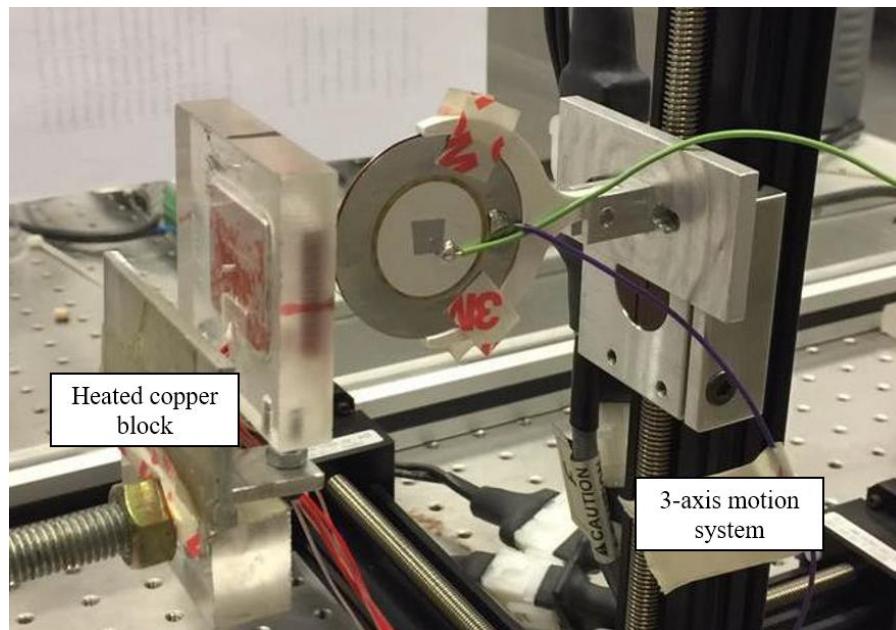


Figure 11: Heat transfer test setup

The main objective for the heat transfer test system is to measure the heat removed from the copper surface by measuring power consumption of power supply for the PID controlled temperature system. Firstly, to acquire the power consumption values for natural convection scenario, steady state conditions are achieved without driving the synthetic jet, having no cooling effect. Then, according to the boundary conditions, synthetic jet is driven and, the steady state conditions are achieved one more time, hence the difference of power required for preserving the copper temperature is obtained as heat removed from surface by the synthetic jet. Using the 3-axis linear motion system, synthetic jet is aligned and targeted to the copper surface. The nozzle-to-target distance, one of the parameters observed in this thesis, is adjusted by the linear motion system.

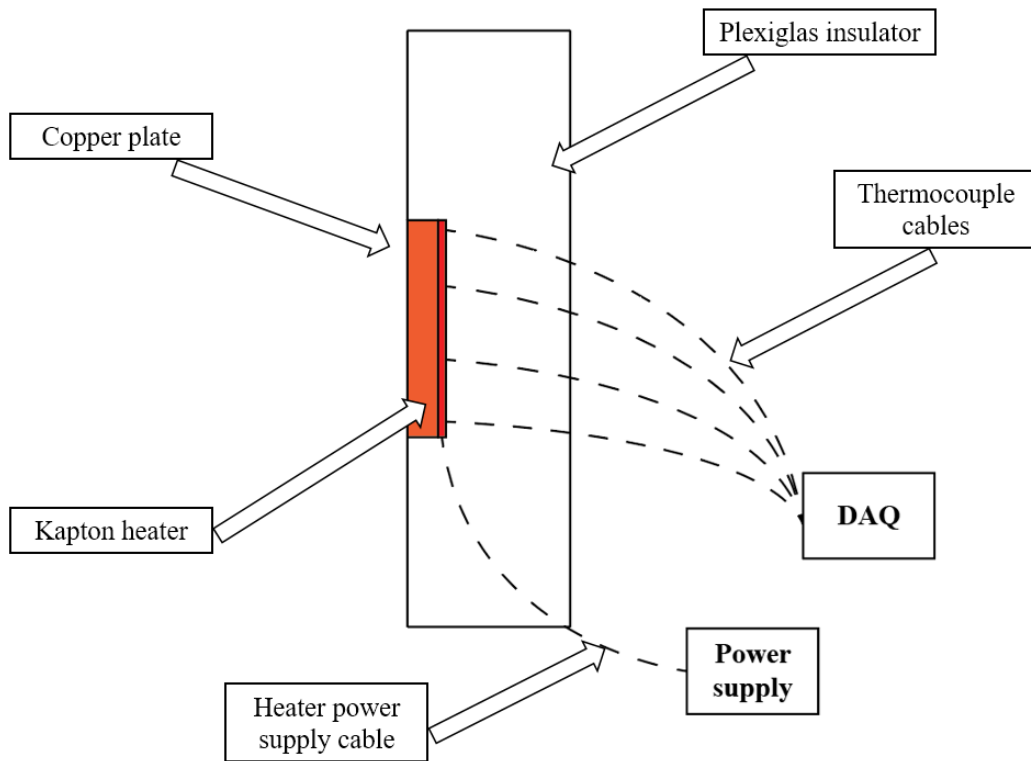


Figure 12: Schematic of heated copper block

Some upgrades have been implemented on the heat transfer system, especially on the fixture mechanism that holds the heated copper and plexiglass structure. Earlier version of the fixture was having misalignment issues, along with the parallelism issues, due to the mounting technique of the fixture to the plexiglass substrate. To improve, an aluminum skid is designed with a capability of vertical adjustability. The new fixture is consisted of a track roller bearing linear carriage that facilitates the positioning of the heated copper group. Furthermore, this mechanism allows the fixture to be fixed orthogonally, eliminating any parallelism issues. Following figure depicts the design of the heated copper fixture (Figure 13).

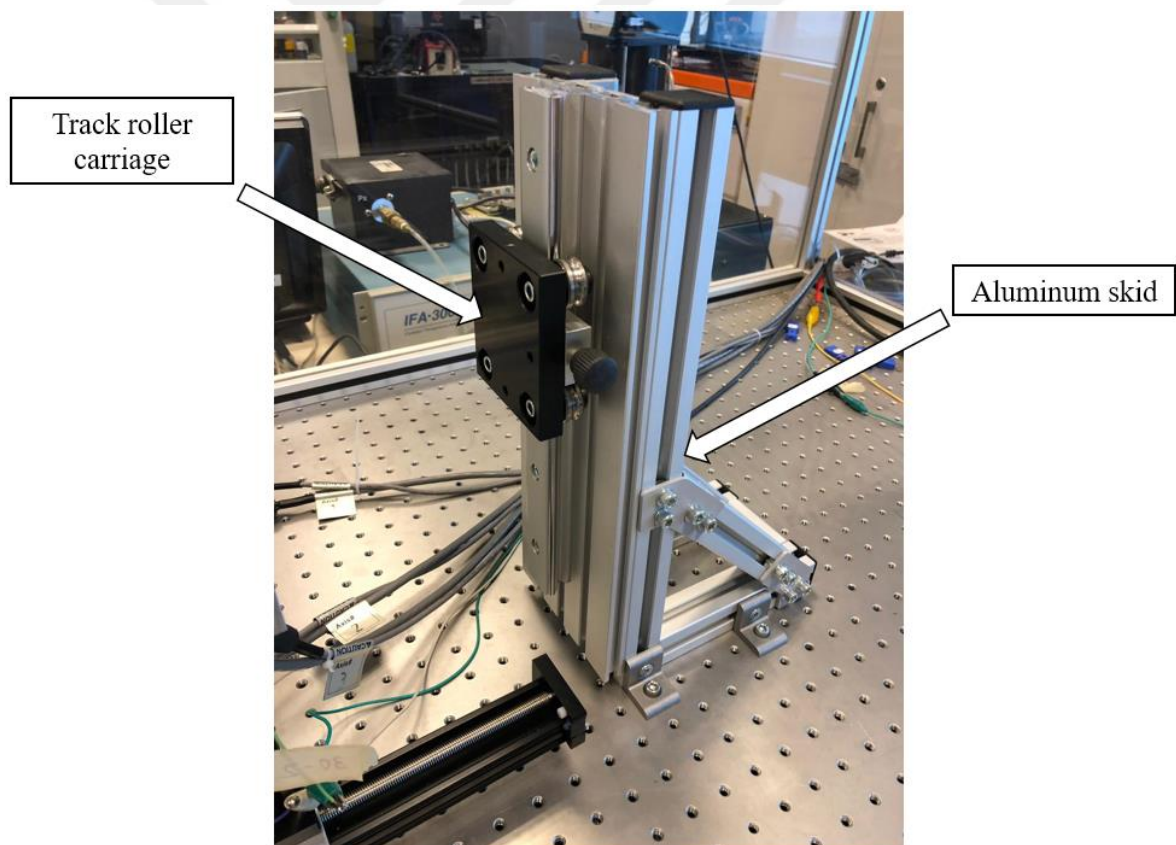


Figure 13: Heated copper adjustable fixture

3.1.3 Uncertainty Analysis

Synthetic jet actuators are operated by a cyclical excitation, meaning a continuous and a harmonic movement. This creates complications in terms of performance variations and uncertainties. As mentioned by Moffad [18], uncertainty analysis is an attribution to an estimation of the possible residual errors in an experimentation or calculation, and in order to accept the reliability of the data, uncertainty analysis must be performed.

To estimate the errors in Nu number calculations, Equation 1 is used. For heat transfer testing, Nu number is calculated to find the cooling performance for comparison.

According to the uncertainty values and their dependent measured values, manufacturing has selected as the first area to tackle, since the repeatability and sampling amount is one of the key points in uncertainties [18], with a respectable impact on results.

$$\delta Nu = [(a_1 \delta D_h)^2 + (a_2 \delta A_1)^2 + (a_3 \delta Q_s)^2 + (a_4 \delta k)^2 + (a_5 \delta (T_s - T_{amb}))^2]^{1/2} \quad (1)$$

As mentioned in Chapter I, a manufacturing rig has been designed in order to achieve a precision and repeatability between jets. Main control point for manufacturing precision is microscopic imaging. Applied wall dimensions are controlled for each production before assembly, to preserve mechanical constraints on the package (Appendix A). Finally, for plates used as central orifice plates and holder, are manufactured by CNC laser cutting technique. Positioning tolerance for this procedure affects hydraulic diameter D_h directly. Table 1 depicts the absolute and relative uncertainty rates, in terms of manufacturing accuracy.

Table 1: Uncertainty analysis for synthetic jet manufacturing

Parameter	Absolute Uncertainty	Relative Uncertainty
Wall thickness (mm)	± 0.1	0.5%
Distance between disks (mm)	± 0.05	1%
Disk parallelism ($^{\circ}$)	± 1	1%
Holder center alignment (mm)	± 0.1	0.5%
Hydraulic diameter (mm)	± 0.05	0.25%

For vibrometer tests, several aspects have been crucial and have substantial impact on experimental results (Table 2). First of them is the alignment of the laser. Readings change depending on the point where laser is pointed on the diaphragm of the synthetic jet. To minimize errors in alignment, fixture of the synthetic jet actuators have been mounted on the 3-axis linear motion system, with an open-loop motion controller, having a movement pitch accuracy of 0.5 mm. Secondly, orthogonal alignment of the laser head with synthetic jet reading surface also affects the readings. To prevent errors from misalignment, water gauge has been included to the vibrometer fixture.

Table 2: Uncertainty analysis for laser vibrometer test results

Parameter	Absolute Uncertainty	Relative Uncertainty
Central alignment (mm)	± 0.1	0.5%
Orthogonality ($^{\circ}$)	± 1	1%

For heat transfer tests, calculation tolerances play a role for uncertainties, such as power measurement accuracy, temperature measurement accuracy, voltage amplitude implemented on synthetic jet actuators, and so on. Moreover, from the mechanical positioning stand point, since synthetic jet actuators are fixed to the 3-axis linear motion system, position accuracy is the same as laser vibrometer tests, which is orifice-to-target distance.

Table 3: Uncertainty analysis for heat transfer test results

Parameter	Absolute Uncertainty	Relative Uncertainty
Temperature (°C)	± 0.5	1%
Power consumption (W)	± 0.05	0.5%
Voltage input (V_{pp})	± 0.01	0.5%
Orifice-to-target distance (mm)	± 0.1	1%

As a result, a total of 3.25% error rate for manufacturing, 1.5% for laser vibrometer measurements, and 3% for heat transfer testing is estimated. For each test type, manufacturing must be included because of the possible accumulation of errors, which is why a maximum of 6.25% for heat transfer tests, and a maximum of 4.75% error is estimated as experimental uncertainty in this study.

3.2 Side Orifice Study

This part of the study mainly focuses on the effect of orifice size, jet excitation amplitude, frequency, on the thermal behavior and cooling performance. Side orifice jets are defined as the cavity along the dispensed wall (Figure 14).

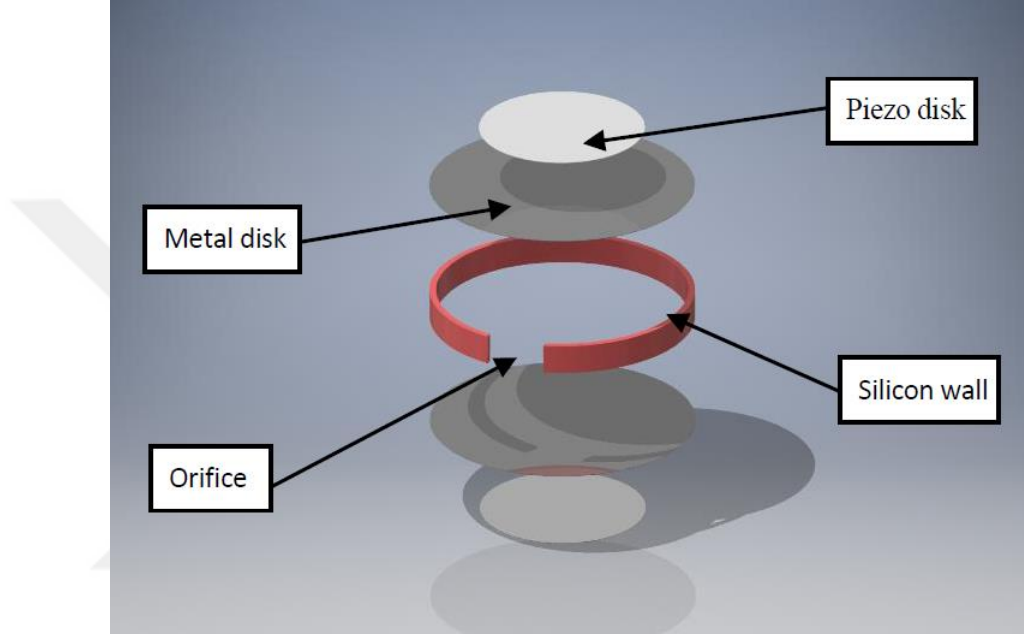


Figure 14: Exploded view of a synthetic jet with side orifice

To correlate heat transfer results with cooling performance, Nusselt number is calculated from the heat transfer test readings, as seen in equation (2).

$$Nu = \frac{h * L}{k} \quad (2)$$

Where

$$h = \frac{Q_{sj}}{A * (T_s - T_{amb})} \quad (3)$$

Where

$$Q_{sj} = Q_{heater} - Q_{loss} \quad (4)$$

3.2.1 Boundary Conditions

Synthetic jet actuators for this study have been constrained by the boundary conditions, in order to observe the effect of the parametric differences, such as orifice size. For that reason, most of the mechanical parameters have been selected and fixed for every type. Table 4 depicts the boundary conditions for the side orifice study. Figure 15 also shows the schematic for orifice size properties from top-section view of synthetic jet actuators. Apart from the parameters and boundary conditions given below, voltage amplitude to drive the synthetic jet actuators are also observed in a range of 15 – 40 Vpp.

Table 4: Boundary conditions table for the side orifice jet study

Parameter	Jet-1	Jet-2	Jet-3
Orifice angle (°)	10	30	60
Wall height (mm)	1.2	1.2	1.2
Jet diameter (mm)	41	41	41
Piezo disk diameter (mm)	20	20	20
Orifice-to-target distance (mm)	10	10	10

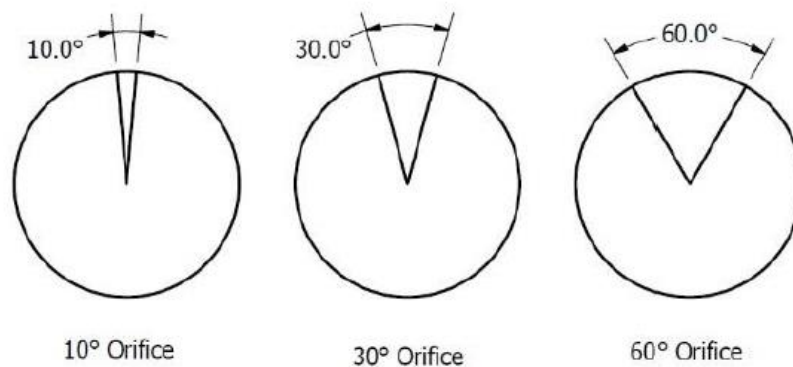


Figure 15: Orifice properties schematic for synthetic jet actuators

As mentioned in several studies ([19],[20], and [23]), length scale is one of the definitive parameters affecting the heat removal rates, due to the turbulence intensity of the flow being formed, air velocity decline, and inhalation of the warm air occurs ([48]). To that end, length scale calculations are also introduced in this study. Table 5 and Figure 15 depict the length scale parameters for the side orifice jet actuators.

Table 5: Length scale table for the side orifice jet study

Parameter	Jet-1	Jet-2	Jet-3
Orifice angle (°)	10	30	60
Wall height (mm)	1.2	1.2	1.2
Orifice length (mm)	3.44	10.22	19.75
Hydraulic diameter (mm ²)	1.78	2.15	2.26
Length scale, H / D_h	5.62	4.65	4.42

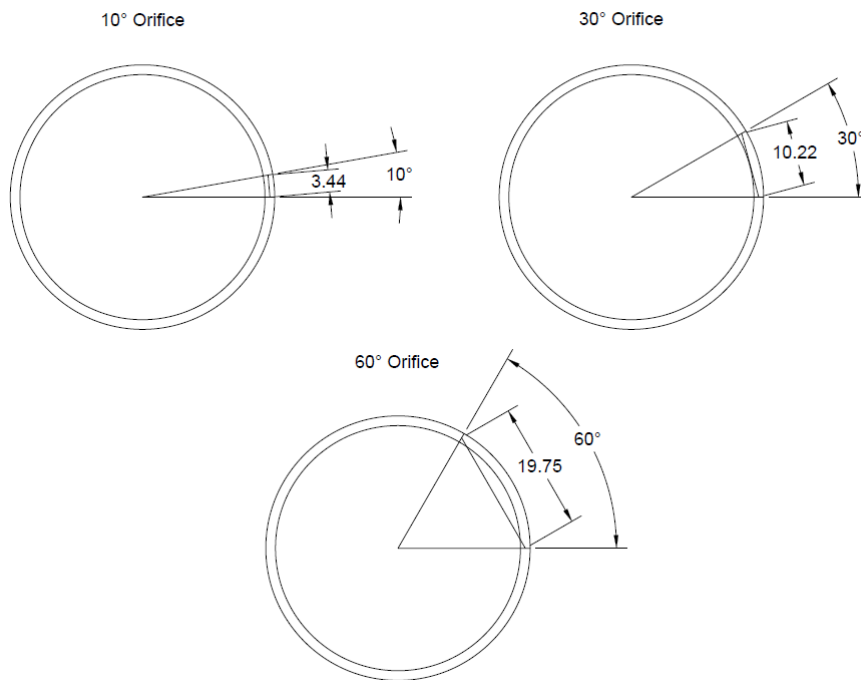


Figure 16: Orifice dimensions for side orifice jet study

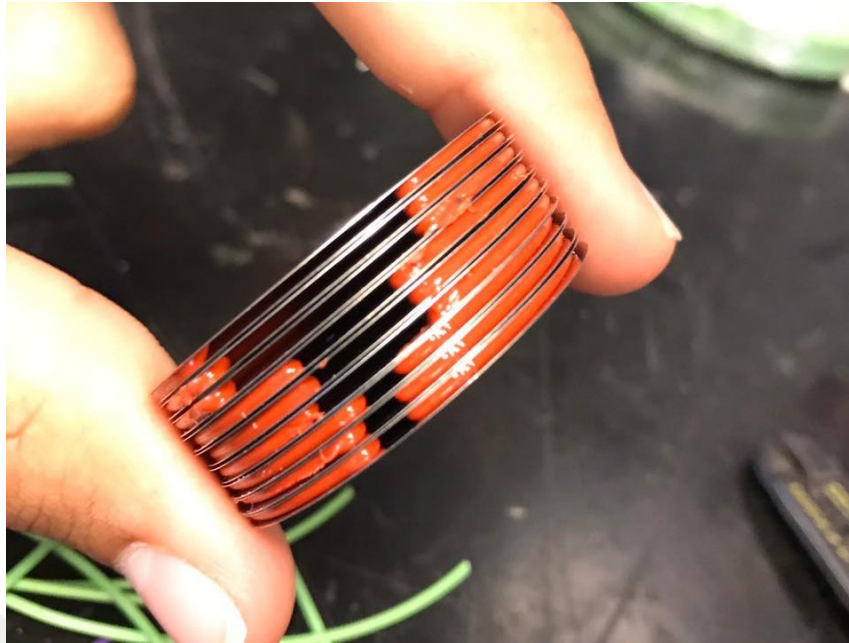


Figure 17: EVATEG side orifice synthetic jets after production

3.2.2 Vibrometer Test Results

Laser vibrometer tests have been performed for each set of the side orifice jets to understand the mechanical response of each jet actuator. For the displacement at the center of the piezo disk, below results are obtained (Figure 18).

For the 10° side orifice jet, a displacement of $60\text{ }\mu\text{m}$ and $79\text{ }\mu\text{m}$ is observed in a two-peak nonlinear slopes in frequency of 300 Hz and 750 Hz, respectively. Compared to other side orifice jets, 10° jets are observed to have their highest displacement at the second peak point, at 750 Hz. For the other two types, 30° and 60° jets, highest displacement has been observed at their first peak points, 400 Hz and 500 Hz, respectively. Comparing displacement values, it has been observed that the higher orifice angle jets have resulted higher displacement at the center of their diaphragm.

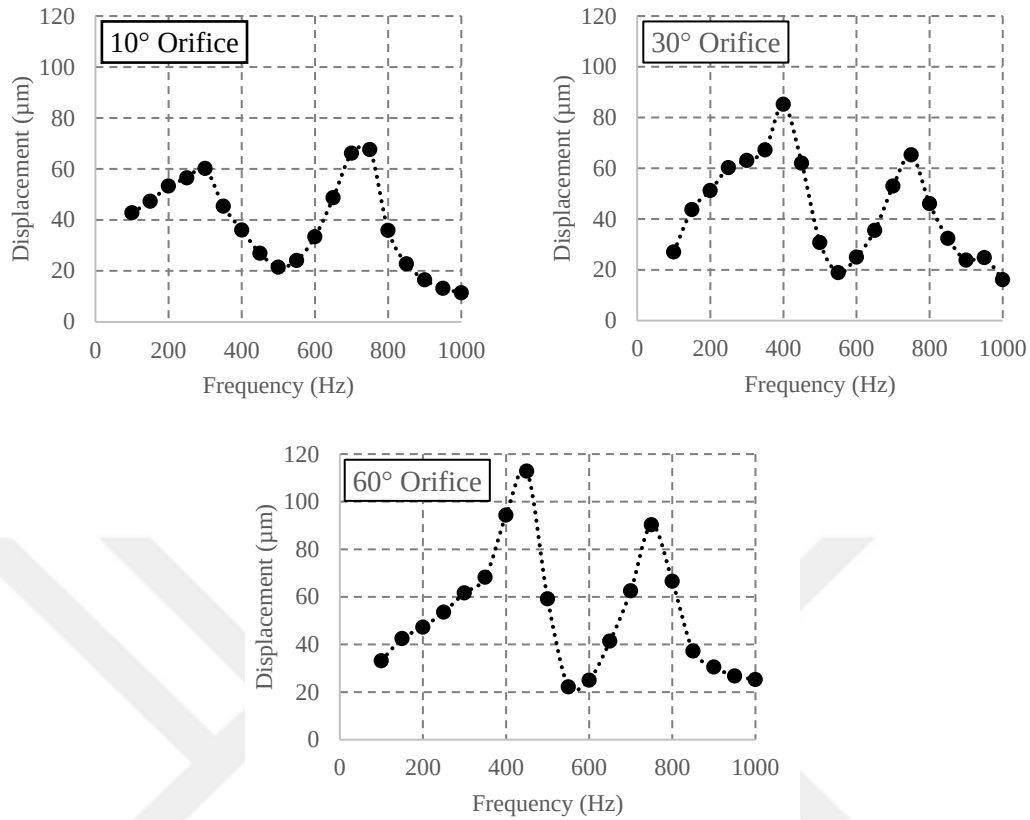


Figure 18: Effect of operating frequency over disk displacement

An increase of first peak displacement of 30% has been observed between 10° and 30° jets, while slight shift in frequency value, resulting 50 Hz difference. Similar differences has been observed between 30° and 60° jets. It is also observed that as the orifice angle increases, the frequency at which the peak displacement occurs, decreases. This can be the effect of the change in stiffness of the mechanical structure, resulting a shift in the frequency domain. First peak value for displacement has been observed in o frequency of 450 Hz, which is a shift of 50 Hz, compared to 30° jets. For the first peak values, increase in displacement for about 25% has been observed in 60° jets.

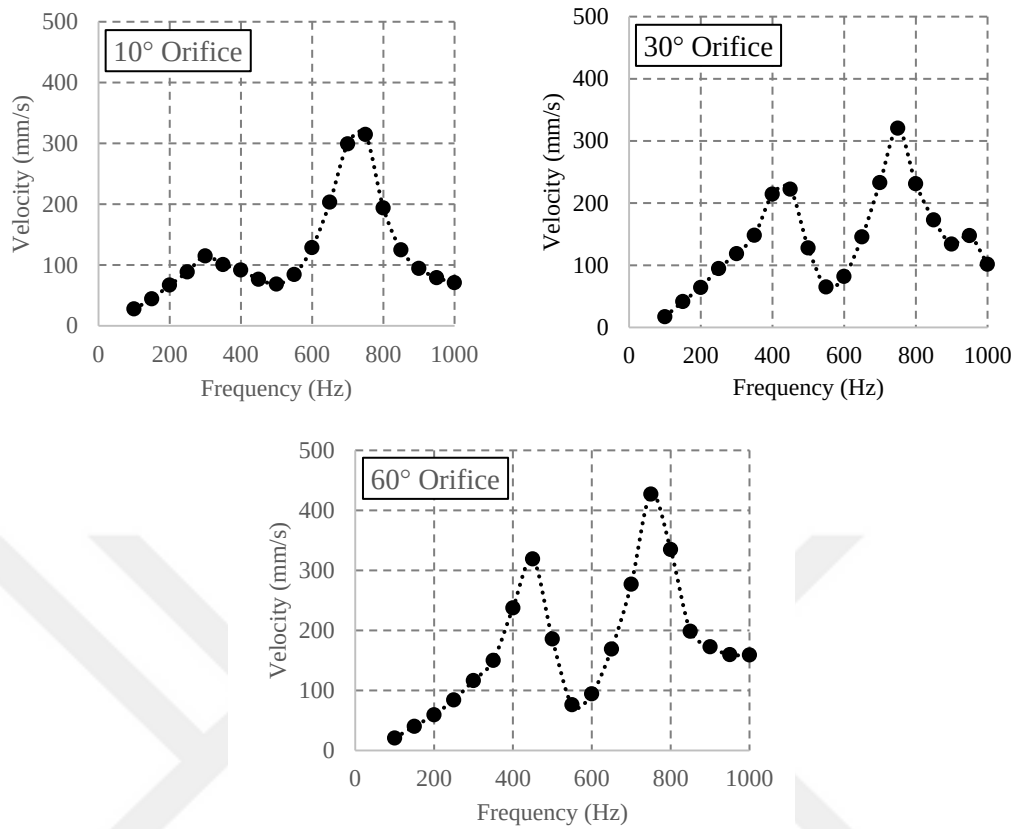


Figure 19: Effect of frequency over disk velocity

During the laser vibrometer measurements, velocity at which the synthetic jet piezo material displaces is also observed. Naturally, a correlation between velocity and displacement is observed (Figure 19). Note that the velocity measurements have been conducted at the center of the piezo disk.

Disk velocity results for 10° jets showed similar results as expected, two peak values have been observed at the same frequency values. A slight decrease in the first peak disk velocity has been observed. 30° orifice jets is observed to have higher disk velocity values on the opposite peaks, compared to the displacement results. Disk velocity is measured to be 28% higher than the first peak value on the 30° orifice jets, with 320 mm/s. 60° jets resulted to have the most elevated disk velocities at the center of the piezo disk, compared to other jets. Their disk velocity has been recorded as 427

mm/s at the second mode, which is at the corresponding frequency of 750 Hz, and 34% higher than the first peak at 450 Hz. For all of the jet results, it is observed that the peak displacements and disk velocities have been occurred at the same frequencies.



3.2.3 Heat Transfer Results

Cooling performance is indirectly measured using heat transfer test system. Using the same frequency interval as laser vibrometer measurements, Nu numbers have been obtained from Equation 2. Below are the heat transfer test results for the side orifice jets, along with the excitation voltage amplitudes (Figure 20).

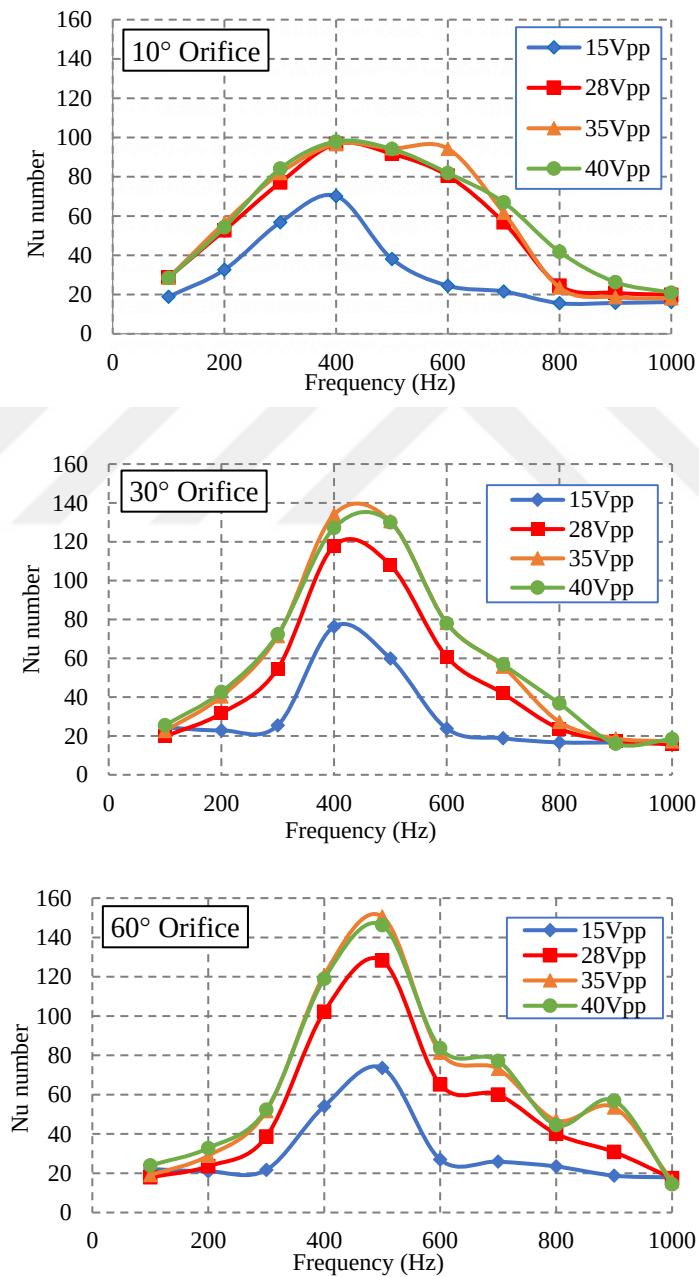


Figure 20: Effect of operating frequency of side orifice jets on Nu number

According to Nu values with respect to driving frequency, similar behavior of cooling performance is observed, compared to the laser vibrometer results. For the 10° side orifice jets, it is observed that Nu values increase substantially at the frequency where center displacement of piezo disk increases. Second peak frequencies show similar Nu values, and in between a slight decrease. For the outside of the first and second peak, a steep decrease in cooling performance is observed. Maximum Nu number for 10° jets is found to be 97.93.

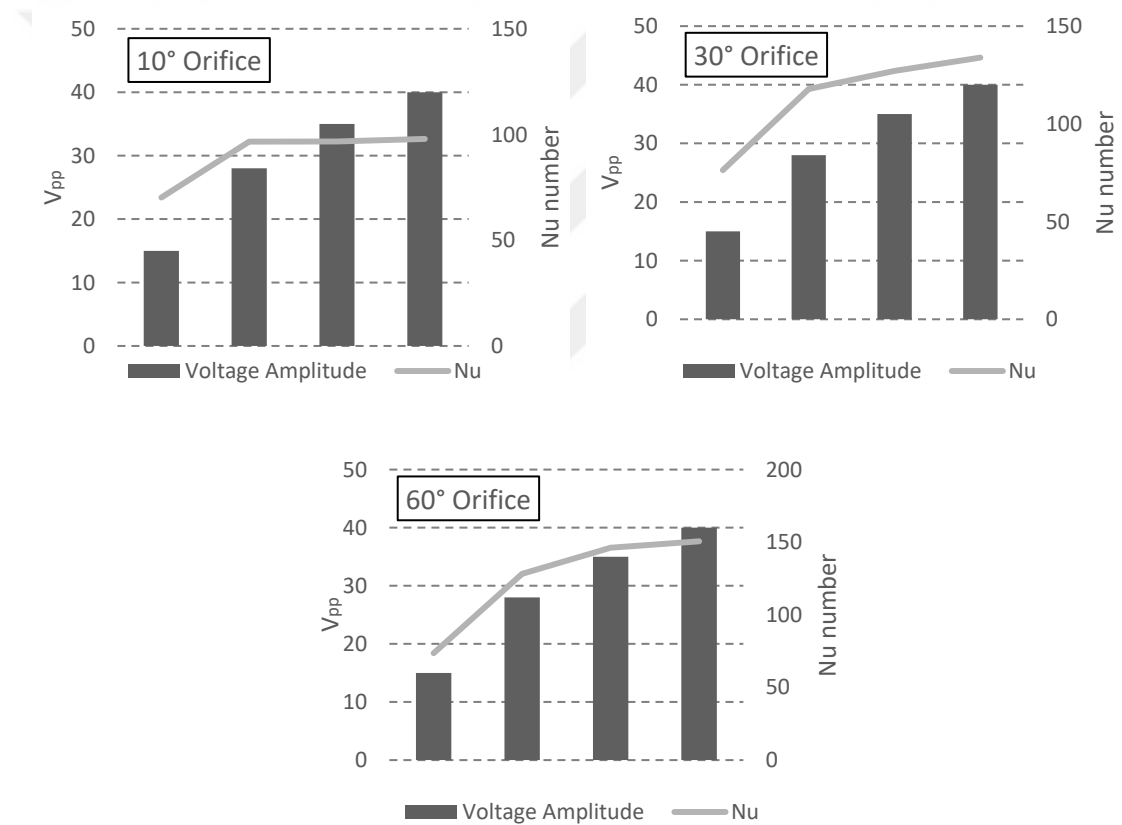


Figure 21: Impact of voltage amplitude on Nu number

30° side orifice jets resulted higher Nu number values compared to those for 10° jets. Moreover, at the second peak frequency, the displacement results showed little to none effect on cooling performance. Maximum Nu number for this type of jet is found to be 133.91. 60° side orifice jets behaved similarly with 30° jets. First peak

frequency values resulted the highest Nu number, with a maximum value of 150.62. Second peak frequencies showed negligible results, though the decline slope is less steep than 30° jets. Interestingly, a third peak in Nu number has been observed for 60° orifice jets. This can be the effect of the triggering of vortex formation and resulting a slightly higher heat removal rates ([32]).

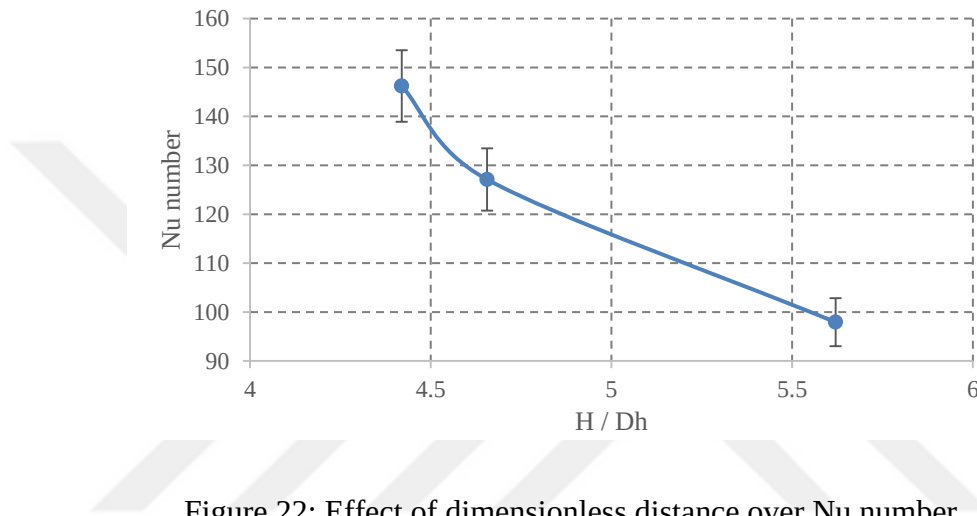


Figure 22: Effect of dimensionless distance over Nu number

Excitation voltage amplitudes have resulted similar results for every type of side orifice jet. In a set of points for voltage amplitude, namely 15, 28, 35, 40 V_{pp} , best cooling performance is obtained with highest amplitude. It must be noted that the performance rise from 15 to 28 V_{pp} has been found to be the highest increase, while a small difference in cooling performance has been observed between 28, 35 and 40 V_{pp} . Figure 21 depicts the effect of driving voltage amplitude on Nusselt number, cooling performance. The effect of voltage input on the synthetic jet cooling performance agrees with the study of Arik et al. [17]. They have observed similar behavior of cooling performance as a function of frequency, with a positive effect of voltage input on the Nu value. It is also observed that the coefficient of performance (COP) decreases at some point, which is why a balance must be determined with

voltage amplitude implemented on synthetic jets.

Comparing from the application based stand point, for cooling of a heated target with given operating and boundary conditions, it is observed that 60° side orifice jets provide the best cooling performance. To make a correlation between cooling performance and length scale, Figure 22 containing the results of Nu values as a function of length scale H/D_h is obtained. According to this result, it is observed that in the interval of 4 and 5 for a length scale, lowest value of H/D_h resulted the best cooling performance. Given that the side orifice jets have a rectangular cross-sectional area, the length scale range in the experimental results showed similar behavior as in the recent studies ([11], [23]).

3.3 Central Orifice Study

This study is based on a different type of synthetic jet actuators, central orifice jets. These jets are produced with a non-actuated disk (Figure 23), containing an opening or an orifice, generally aligned with the center of the actuated diaphragm. Main objective in this study is to determine the effect of orifice geometry on the cooling performance and nozzle-to-target distance providing best heat removal rates.

Manufacturing of the non-actuated synthetic jet actuators are performed by CO₂ laser cutting technique. To be able to achieve high dimensional accuracy, stainless steel material (AISI 304) is selected, due to its high finishing quality with CO₂ laser cutting.

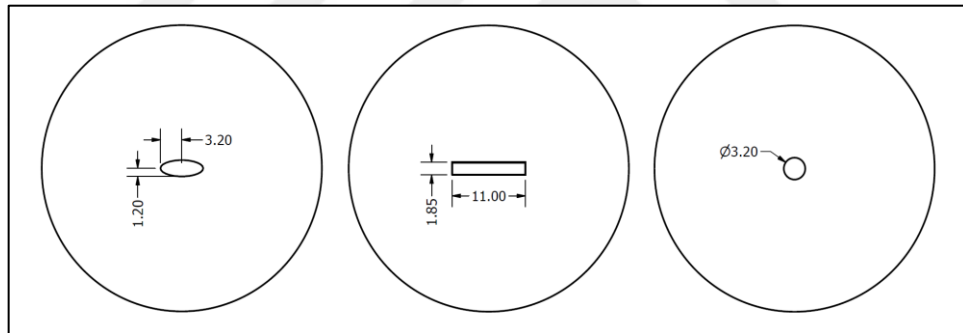


Figure 23: Non-actuated disk of central orifice synthetic jet actuators

3.3.1 Boundary Conditions

Geometric parameters of round, elliptic and rectangular orifices having similar hydraulic diameter is aimed in this study. Table 6 represents the boundary conditions of the orifice geometry. Hydraulic diameter calculation parameters are given in Eq. 5.

$$D_h = \frac{4 * A_0}{P_0} \quad (5)$$

Where A_0 is the area of the orifice of the central jet, and P_0 is the wetted perimeter. Furthermore, aspect ratio for the orifice geometry has also been taken into consideration, which is why two more orifice sizes have been introduced in the study. Other operating and boundary conditions are selected as previous study, current study manufacturing boundary conditions are stated in Table 7.

Table 6: Geometric details for the central orifice jets

Orifice Shape	Round	Rectangular	Ellipse	Rectangular	Ellipse
Radius (mm)	1,6				
Length (mm)		11,00	1,20	6,96	0,98
Width (mm)		1,85	3,20	1,16	2,61
Area, A_0 (mm ²)	8,04	20,35	12,06	8,07	8,04
Perimeter, P_0 (mm)	10,05	25,70	15,18	16,24	12,39
Hydraulic Diameter, D_h (mm)	<u>3,20</u>	<u>3,17</u>	<u>3,18</u>	1,99	2,59
$\sqrt{A}$ (mm)	<u>2,84</u>	4,51	3,47	<u>2,84</u>	<u>2,84</u>

Table 7: Manufacturing boundary conditions table for central orifice jet study

Parameter	Round	Ellipse	Rectangular
Wall height (mm)	12	12	12
Jet diameter (mm)	41	41	41
Piezo disk diameter (mm)	20	20	20

3.3.2 Vibrometer Test Results

Laser vibrometer measurements for the first batch, having similar traditional hydraulic diameter $D_h = 3.2\text{mm}$, are shown in the Figure 24. For all of the central jet actuators, there is a slight increase in displacement observed at driving frequency of 200 Hz. Also, it can be seen a decline of displacement in the interval of $200\text{ Hz} < f < 300\text{ Hz}$, for every actuator type.

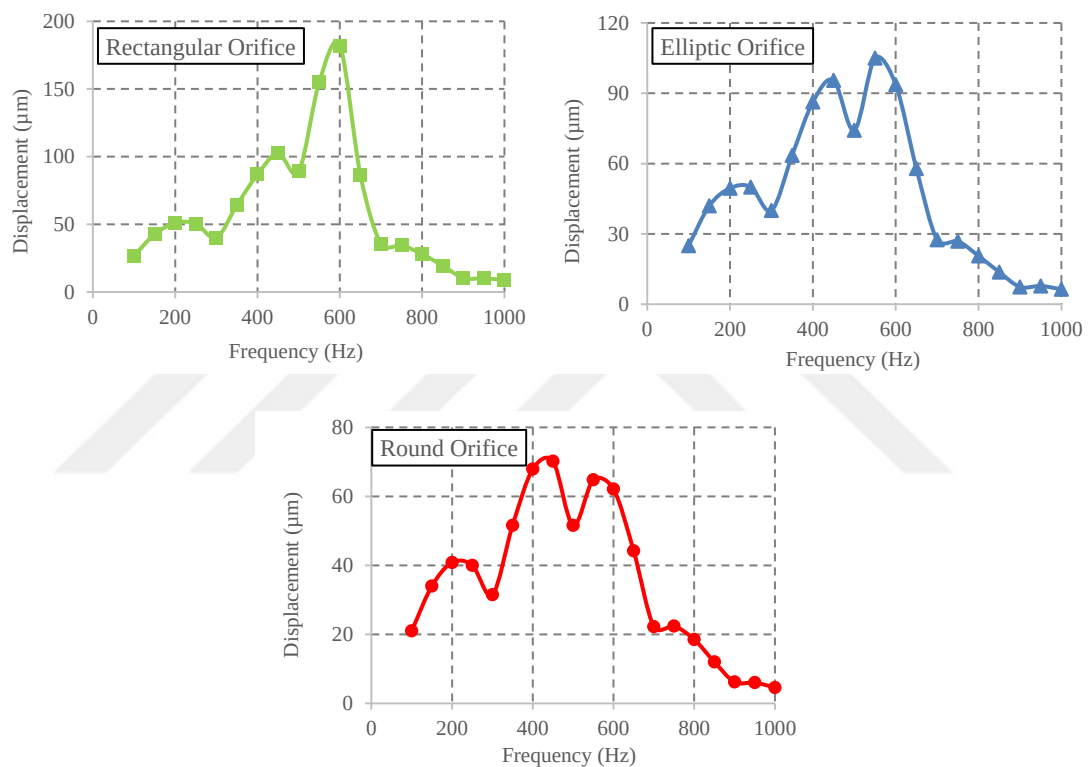


Figure 24: Effect of operating frequency over disk displacement

At peak points, round orifice jets have shown the smallest displacement values, in a maximum of $72\text{ }\mu\text{m}$ at driving frequency of 450 Hz . As for the elliptical and rectangular central jets, higher displacement values have been recorded. Especially rectangular orifice jet has shown displacement values up to $185.40\text{ }\mu\text{m}$. Elliptic orifice has its maximum displacement at the frequency of 550 Hz , as $109.10\text{ }\mu\text{m}$. In a frequency interval of $400\text{--}600\text{ Hz}$, a decrease in the displacement for every jet type

is observed. This shows a similar pattern with side orifice jet results.

3.3.3 Heat Transfer Results

Using the Eq. 1, 2, and 3, cooling performance of central orifice jet actuators have been measured using heat transfer test system. Orifice-to-target distance has been determined as $H = 10\text{mm}$. Figure 25 represents the results calculated from those tests. The highest heat transfer coefficient has been recorded at the frequency values where the disk displacement results have shown peaks. Lowest peak heat transfer coefficients have been observed from round shaped central orifice jet. Elliptic jet resulted almost 2 times higher Nu number, where rectangular has shown 2.25 times better cooling performance.

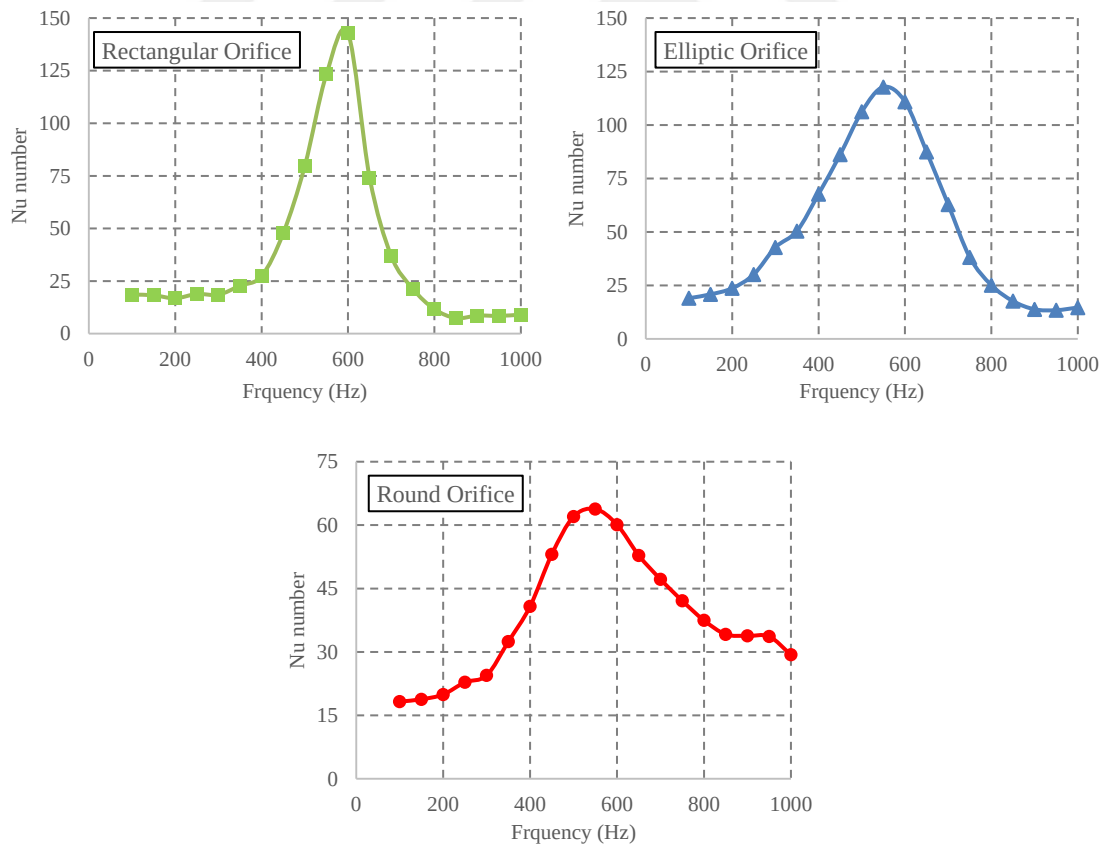


Figure 25: Effect of operating frequency over Nu for central orifice jets

According to the cooling performance measurements conducted by heat transfer test system, a correlation of hydraulic diameter between thermal performances was not reflected, as it can be misleading. Second batch of elliptic and rectangular orifice jets have been introduced to the study for the purpose of presenting an alternative length scale. Modified hydraulic diameter D_m has been determined in the following form:

$$D_m = \sqrt{A_0} \quad (6)$$

Apart from the Nusselt number calculations in a frequency domain, orifice-to-target distance is also varied in order to observe the effect of orifice geometry on the length scale H/D_m .

Even though it is difficult to calculate the air flow rate in synthetic jet actuators, some simplified estimations can be performed. Average velocity is calculated using the volume change rate of piezo diaphragm, obtained from laser vibrometer results. Since synthetic jet actuators are zero-net mass flux devices, at the displacement point, volume difference is considered as flow amount applied to the target. Average velocity and flow rate can be calculated as:

$$\dot{V} = \frac{4}{3}\pi A_o f \lambda D_d^2 \quad (7)$$

Where λ is the displacement measured at the center of the piezo disk.

$$u_{av} = \frac{\dot{V}}{A_o} = \frac{4}{3}\pi f \lambda D_d^2 \quad (8)$$

Non-dimensional calculated data at $H=10$ mm can be seen in Table 8. It can be seen that at the peak displacement points, rectangular orifice jet produces the highest heat transfer coefficient, which is 1.20 times higher than the coefficient value of round

orifice jet. According to the modified hydraulic diameter D_m , this fraction reaches approximately 2.50.

Non-dimensional values with $D_m=2.84$ mm and $H=8.2$ mm are presented in Table 9. When the conventional hydraulic diameter D_h is considered, cooling performance difference between rectangular and round orifice jets is 88%. With modified hydraulic diameter D_m , this value reduces to approximately 30%, and the highest Nusselt number is resulted by the elliptical orifice jet.

Table 8: Summary of findings, $H = 10$ mm

Orifice Shape	D_h (mm)	D_m (mm)	f_p (Hz)	λ (um)	$u_{o,av}$ (m/s)	Re_{Dh}	Re_{Dm}	St_{Dh}	St_{Dm}	Nu_{Dh}	Nu_{Dm}
Round	<u>3.20</u>	2.84	550	64.8	32.7	6965	6182	0.054	0.048	7.84	6.96
Rectangular	<u>3.17</u>	4.51	550	155.0	31.0	6545	9311	0.056	0.080	15.2	21.4
Rectangular	<u>3.17</u>	4.51	600	182.0	39.7	8383	11927	0.048	0.068	17.5	24.72
Ellipse	<u>3.18</u>	3.47	550	105.0	35.4	7513	8198	0.049	0.054	14.47	15.69

Table 9: Summary of findings, $D_m=2.84$ mm, $H = 10$ mm, ($H/D_m=2.88$)

Orifice Shape	D_h (mm)	D_m (mm)	f_p (Hz)	λ (um)	$u_{o,av}$ (m/s)	Re_{Dh}	Re_{Dm}	St_{Dh}	St_{Dm}	Nu_{Dh}	Nu_{Dm}
Round	3.20	<u>2.84</u>	600	79.1	43.6	9307	8248	0.044	0.039	7.00	6.21
Rectangular	1.99	<u>2.84</u>	600	57.5	31.6	4193	5984	0.038	0.066	8.18	11.68
Ellipse	2.59	<u>2.84</u>	600	61.2	33.6	5808	6369	0.046	0.062	9.20	10.08

Correlation of length scale with as a function of frequency can be made, comparing the orifice geometry variations. Table 8 represents for each geometry type, the change

in heat removal capacity, Nu , as a function of distance to target. According to the results in Figure 26, it can be stated that the elliptic and rectangular orifice jet actuators behave similarly in terms of distance to target. However, round orifice jet has shown different behavior by having low Nu values near target, and increased Nu values at around 14.5 mm. On the other hand, elliptic and rectangular orifice jet actuators resulted their maximum cooling performance at $H = 10$ mm.

Table 10: Summary of findings, $D_m=2.84$ mm, $H = 14.2$ mm, ($H/D_m=5$)

Orifice Shape	D_h (mm)	D_m (mm)	f_p (Hz)	λ (um)	$u_{o,av}$ (m/s)	Re_{Dh}	Re_{Dm}	St_{Dh}	St_{Dm}	Nu_{Dh}	Nu_{Dm}
Round	3.20	<u>2.84</u>	600	79.1	43.6	9307	8248	0.044	0.039	7.00	6.21
Rectangular	1.99	<u>2.84</u>	600	57.5	31.6	4193	5984	0.038	0.066	8.18	11.68
Ellipse	2.59	<u>2.84</u>	600	61.2	33.6	5808	6369	0.046	0.062	9.20	10.08

Since there is no correlation corresponding to every synthetic jet actuator used in this study, both conventional and modified hydraulic diameter, D_h and D_m , do not have an analytical relationship with cooling performance.

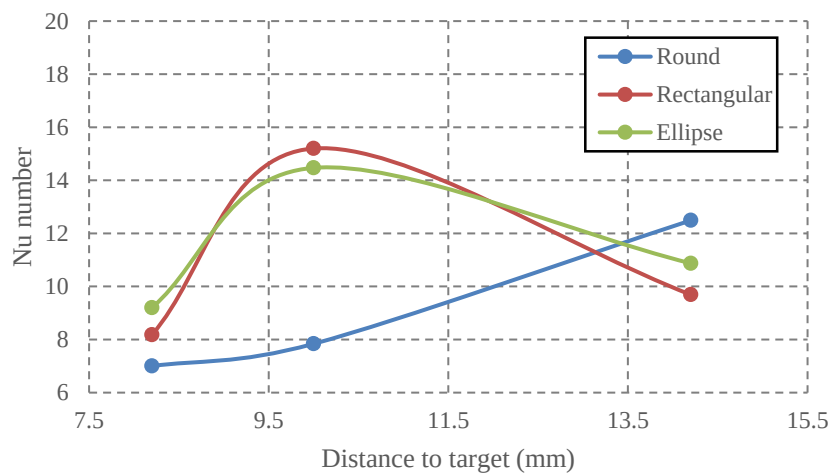


Figure 26: Variation of Nu number with jet-to-target spacing for various jets

CHAPTER IV

COMPARISON STUDY

This chapter provides an analysis and an experimental comparison of in-house manufactured EVATEG synthetic jet actuators with a commercial synthetic jet actuator and a mainly used alternative, axial cooling fan. Geometric and modal analyses of commercial synthetic jet are performed in order to understand the design criteria. Mechanical analyses are supplemented with oscilloscope measurements and a discussion of the findings are provided. Lastly, experimental comparison of synthetic jet and axial fan with EVATEG jet actuators is conducted.

4.1 Jet Comparison Study

This study focuses on the comparison of thermal performance achieved by recently manufactured synthetic jet actuators with a commercial actuator, a product by AAVID. Selected product is designed as a side orifice jet, with a rectangular disk geometry. CAD model is generated for the commercial model. Using the model, geometric and mechanical analysis on the material, resonant frequency and mode shape is performed on the commercial jet. Later, heat transfer test system is used for calculations of heat transfer coefficient.

4.1.1 Commercial Jet Actuator Analysis

AAVID jet actuator is implemented with holder mechanism to prevent any vibration related problems on the mounted surface. Three small extensions form wall material are used to connect the actuator with mounting kit. Both disks are mounted with piezo material, to increase the air flow in a cycle. Wall material is composed of silicon,

dispensed with a robotic dispenser. Orifice is designed to be a completely opened cavity on the edge of the actuator. Piezo disks are driven with a small electronic controller. A 5V DC input is required as power supply. Figure 27 depicts the synthetic jet actuator by AAVID.

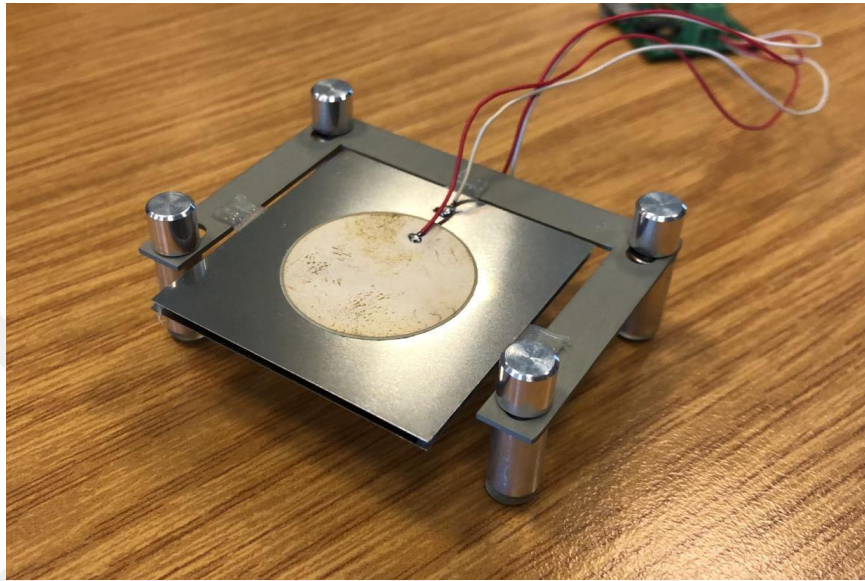


Figure 27: Synthetic jet actuator by a commercial vendor

To perform a geometric analysis and determine the resonant frequency of the mechanical structure, CAD model of the commercial synthetic jet is created. Using the mounting interface as the mechanically fixed constraint, modal analysis is performed.

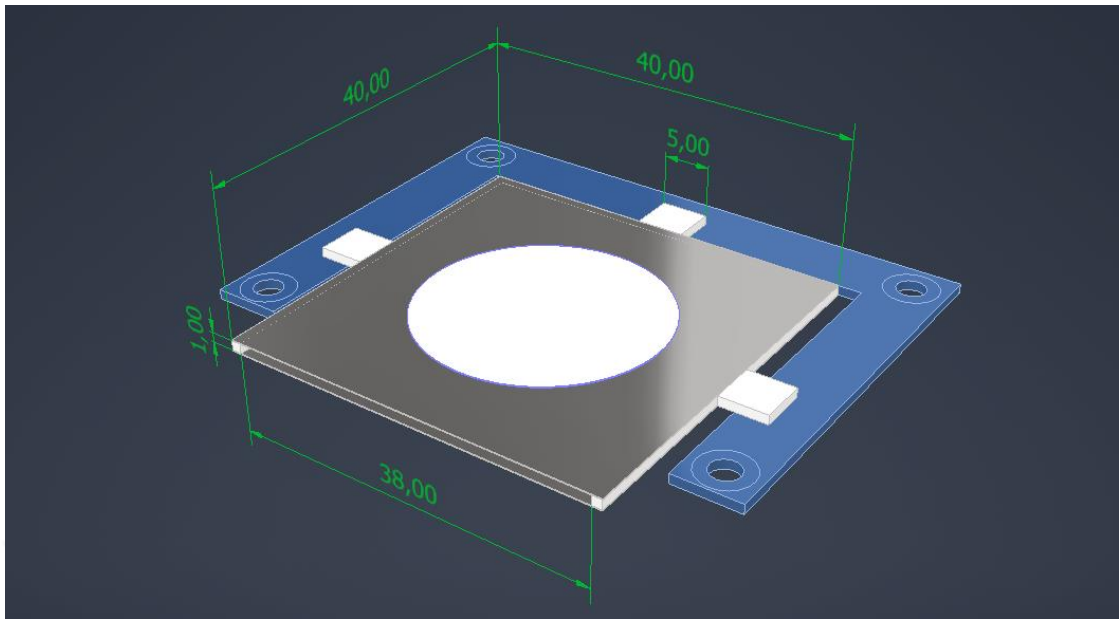


Figure 28: CAD view of synthetic jet actuator by a commercial vendor

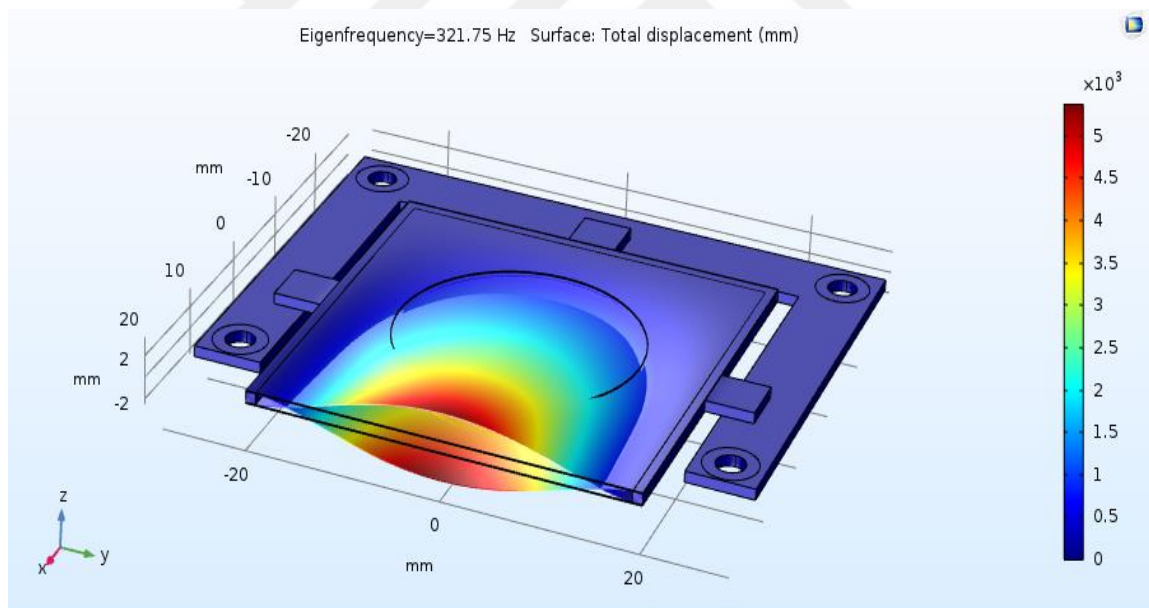


Figure 29: Mode of resonant frequency for the commercial jet actuator

Modal analysis solution has resulted that the mode shape has the displacement at the free edge of the actuator, rather than the center of the piezo disk. This mode shape behavior is different than the actuators used in the EVATEG lab, which is observed at the center of the piezo disk. The aim for this mode can be to decrease the resonant

frequency value, providing a lower noise at operation. It is observed that the mechanical structure has a resonant frequency at 321.75 Hz.

To obtain the driving parameters, oscilloscope readings have been performed. Driving parameters and their comparison with in-house manufactured jets are given in Table 11. To calculate peak-to-peak voltage and fall times for comparison, following equations are used:

$$FallTime = \frac{\arcsin(0.8)}{\pi * f} = \frac{0.2952}{f} \approx 0.3 * T \quad (9)$$

$$V_{pp} = V_{rms} * 2\sqrt{2} \quad (10)$$

Where f is the driving frequency and T is the signal period.

Table 11: AAVID and in-house synthetic jet actuators driving parameters

Parameter	Driving Frequency (Hz)	Fall Time (ms)	RMS Voltage (V_{RMS})	Duty Cycle	Peak-to-Peak Voltage (V_{pp})
AAVID Jet	194.9	1.486	25.0	49.32%	70.71
Side Orifice Jet	450-500	0.656-0.590	12.38	~50%	35.00
Central Orifice Jet	550-600	0.537-0.492	12.38	~50%	35.00

It is observed with oscilloscope readings that the AAVID synthetic jet is driven with less frequency than in-house manufactured jet actuators. This is expected due to the rectangular shape of the disks, along with the thin layer of silicone wall. Lower driving frequency has been a goal for synthetic jet actuators, because of the acoustic comfort required in practice examples ([63]). As for the driving voltage amplitudes, AAVID jets have 2 times the peak-to-peak voltage, compared to side orifice and central orifice jet actuators. This can be a result of the dual-actuated diaphragm disks

for AAVID jets, where other jets have been actuated with only one diaphragm. Interestingly, driving frequency of the jet is much lower than the resonant frequency of the mechanical structure. The difference of resonant frequency is found as 127 Hz.

4.1.2 Heat Transfer Results

To compare thermal performances of the jet actuators, heat transfer test system is used. Below is the operating conditions for the heat transfer tests (Table 12).

Table 12: Heat transfer test operating conditions

<i>Orifice-to-Target Distance (mm)</i>	<i>Copper Temperature (°C)</i>	<i>Steady State Temperature Tolerance (°C)</i>	<i>Steady State Duration (min)</i>
10	40	±0.5	3

For the comparison, 60° side orifice, rectangular and elliptic central orifice jet actuators are selected. These products resulted the best cooling performances in their corresponding study. Nusselt number in best frequency values are compared in the following figure (Figure 30).

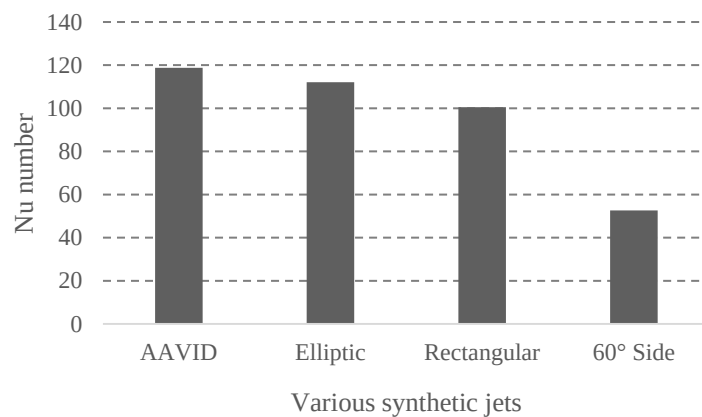


Figure 30: Experimental Nu numbers for various synthetic jets

According to the results, AAVID synthetic jet actuator performed the best, comparing

with other in-house manufactured actuators. 60° side orifice has resulted the lowest heat removal rates from the target surface, approximately 50% of the AAVID actuator. The reason for this is the dual disk excitation of the AAVID jet, while 60° jet is driven from only one piezo disk. Performance values for this manner agree with earlier studies performed on axisymmetric jets ([46], [61]). Elliptic and rectangular central orifice jets performance were close to each other, with around 10 to 15% lower Nu number compared to AAVID. Interestingly, elliptic central orifice jet performed slightly better than rectangular orifice jet, while previous study indicated otherwise.

4.2 Axial Fan Thermal Performance Comparison Study

Synthetic jet actuators are considered as advanced cooling alternatives to axial fans, especially in mesoscale cooling. This study aims to compare synthetic jet actuators with a conventional axial fan (Figure 31), in terms of cooling performance and length scale. Heat transfer tests and infrared camera temperature readings have been performed, using an LED chip as the cooling target.

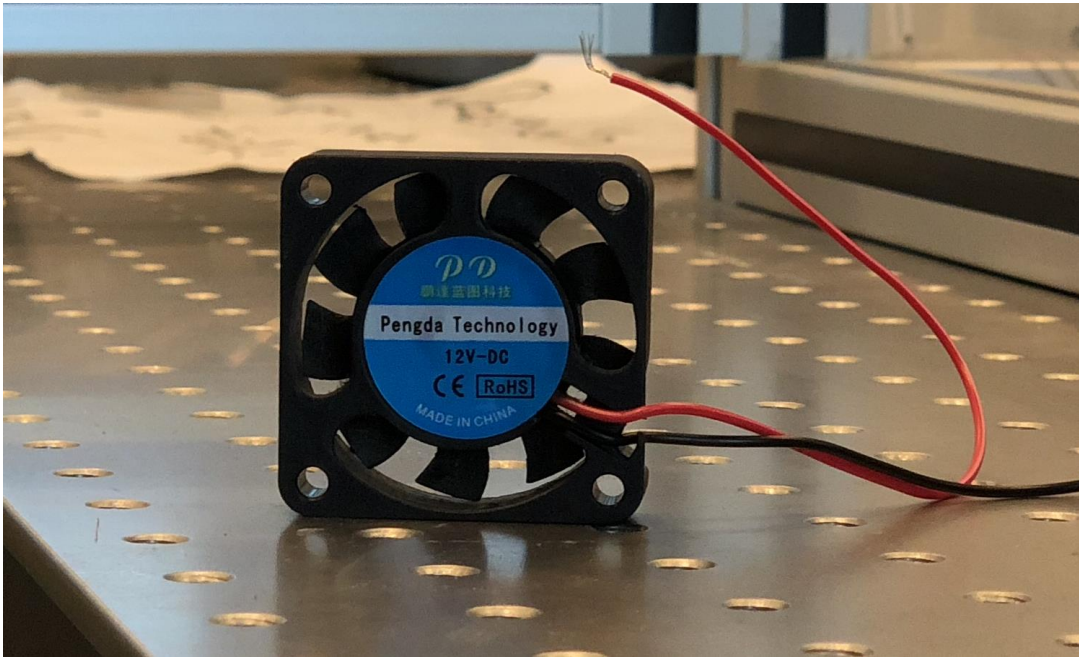


Figure 31: Axial cooling fan used in performance tests

4.2.1 Operating Conditions

Synthetic jet actuators to be compared are selected from the central orifice study, namely the elliptical orifice jet has been determined. Heat transfer tests are performed with the PID controlled heater system, same as the previous studies. As for the LED chip, a single chip green LED has been used. Below are the operating conditions for the study (Table 13).

Table 13: Operating conditions for thermal performance comparison study

<i>Operating Conditions</i>	Fan Diameter (mm)	Fan Power (W)	LED Chip Power (W)	SJA Voltage Amplitude (V_{pp})	Copper Plate Set Temperature ($^{\circ}\text{C}$)
	37.50	1.032	0.64	28	40

4.2.2 Analytic Solution

To validate results for all of the cases, thermal resistance network calculation and modeling of heat transfer test setup mechanical construction are performed. Below is the thermal resistance network representative schematic, taking heater, copper slab and plexiglass substrate into consideration.

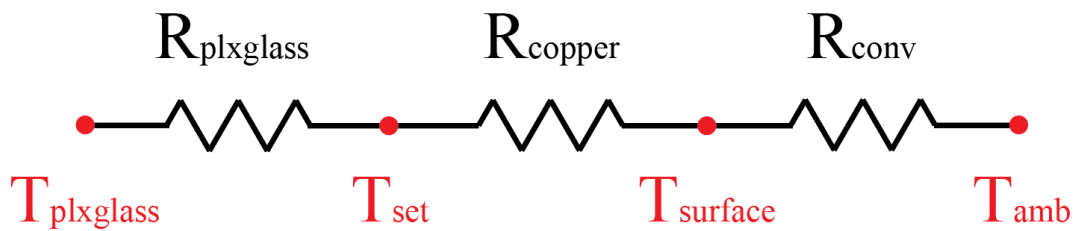


Figure 32: Simplified thermal resistance network for the test structure

Thermal resistance network system is implemented into Engineering Equation Software (EES), to observe its effect on parameters. Figure 32 represents the parameters and equations used in EES. Heat loss with radiation is neglected in the analytical solution.

Variables	Equations
$a_{\text{plexiglass}} = 0,05$ [m] width of the plexiglass	$R_{\text{plexiglass}} = \frac{t_{\text{plexiglass}}}{k_{\text{plexiglass}} \cdot \text{CROSS}_{\text{plexiglass}}}$ thermal resistance of plexiglass
$b_{\text{plexiglass}} = 0,05$ [m] height of the plexiglass	$R_{\text{copper}} = \frac{t_{\text{copper}}}{k_{\text{copper}} \cdot \text{CROSS}_{\text{copper}}}$ thermal resistance of copper block
$t_{\text{plexiglass}} = 0,01$ [m] thickness of the plexiglass	$R_{\text{conv}} = \frac{1}{h \cdot \text{CROSS}_{\text{copper}}}$ convection resistance
$\text{CROSS}_{\text{plexiglass}} = a_{\text{plexiglass}} \cdot b_{\text{plexiglass}}$ cross section area of the plexiglass	$R_{\text{total}} = R_{\text{plexiglass}} + R_{\text{copper}} + R_{\text{conv}}$
$k_{\text{plexiglass}} = 0,19$ [W/m-K] thermal conductivity of the plexiglass	$T_{\text{set}} - T_{\text{surface}} = R_{\text{copper}} \cdot Q_{\text{heat}}$ T_{surface} is surface temperature
$a_{\text{copper}} = 0,025$ [m] width of the copper block	$h = \frac{Q_{\text{net}}}{\text{CROSS}_{\text{copper}} \cdot (T_{\text{set}} - T_{\text{amb}})}$ heat transfer coefficient formula
$b_{\text{copper}} = 0,025$ [m] height of the copper block	$Nu = h \cdot \frac{D_h}{k_{\text{air}}}$ Nusselt number formula
$t_{\text{copper}} = 0,0015$ [m] thickness of the copper block	
$\text{CROSS}_{\text{copper}} = a_{\text{copper}} \cdot b_{\text{copper}}$ cross section area of the block	
$k_{\text{copper}} = 387,6$ [m] thermal conductivity of the copper	
$k_{\text{air}} = 0,024$ thermal conductivity of air	
$D_h = 0,0317$ [m] Hydraulic diameter of Jet EVATEG	
$T_{\text{set}} = 40$ set temperature	
$T_{\text{amb}} = 27$ ambient temperature	
$Q_{\text{heat}} = 1,487$ heat input	
$Q_{\text{loss}} = 0,024 \cdot (T_{\text{set}} - T_{\text{amb}}) - 0,0056$ loss heat-correlation developed by EVATEG	
$Q_{\text{net}} = Q_{\text{heat}} - Q_{\text{loss}}$ net heat	

Figure 33: Analytic model parameters and equations used in this study

Total resistance of the heat transfer test structure is calculated as 32.07 m²K/W. Plexiglass structure, with a thermal resistance of 21.05 m²K/W, resulted the highest thermal resistance. Following figure depicts the results from the analytic solution (Figure 34).

$a_{\text{copper}} = 0,025$ [m]	$a_{\text{plexiglass}} = 0,05$ [m]	$b_{\text{copper}} = 0,025$ [m]	$b_{\text{plexiglass}} = 0,05$ [m]
$k_{\text{air}} = 0,024$	$k_{\text{copper}} = 387,6$ [m]	$k_{\text{plexiglass}} = 0,19$ [W/m-K]	$ν = 191,9$
$R_{\text{copper}} = 0,006192$	$R_{\text{plexiglass}} = 21,05$	$R_{\text{total}} = 32,07$	$T_{\text{amb}} = 27$
$\text{cross}_{\text{copper}} = 0,000625$	$\text{cross}_{\text{plexiglass}} = 0,0025$	$D_h = 0,0317$ [m]	$h = 145,3$
$Q_{\text{heat}} = 1,487$	$Q_{\text{loss}} = 0,3064$	$Q_{\text{net}} = 1,181$	$R_{\text{conv}} = 11,01$
$t_{\text{copper}} = 0,0015$ [m]	$t_{\text{plexiglass}} = 0,01$ [m]	$T_{\text{set}} = 40$	$T_{\text{surface}} = 39,99$

Figure 34: Analytic results on the heat transfer

Correlation between Nu number and heat transfer coefficient in terms of heat input to the system can be seen in Figure 35. A linear relationship is observed.

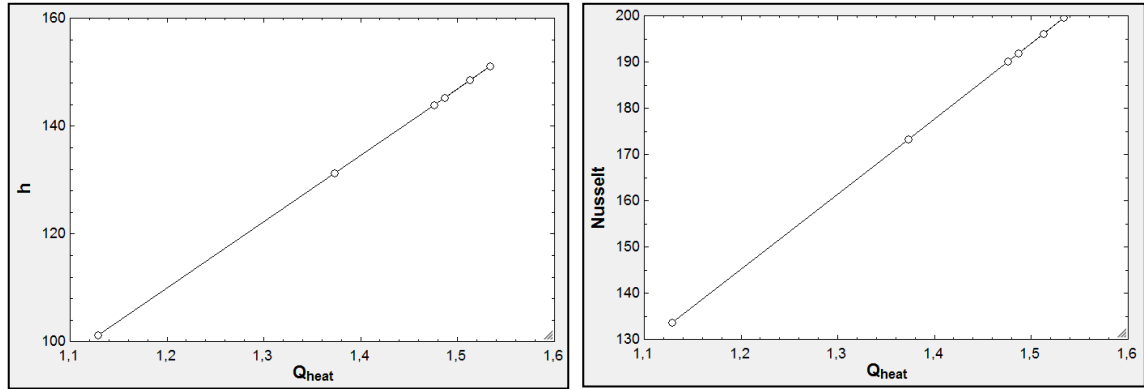


Figure 35: Nu number and heat transfer coefficient variation with heat input

4.2.3 Numerical Analysis

Analytical solution represents a one-dimensional heat loss, which is not a sufficient method in real life experiments. To supplement the data, numerical analysis solving three-dimensional models has been performed. CAD data is created and implemented into ANSYS Icepak software, for steady state thermal simulation. In order to present reliable results with efficient solving durations, mesh study is conducted. Mesh convergence study is presented in Figure 36.

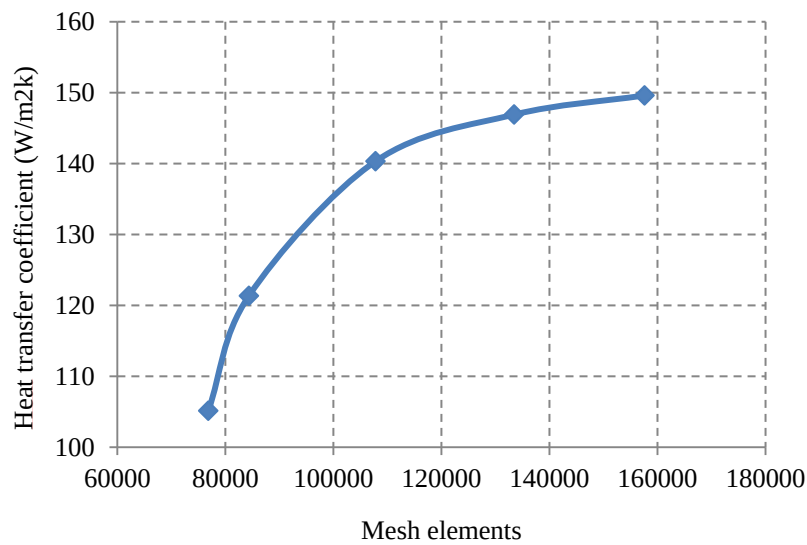


Figure 36: Mesh convergence study

According to the convergence graph, it can be seen that the resulting value is saturated, starting from a mesh element number of 13,000, creating negligible differences. Element number of 14,000 is selected for this study.

For the numerical simulation, three cases were to be run: Natural convection, fan cooling, and synthetic jet cooling. Simulation results at the steady state conditions are shown in Figures 37, 38, and 39.

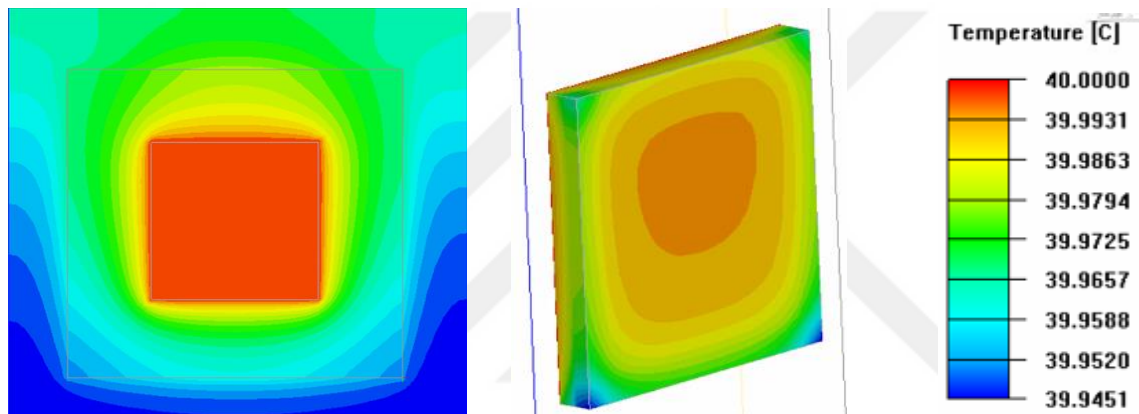


Figure 37: Natural convection simulation results

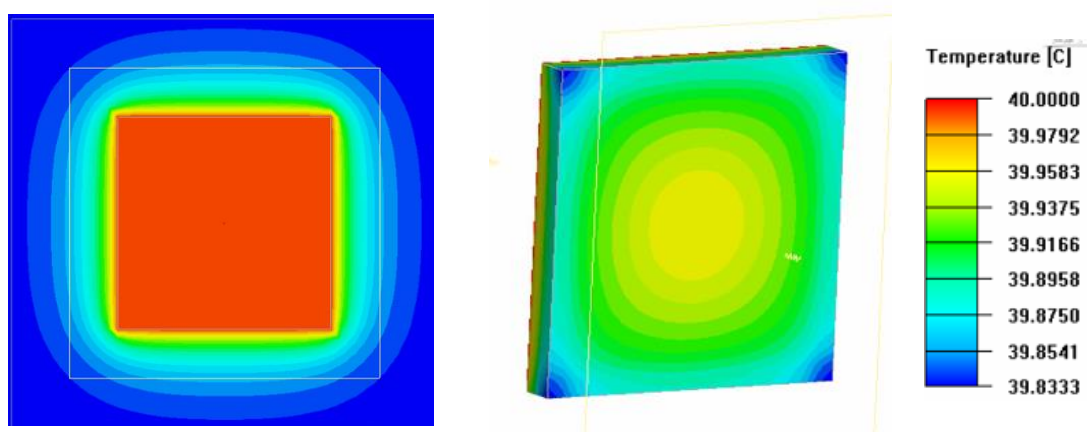


Figure 38: Axial fan cooling simulation results

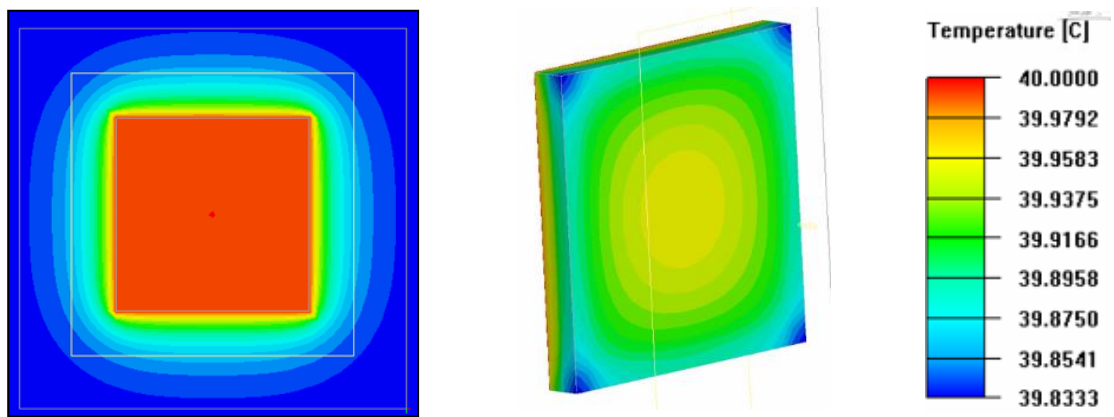


Figure 39: Synthetic jet cooling simulation results

According to the results from the numerical simulation and analytic calculation, axial cooling fan is observed to provide 4% more heat transfer coefficient, with a relative error of 0.4%. As for the heat flux for both cases, similar difference is obtained. Numeric results and relative error data is provided in Table 14.

Table 14: Numeric simulation result and relative error table

Parameter	Maximum Heat Transfer Coefficient (W/m ² K)	Relative Error (%)	Maximum Heat Flux (W/m ²)	Relative Error (%)
Natural Convection	14.2	--	252.1	--
Axial Fan Cooling	140.9	0.4	140.3	9.2
Synthetic Jet Cooling	134.8	2.3	131.6	4.8

4.2.4 Experimental Results

Numerical solution resulted that the axial fan performance is slightly better than synthetic jet cooling. Note that the length scale H/D_h is not implemented into calculations. To expand on that point of view, experimentation is performed. 2 cases of experiments have been studied: heat transfer tests, and infrared camera (IR) readings on an LED chip while cooling. After heat transfer testing, the obtained data

has been sorted out and the below Nu number results (Figure 40) have been obtained.

Furthermore, the effect of length scale is observed by varying the distance to cooling target H , with an interval of 5 mm to 30 mm (Figure 41).

Nusselt number calculations proved that the axial fan performed slightly better than synthetic jet actuator at $H = 5$ mm with 16% more heat transfer coefficient. The peak Nu value for the axial fan, at $H = 10$ mm, Nu value was 34% more than the synthetic jet. However, as the distance to target increased, synthetic jet actuator resulted higher values, and at $H = 20$ mm, both axial fan and jet actuator resulted the same Nu values. Moreover, after that point, as the distance is increased, axial fan cooling performance dropped significantly, while the synthetic jet actuator has resulted small decline and performed better than the axial fan.

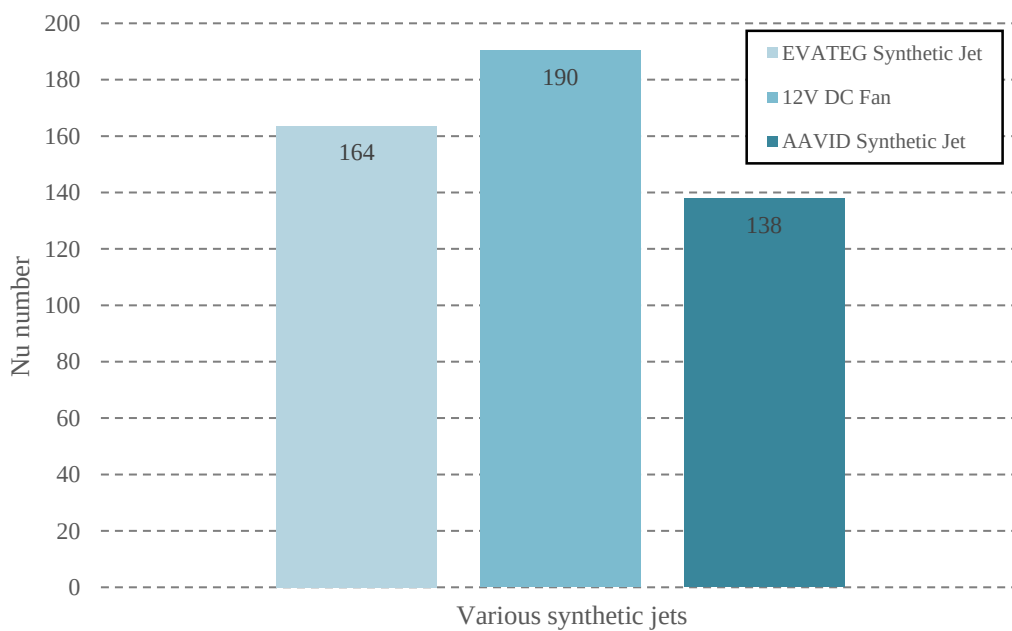


Figure 40: Nu number variation at $H = 10$ mm

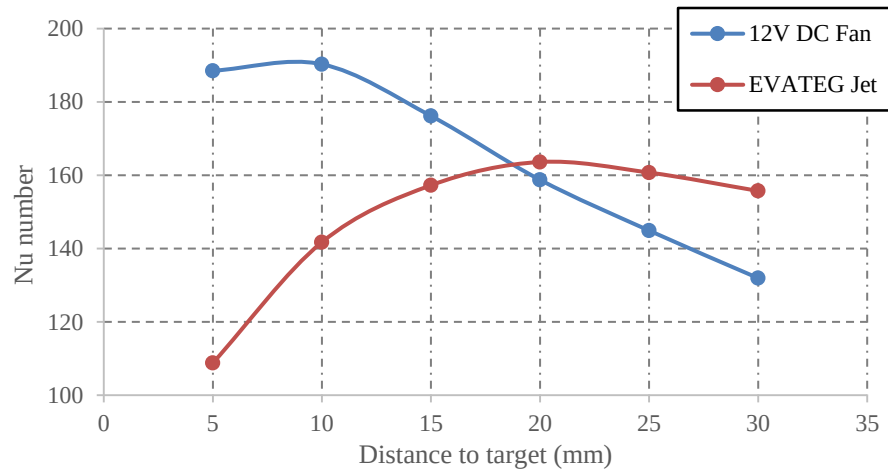


Figure 41: Variation of Nu number for axial fan and EVATEG jet

After heat transfer tests, LED chip cooling scenario is observed with infrared camera (IR) camera readings. Since temperature readings required direct visual contact, impingement cooling is changed to parallel cooling. Figure 42 depicts the test setup for IR camera temperature readings. Readings are obtained at the steady state conditions of LED chip operation, which is at temperature deviation of maximum 0.5°C , for a duration of 5 minutes. Duration of that, which is the stabilizing of the temperature distribution along the LED chip

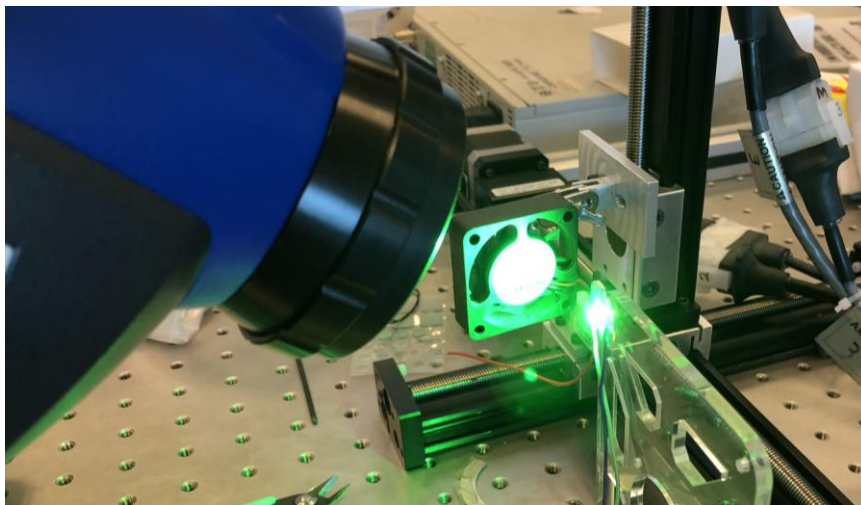


Figure 42: IR camera reading test setup

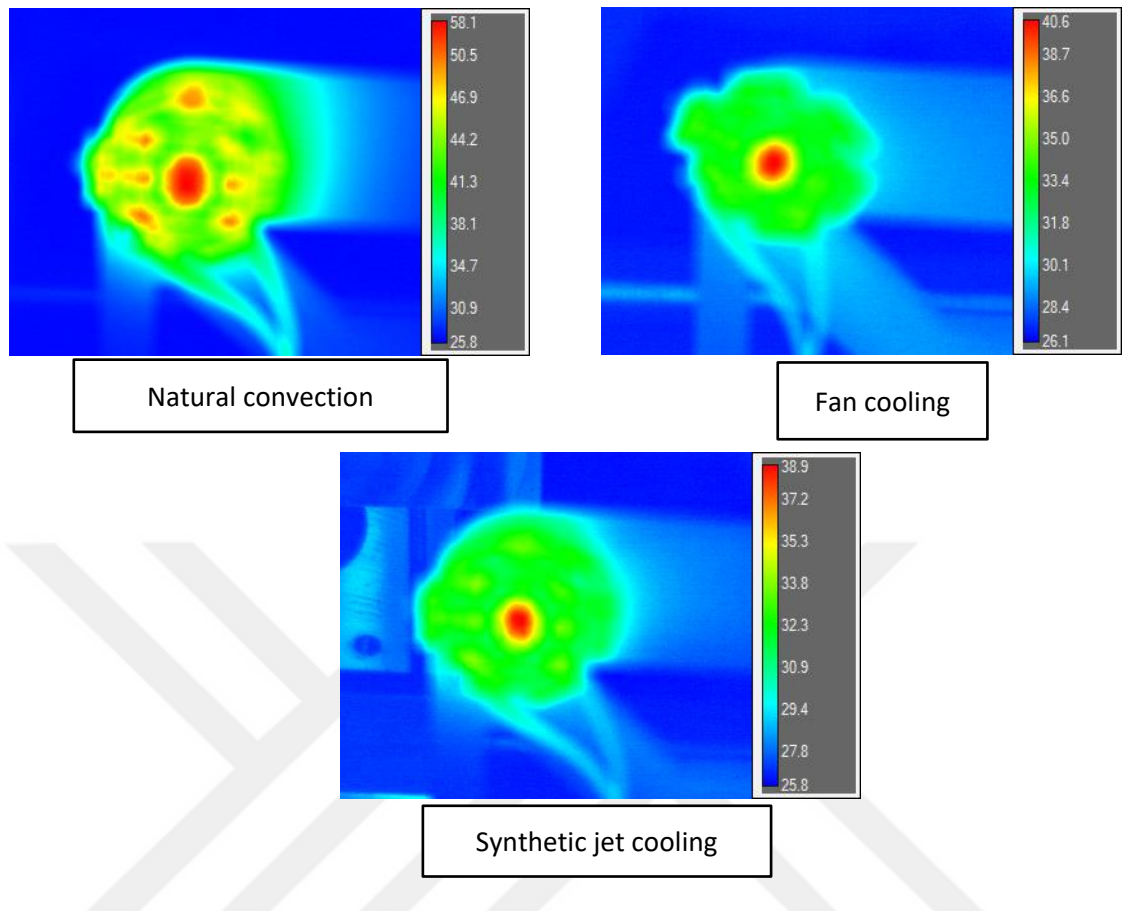


Figure 43: IR camera measurement results from LED cooling study

IR camera readings (Figure 43) have been performed at the break point at which the Nu values showed approximately equal values, which is $H = 15\text{mm}$. It is observed that at the steady state conditions, natural convection case has resulted maximum temperature of 58.1°C . Synthetic jet cooling case has resulted lower maximum temperature result of 38.9°C , compared to that of fan cooling, which is 40.6°C . IR reading resulted that at the given operating conditions, synthetic jet provided a slightly more increase in cooling performance.

Furthermore, natural convection measurements show that there are other hot spots on the PCB of the LED chip, having a maximum temperature of around 50.5°C . These hotspot points correspond to the junction of electronic components that are soldered

on the PCB. It can be observed that at these hot spots have a temperature difference of around 8°C compared to the temperature of LED's dome. This difference is negligible, comparing the temperature difference between PCB's lowest temperature surface and LED dome, which is approximately 20°C . The hot spots that are seen at locations other than the dome, are eliminated with the axial fan cooling, while a slight increase of temperature on these points are observed at synthetic jet cooling, with a maximum temperature of 33.8°C . It must be noted that synthetic jets perform better at impingement cooling, as the cyclical creation of air jets increase the heat removal rates substantially. That is why at impingement cooling scenario, synthetic jets are expected to perform better than parallel cooling. To specifically compare the LED cooling scenario, further analysis using novel measurement techniques, such as forward voltage measurement can be used to compare axial and synthetic jets in impingement cooling of LED chips.

CHAPTER V

MECHANICAL BEHAVIOUR OF SYNTHETIC JETS

This chapter provides the study of a mechanical design for synthetic jet actuators. In the light of the experimental studies, mechanical behavior of jet actuators has been observed to have a direct impact on the thermal performance. To this end, the mechanical design of synthetic jets, and the effect of mechanical parameters on resonant frequencies have been discussed. To provide a background on the packaging optimization, the effect of holding technique is investigated with parametric design variables. FEA results have been supplemented with experimental results for validation. Lastly, a novel packaging design is introduced, along with the numeric and experimental comparison between synthetic jets used in earlier studies.

5.1 Mechanical Modeling of Synthetic jet actuators

Mechanical behavior of synthetic jet actuators is directly in relation with the thermal performance. As mentioned and studied in some of the earlier studies ([12][13] and [16]), maximum mechanical deflection, which is the resonance point, results higher air flow and thus better thermal performance. On the other hand, resonating actuators may cause vibration related problems on the surface where they are mounted. To tackle this, mechanical connection with mounting part is made out of material with high damping coefficient, which is usually the wall material. Furthermore, the connection type directly affects the mechanical behavior, since the mechanical constraints on the system change drastically. This study focuses on the mechanical design of the holder for synthetic jet actuators, comparing their effect on the mechanical behavior. Along with the experimental study, mechanical modeling and nodal analysis is also performed.

5.1.1 Synthetic Jet Novel Packaging Approaches

Various mechanical models as jet actuator holder have been generated, by taking resonant frequency, mechanical constraints, and ease of manufacturing into consideration. First model was created for central orifice synthetic jet actuators, to compare their behavior with recently performed studies. Main criterion for the holder design was to eliminate physically, the connection between holder and the piezo disks. Figure 44 depicts the first design of synthetic jet actuator holder, which is inspired from the studies performed on piezo actuated synthetic jets ([29], [65]).

According to the design of holder, several mounting holes are included for the testing and mounting capabilities. Holder material is chosen to be stainless steel, with a thickness of 0.5 mm. Holder is manufactured by CO₂ laser cutting technique. To be compatible with both side and central orifice jet actuators, designed holder is consisted of 3 arms and an opening at one side. The effect of this opening on the mode shape and resonant frequencies are to be observed in this study.

Before the numerical analysis, manufacturing and testing is performed in order to observe the mechanical and thermal performance of the jets. In the light of the experimental results, modal analysis is performed to validate the mechanical behavior, along with the mode shape observation.

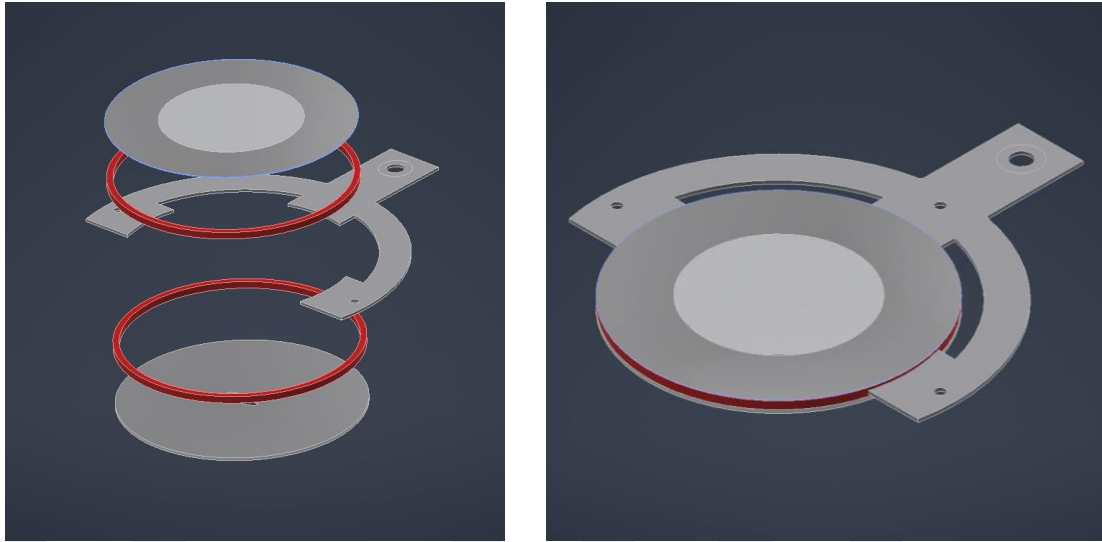


Figure 44: Exploded and assembled design of central orifice jet with holder

5.1.2 Laser Vibrometer Experimental Results

Manufactured synthetic jet actuators for the holder design have been studied under laser vibrometer measurements, for mechanical behavior comparison. The results are to be compared with recent study of central orifice jet actuators, to see the effect of holder on mechanical values. Manufactured synthetic jet actuators that are mounted on the test rig are shown in Figure 45.

It is observed that the holder design introduced to the model reduced the maximum peak displacements of synthetic jet actuators, approximately by an average of 60%. Moreover, especially for elliptic and rectangular jets, single peak value with uniform inclination and declination is observed. Since an introduction of metallic material to the design increases the global stiffness of the system, decrease of maximum displacement for every case is expected.

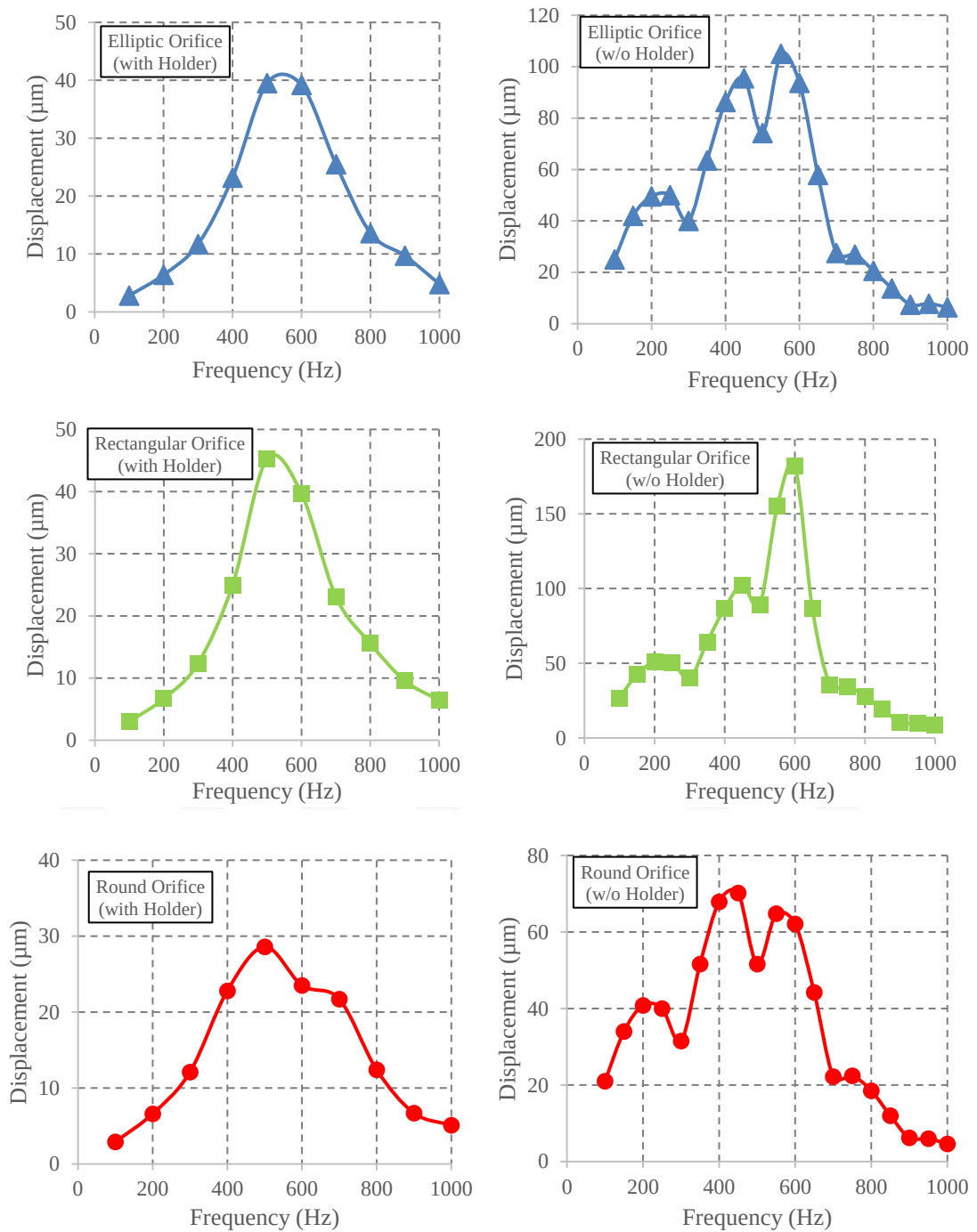


Figure 45: Laser vibrometer results for holder design study

Single peak behavior is similar with the Nu results as a function of frequency. As for the effect of holder to the cooling performance, it is observed that there is a slight difference for the round and elliptic orifice jet, but a drastic change of approximately

48% drop for the rectangular jet. Elliptic orifice jet has resulted 12% and round orifice jet has resulted 14% drop in the Nu values.

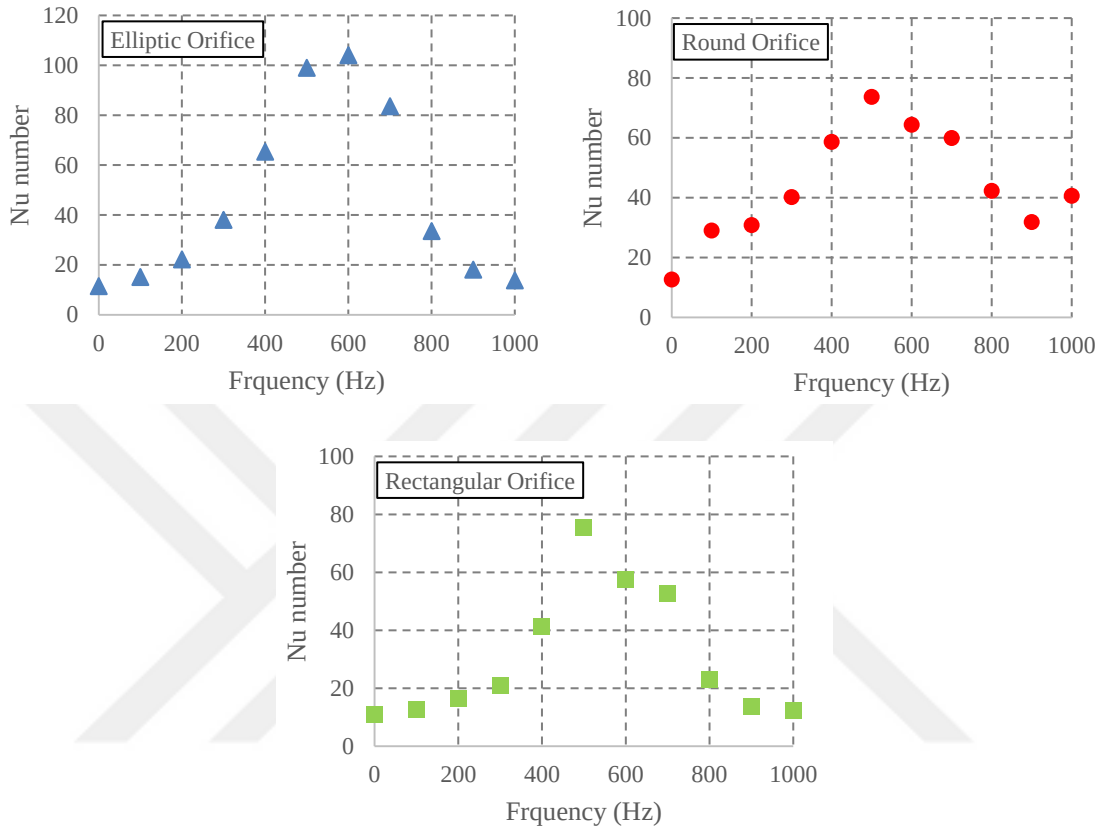


Figure 46: Nu number results for various packaged jets

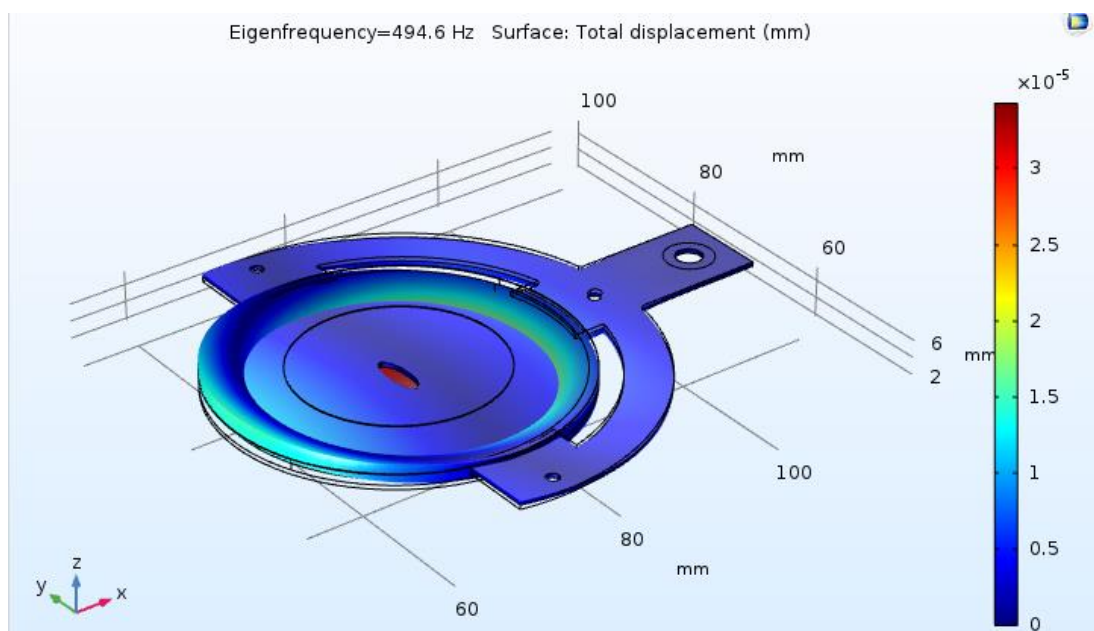
5.2 Finite Element Analysis

In order to validate the resonant frequencies, modal analysis has been performed on the synthetic jet model with holder. For this purpose, material properties has been determined according to the manufacturer datasheet for wall material, and mechanical properties of the metals and piezo material used. Below parameters are used for the modal analysis (Table 15).

Table 15: Material data used in the modal analysis

Parameter	Wall Material	Holder Material	Piezo Material	Disk Material
Material Name	RTV Silicone	Stainless Steel	PZT-5A	Stainless Steel
Modulus of Elasticity (GPa)	0.003	200	63	200
Poisson's Ratio	0.495	0.3	0.31	0.3
Density (kg/m ³)	2000	7850	7700	7850

After running the parameters for the mechanical model, below results have been achieved. First and second mode for the holder design shows similar results with the experimental results. For the experimental results, it has been observed that the synthetic jet actuators with holder design resulted peak displacement and Nu values at 450 to 550 Hz. Modal analysis resulted the first two resonant frequencies to be 495 and 535 Hz, respectively.



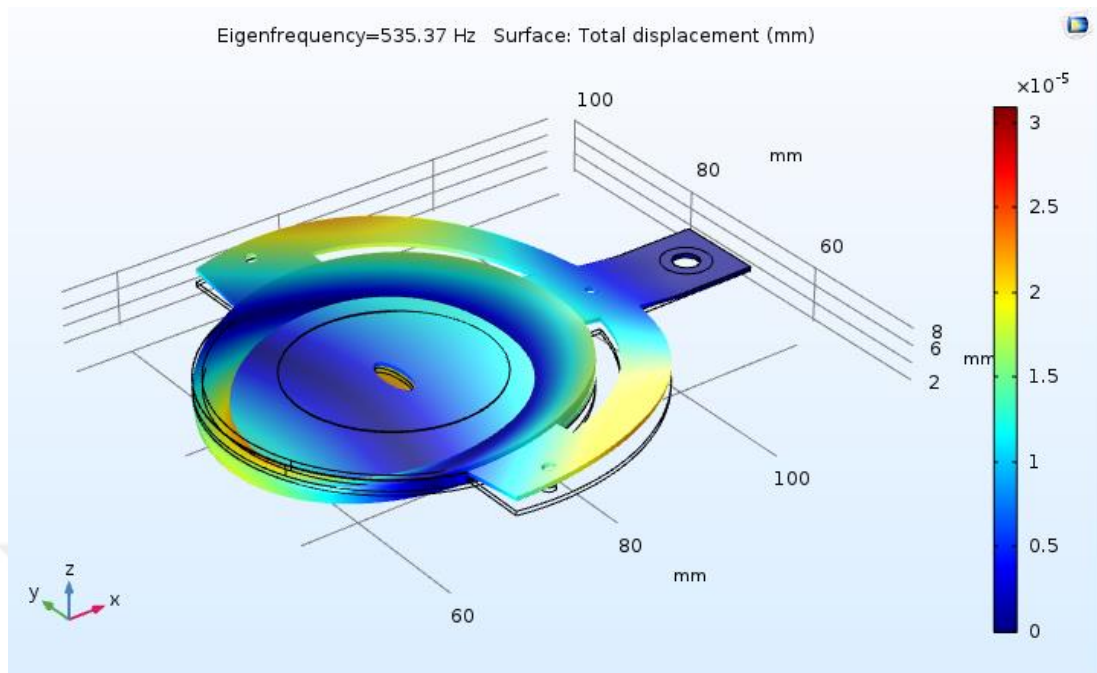


Figure 47: Holder design modal analysis results at 1st and 2nd mode

For each orifice type central jet with holder, namely elliptic, rectangular and round, relative error with respect to experimental results have been 4%, 2% and 2%, respectively.

5.3 Parametric Modal Analysis for Holder Design

In the light of the mechanical modeling and optimization, further studies have been performed in order to analyze their effect on the mechanical behavior. For this reason, a parametric CAD model of a synthetic jet holder is introduced. Several parameters have been determined to see their effect on the synthetic jet actuator mechanical behavior. Firstly, as seen from the FEA results in holder design, holder presence increases the rigidity and lowers the maximum displacement at the center. This is why the holder parameters are included, such as holder thickness, arm size, and holder material (Figure 48).

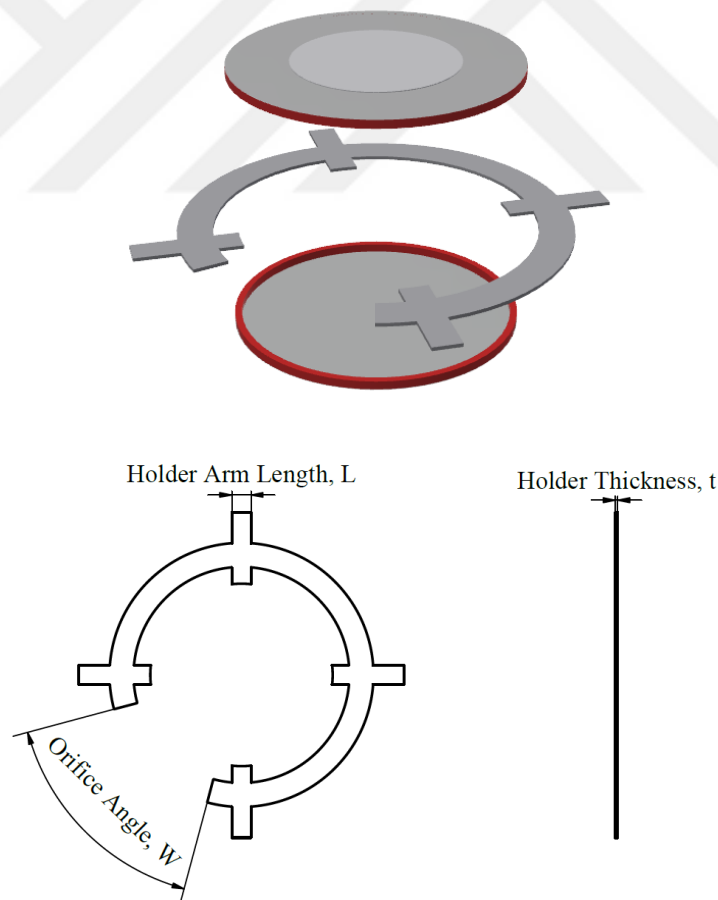


Figure 48: Parametric study dimensional boundary parameters

Moreover, effect of side orifice angle W is also included to observe their effect on the resonant frequency. Below table represents the parametric study conditions that are performed in FEA (Table 16). Note that the minimum side orifice angle is determined as 30° , since the recent study resulted lower heat transfer coefficient values. Instead, 45° orifice angle introduced in the study. Parametric study complete results are included in Appendix A.

Table 16: Parametric study and operating conditions

Parameter No	1	2	3
Holder Material	Stainless Steel	CK45 Steel	Copper
Holder Arm Length, L (mm)	2	3	4
Holder Thickness, t (mm)	0.1	0.3	0.5
Orifice Angle, W ($^\circ$)	30	45	60

According to the results, comparing the holder material, copper material resulted the lowest resonant frequencies, while stainless steel structure resulted the highest values. For the arm length, it is observed that the smaller arm sizes resulted lower resonant frequencies. Same behavior is observed for the holder thickness as well. As for the orifice angle, it is proved that the higher orifice angle resulted much lower resonant frequencies at the first mode, due to the absence of wall material and less constraint of 15%. Following figures depict the effect of the parameters studied on the resonant frequency (Figure 49).

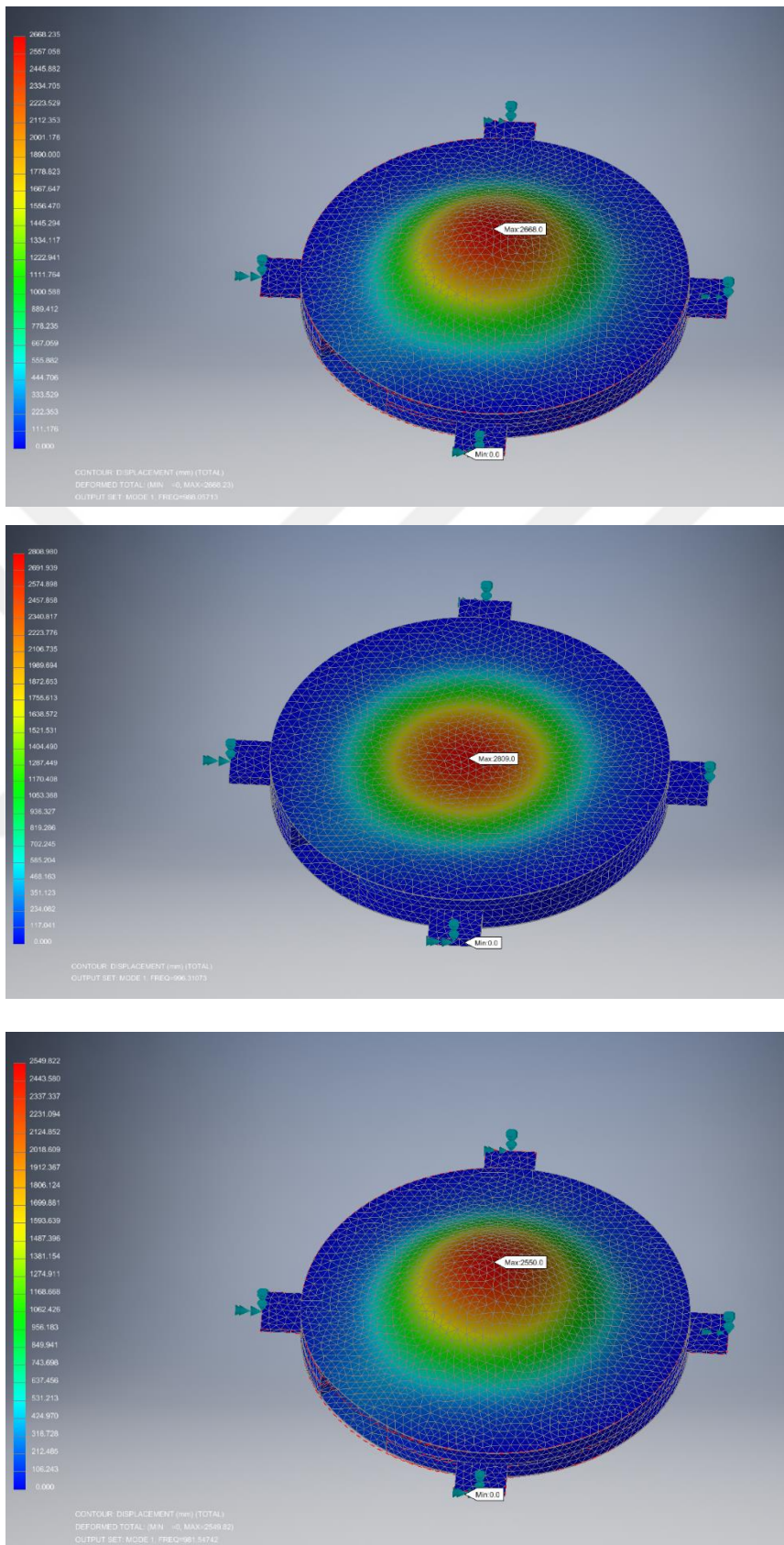


Figure 49: Parametric study mode shape at resonant frequency of 1st mode

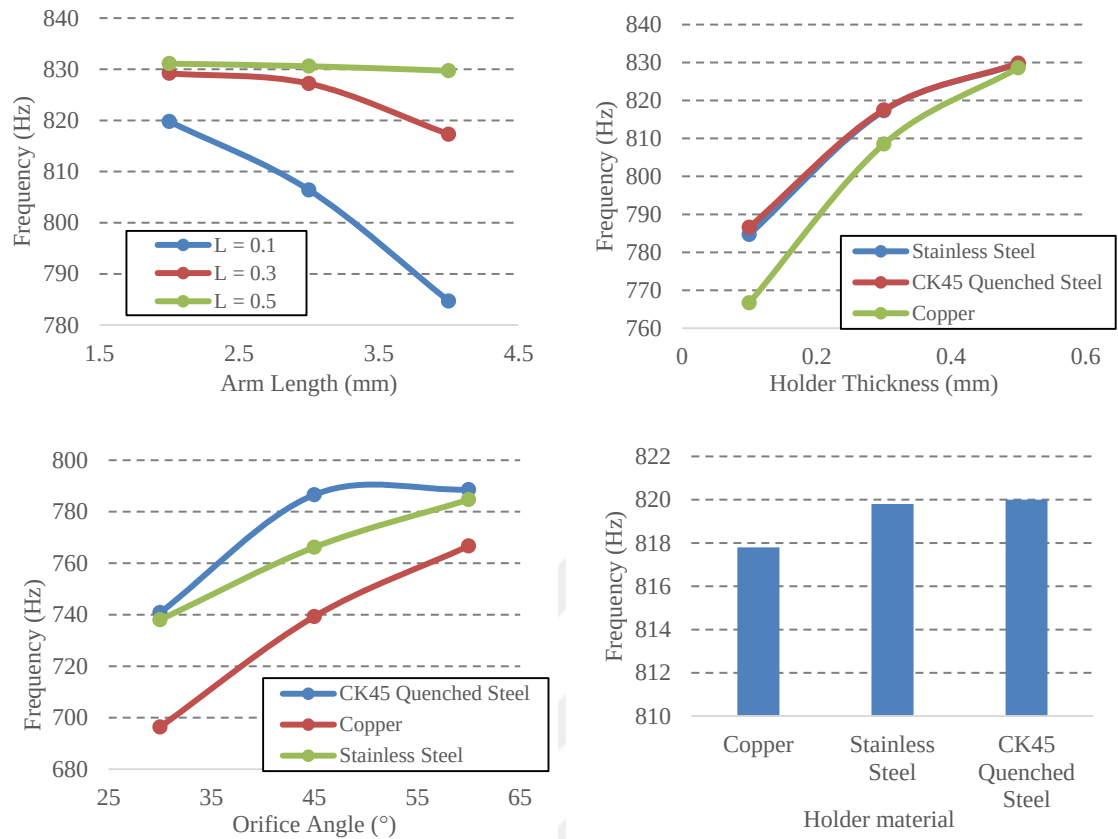


Figure 50: Parametric study results for synthetic jet mechanical design

According to the results on the parametric FEA study, it is found that the frequency interval for all of the combinations is 696 Hz to 831 Hz. Among the parameters, holder material has resulted the lowest effect on the resonant frequency, providing a shift average of 20 Hz, for given parameters. However, material effect on resonant frequency such as holder thickness and orifice angle provided relatively high changes of frequency, such as a shift of 42 Hz for orifice angle at $W = 30^{\circ}$, and 25 Hz for holder thickness at $t = 0.1$ mm. The lowest resonant frequency is observed at 696 Hz, with copper holder at orifice angle of 60° , thickness of 0.1 mm and an arm length of 4 mm. Synthetic jet actuator with quenched steel holder having an orifice angle of 30° , thickness of 0.5 mm and an arm length of 2 mm resulted the highest resonant frequency, which is 831 Hz. Apart from the frequency at which the synthetic jet

resonates, the way it displaces must also be taken into consideration. According to the mechanical constraints and holder parameters, resonance points of the mechanical structure of jets differ, resulting a vibration with a behavior other than expected. To expand on the point, mode shapes are observed, so that the most efficient vibration form is determined.

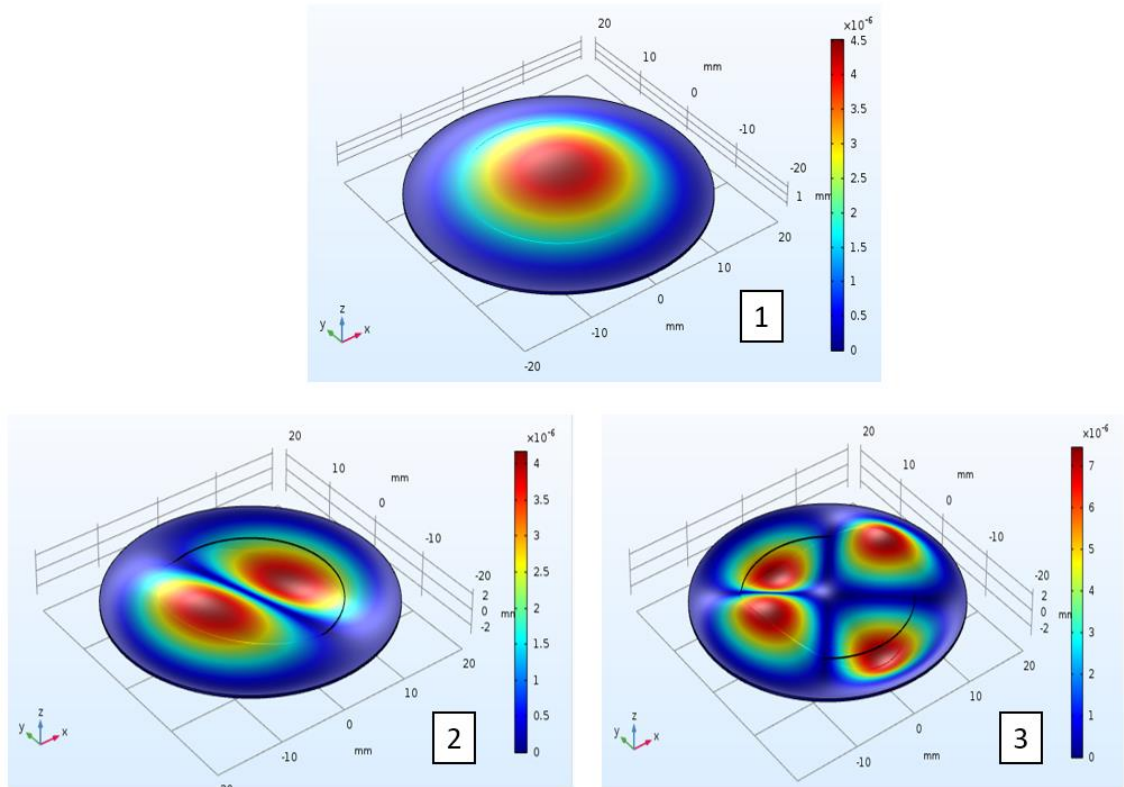


Figure 51: Mode shapes for the first 3 resonant frequencies of piezo disk

Above figures from the parametric study illustrate the behavior of the jets at the resonant frequencies (Figure 51). First mode shape of the figure represents a resonance frequency in which the piezo disk behaves as a diaphragm, resulting the desired air flow. On the other hand, second and third mode shapes have symmetrical displacement mode shapes. Displacement at the disk surface is cancelled out by a symmetric surface from the center of the piezo disk, meaning that the theoretical

volume change for the cavity of the synthetic jet is zero. Regardless of the frequency or displacement, mode shape results a poor air movement inside the cavity, and thus the cooling performance is also affected.

Given the results, it can be concluded that the holder material and thickness affect the resonant frequency, which is the highest disk displacement at the center of the disks, thus the designing parameters must be corresponding with these results. It is also observed that the mode shape must be corresponding to the behavior of a diaphragm, in order to achieve high displacement at the center and a significant amount of air flow. Mode shape at the resonant frequency must be observed in order to prevent unwanted resonant values, which causes air flow to drop. Thickness of the holder must be designed such that the mechanical constraints on the piezo disk allow only displacement at the center of the disk. Other mode shapes, even though they might occur at the lower frequencies, result symmetric mode shapes, that reduces or prevents air flow, hence the stabilization problems of the mechanical structure.

Due to the availability and cost, it is found that the stainless steel holder material is a better choice for the packaging of the synthetic jets, even though it presents a drawback in terms of rigidity and high resonant frequency.

In their study, Ikhlaiq et al. [13] investigated the mechanical performance in the computational model, by only modeling and constraining the perimeter of the disk. This method has a disadvantage in terms of application conditions, due to the difference of mechanical constraints for the packaged structure. To prevent it, this study will focus on the mechanical constraints applied at the real application conditions.

5.4 Novel Packaging Design

As a result of the thermal performance and parametric mechanical behavior studies, a novel design (Figure 52) with compatibility to both side orifice and central orifice jet actuators are introduced in this part of the study. Up to this point, thermal performance is found to be dependent on the resonant frequency f , length scale H/D_h , orifice shape, and driving voltage. Combining the best parameters achieved by the studies, a central orifice jet actuator with a D_h of 3.5 mm, driven at a 15 mm of distance to target is set as design parameters. Apart from the thermal performance, mounting ability and feasibility in applications is aimed, so that the package can be mounted on a desired circuit or surfaces easily, without over or under-applying constraints. It is observed in the recent studies that a mechanical system with high rigidity will result low deflection at the diaphragm, while low rigidity will result circumstantial uncertainties that will divert the cooling capacity.

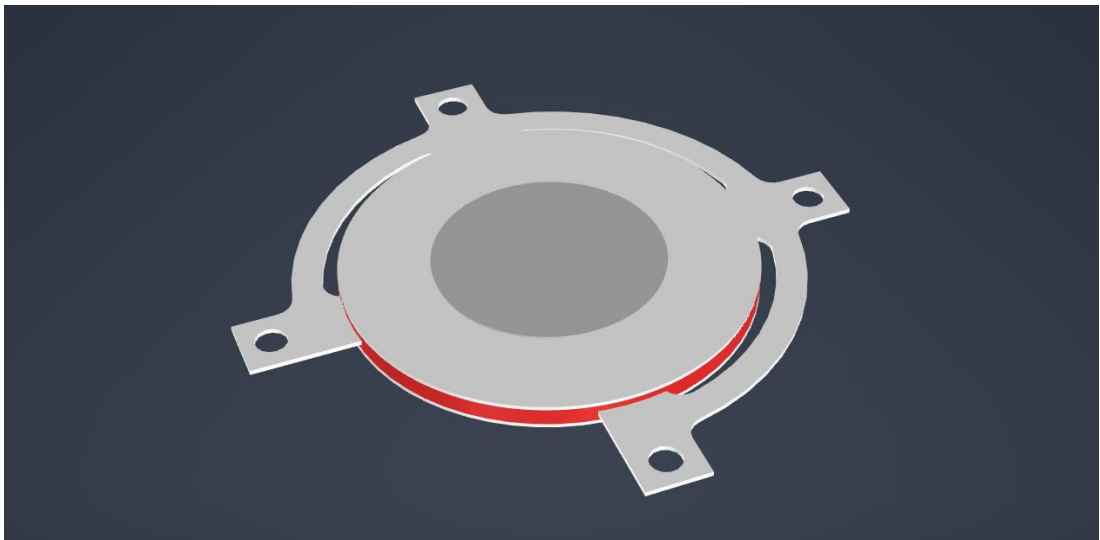


Figure 52: Novel design for the optimized synthetic jet actuator

Using the finite element analysis (FEA), mechanical optimization on the current holder design, laser vibrometer testing and thermal performance calculations are performed. To be able to stay in the low frequency range, mechanical design is performed regarding the results from the parametric study. With current state, resonant frequency of the novel packages of side orifice and central orifice jet is around 495Hz and 561 Hz, respectively.

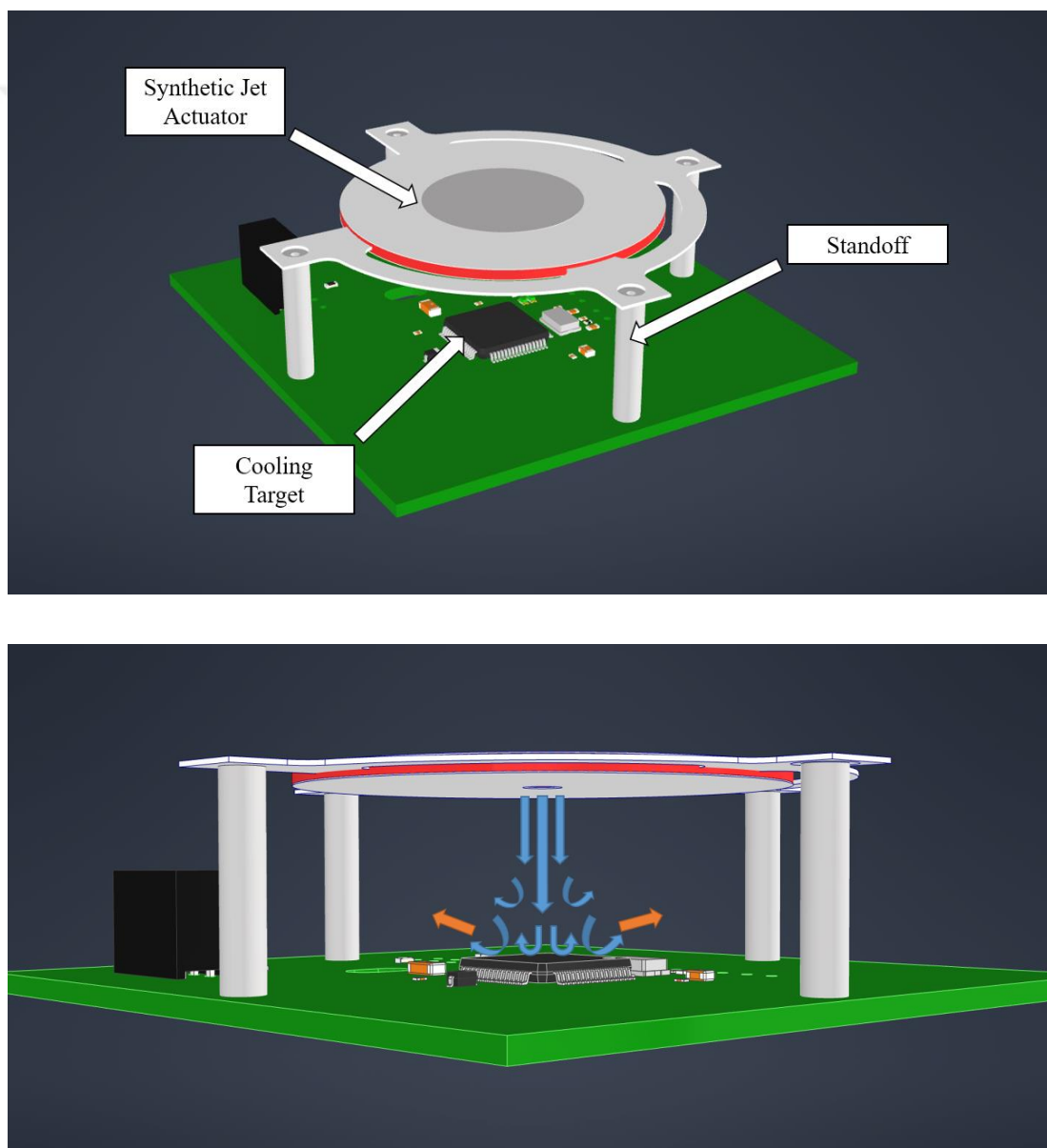


Figure 53: Representative schematic of synthetic jet application

After designing and manufacturing the jet actuator, experimentations are conducted to validate and observe the mechanical and thermal performance. As depicted in the design, there are 4 mounting holes for the holder, by which a M3 threaded bolt can be used with a 15 mm standoff, to the target surface (Figure 53).

In the light of the modal analysis results, it is observed that the mechanical constraints in experimentations may differ from the mechanical modeling. In the mechanical modeling and the predicted real life application, synthetic jet holder is mounted by the mounting holes. This creates a mechanically fixed constraint, for both mechanical FEA model and real life application. However, while testing, synthetic jets are mounted with only one fixation point. This creates an absence of constraint points, and results a difference in mode shapes and resonant frequencies. That is why the need of mechanical constraints while testing emerges. That said, to eliminate uncertainties and experimental errors, mechanical constraints are implemented into test procedure. To achieve that, testing holder design (Figure 54) is introduced to the study. Consisted of bolted connection interface, packaging holder is fixed from the connection surface, such that the fixation of holder in real life application is imitated. Testing holder is also manufactured with a relatively larger thickness and mounting interface, in order to provide rigidity and global stiffness, similar to the application conditions.

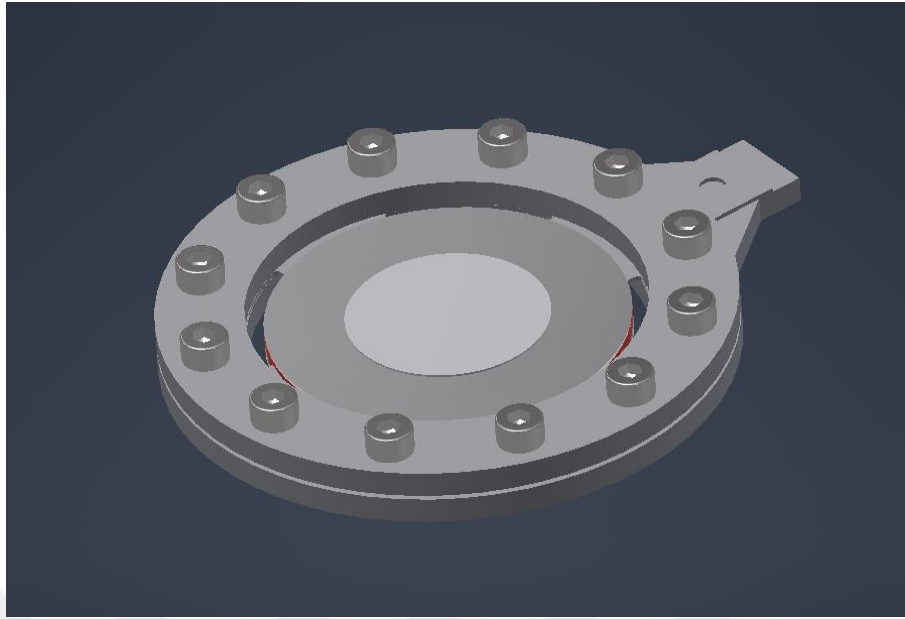


Figure 54: Mechanical design of test holder

Validation of the test holder is performed by comparing the modal analysis results and mode shapes between package mounting and test holder mounting. The objective is to achieve similar mode shape behavior and resonant frequencies.

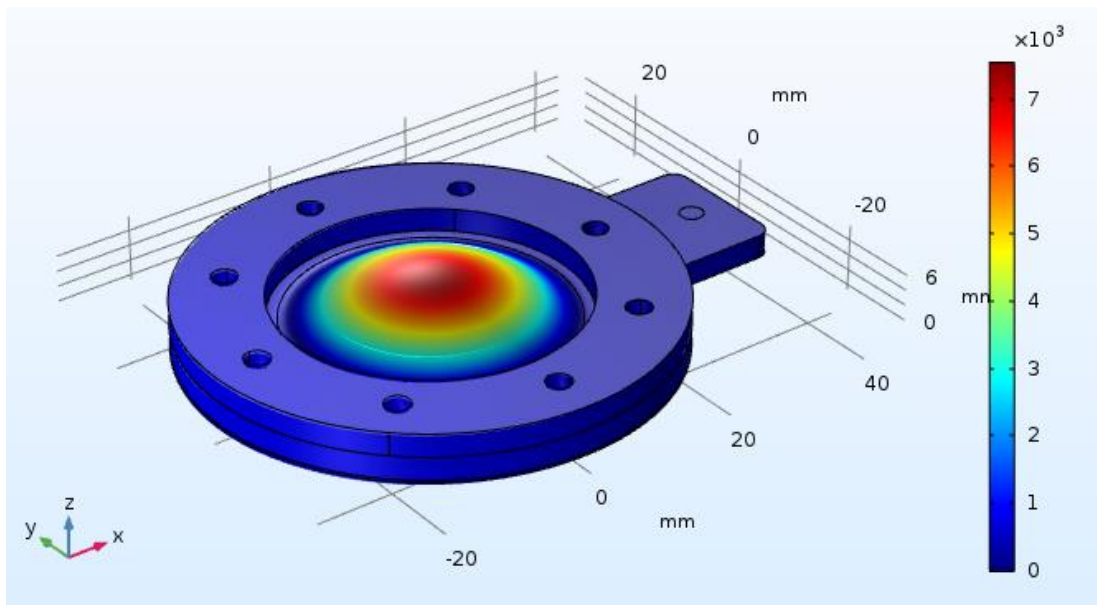


Figure 55: Mode shape of synthetic jet with mounted test holder

It is observed that the desired mode shape for the testing holder mechanism is occurring at a frequency of 653 Hz, while packaged version results a frequency of 608 Hz. Figure 55 depicts the mode shape of the test holder mechanism, at its 4th mode of resonance. Former mode shapes resulted displacement points on the elastomeric wall and holder itself, which is why they are not taken into account for operation. A relative error of 7% is found for the resonant frequency. Less stiff mechanical design approaches for the test holder resulted smaller relative errors, but drastically changing the mode shape, resulting symmetrical displacements and a theoretical volume change of zero.

5.4.1 Comparison of Numerical Results with Experimental Findings

Laser vibrometer readings have been performed on the newly designed synthetic jet actuator. Following figure represents the displacement at the center of the actuator, as a function of driving frequency f (Figure 56).

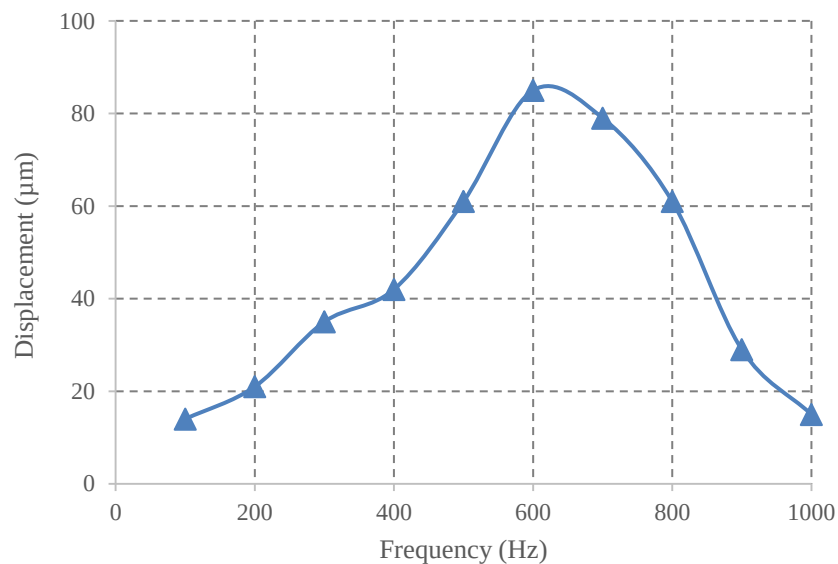


Figure 56: Laser vibrometer results for novel actuator design

As seen from the results, new actuator is found to have its resonant frequency around 600 Hz. The relative error of resonant frequency is 2% with the numerical solution (Figure 57). A slow increase in displacement at a frequency range of 100-600 Hz is observed, while a steeper decline is observed at the 800-1000Hz frequency range. A slight peak at the 800 Hz is also observed, considered as another mode of resonant frequency, which does not result a higher displacement at the center of the disk. Similar behavior has been observed in the earlier central orifice and side orifice studies, as well as the studies by Travnicek et al. [66]. Comparing experimental results and numerical solution, a relative error of 9% is observed, where a stiffer mechanical structure with test holder resulted higher resonant frequency.

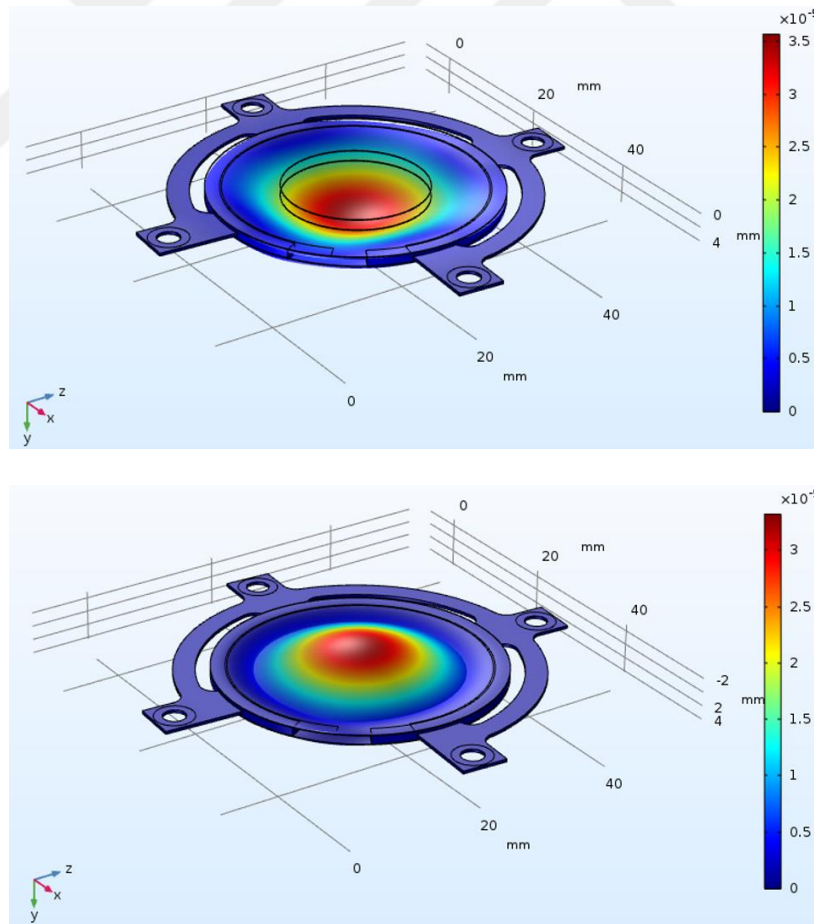


Figure 57: Modal analysis result, first and second mode (486 Hz, 608 Hz)

Focusing on the thermal performance, same two-peak behavior is also observed at the given frequencies. It can be seen that the packaging method has a drastic impact on the mechanical behavior, especially on the resonant frequency and mode shape. Many studies in the technical literature ([47], [49], and [53]), though it is not mentioned, use less stiff materials such as perplex plastic material or soft mounting interface used for speakers, in order to diminish mechanical constraints and lower the operating frequency. Novel synthetic jet actuator design has been put into thermal measurement by the heat transfer system, in which the Nusselt number is calculated for a frequency range of 100 to 1000 Hz. It is found that, as expected, synthetic jet actuator performed at its peak near the resonant frequency (Figure 58). Comparing the new design with recent studies, it has performed much better than the jet actuator manufactured in side orifice study, and slightly better than those in central orifice study. Below table depicts the comparison of mechanical and thermal behavior of the manufactured jets in the laboratory (Table 17).

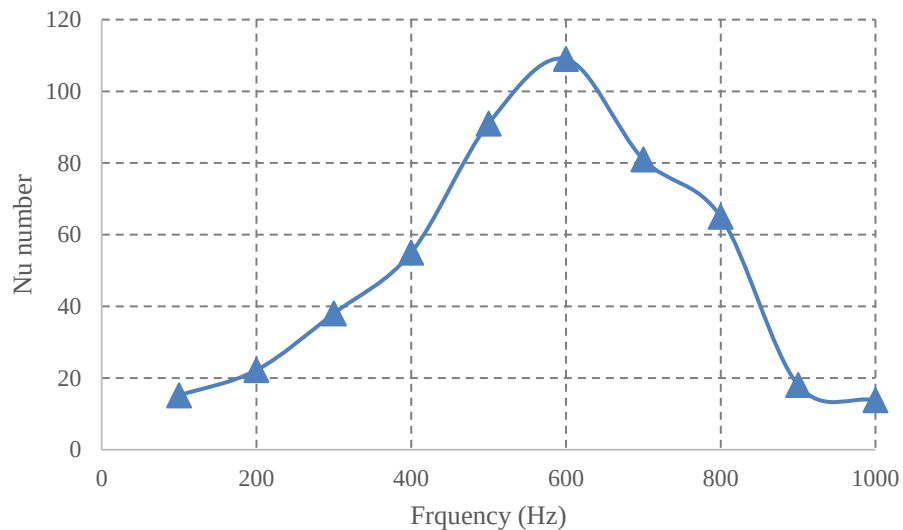


Figure 58: Heat transfer test results for novel jet actuator

A second relatively lower peak is observed at 800 Hz. At this point, it is also observed that a small increase in disk displacement occurs. Having a second mode peak at a range of 100-1000 Hz is also observed in other studies [60]. Using a high frequency micro-blower, Ikhlaiq et al. ([13] and [21]) observed the second peak in a frequency range of 100-5000 Hz, due to the smaller form factor of the micro-blower, while Ghaffari et al. [65] has observed three peak points at a frequency range of 100-2500 Hz.

Table 17: Manufactured synthetic jet actuator comparison, at H = 15 mm

Jet Type	Side Orifice	Central Orifice	Novel Design
Selection	60°	Elliptic	Central Elliptic
Resonant Frequency (Hz)	750	550	608
Maximum Displacement (μm)	90	105	85
Nusselt Number	128	117	109

CHAPTER VI

SUMMARY AND CONCLUSIONS

An experimental and theoretical study has been completed as a part of this research. Operating parameters of synthetic jet actuators are experimentally observed, and few correlations regarding their effect on the mechanical and thermal behaviors, have been performed in this study. To achieve repeatability and accuracy in actuator manufacturing, a pneumatic pick and place system is introduced to the study. Manufacturing accuracy and precision is achieved, so that the dimensional accuracy in between jet actuators is below 3%.

Side orifice synthetic jet actuators are studied to compare the effect of orifice size in a fixed distance to target $H = 10$ mm. Mechanical and thermal performances of jet actuators are observed.

Moreover, central orifice synthetic jet actuators with varying orifice shapes are introduced in the manufacturing side of the study. Effect of orifice geometry and length scale is observed, and comparison of mechanical and thermal performance is conducted.

Furthermore, in the light of the results from experiments, mechanical modeling and FEA study is performed to validate the mechanical characteristic from experiments and numerical solution. During the mechanical modeling, the effect of the holder on the mechanical behavior of the synthetic jet actuators is parametrically investigated, and experiments have been conducted to observe their effect on cooling performance. Finally, a novel design compatible with packaging needs is introduced and manufactured. Mechanical and thermal experiments have been performed to compare it with the former studies.

For the side orifice study, the following conclusions are drawn:

- For the given distance to target, 60° orifice jet actuators, having the length scale of $H/D_h = 4.41$ resulted the highest cooling performance.
- Displacement at the center as a function of time showed similar behavior as the displacement velocity at the center of the piezo disk.
- Length scale has an inverse effect on the cooling performance, for an interval of $4 < H/D_h < 5$. This correlation is also mentioned in a study of Ikhtlaq [21] and Greco, [23]. Wider length scale observation would be supplemental to this study.

For the central orifice study, the following conclusions are drawn:

- Orifice geometry has a significant effect on the heat removal rates from the cooling target. Especially rectangular orifice jet actuators performed the best, compared to the others, followed by the elliptic jet actuators.
- Both conventional and modified hydraulic diameters did not show any correlation with the Nu values, that is to say there should be a further investigation on the flow characteristics of the synthetic jet actuators with orifice geometry variation.,

For the comparison study of in-house manufactured synthetic jet actuators with commercial synthetic jet actuator and an axial fan, the following conclusions are drawn:

- Commercial synthetic jet actuator has less rigid body with a driving frequency of 194.5 Hz. A body with less rigidity provides better displacement at lower frequency, resulting better acoustic comfort.
- Double actuated AAVID synthetic jet resulted Nu numbers that are two times

higher than the Nu numbers of side orifice jets. This can be concluded as the double actuation of the synthetic jet actuators provide twice the air flow and thus twice the cooling performance.

- Axial fan with 37.5 mm cross section diameter provided higher cooling capacity at lower distance to target scenarios, however at $H > 15$ mm, synthetic jet actuators provided more heat removal rates. It can be concluded that at higher length scales, synthetic jets result better performance.

For the studies of mechanical modeling and a novel design of a synthetic jet actuator, the following conclusions are drawn:

- Holding technique directly affects the mechanical behavior of synthetic jet actuators. That is why, in an application based solution, holder mechanism must be taken into consideration while designing.
- Mounting arm length, holder thickness, and holder material are some of the parameters that significantly affect the resonant frequency of the synthetic jet actuator. For better performance at lower frequency range, arm length and holder thickness must be designed as low as possible. That said, mechanical structure of the package becomes prone to unsteady operating conditions, which is why a balance between cooling performance and driving frequency must be aimed.
- Apart from the resonant frequency values, formation of the vibration at the resonant frequency must also be taken into consideration. Mode shape results from the FEA studies suggested that the best cooling performance occurred at the modes in which the most displacement at the center of the jet disk appears.

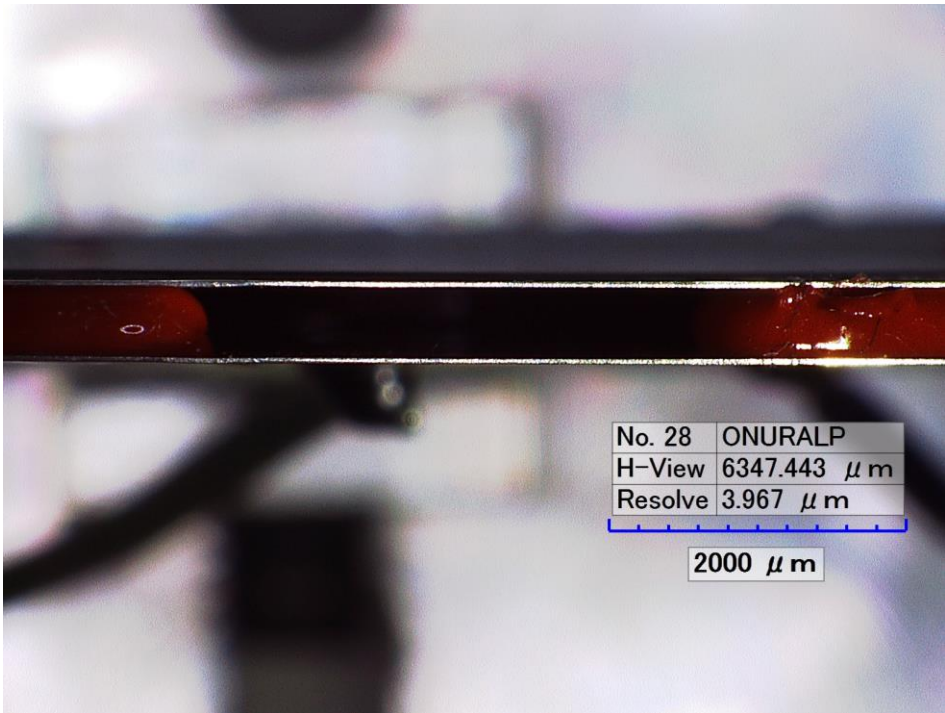
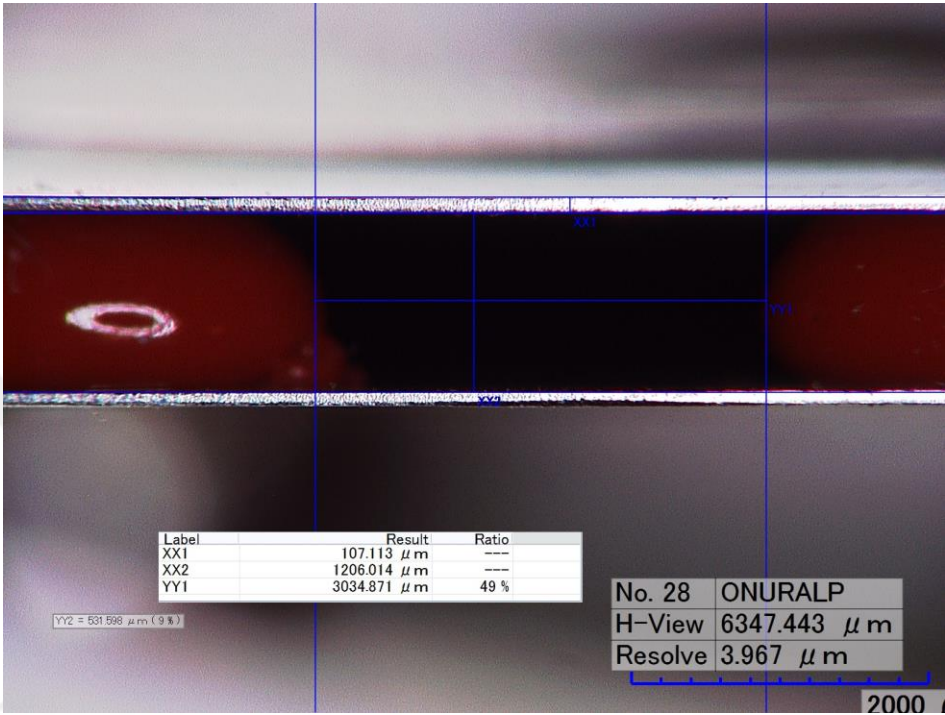
RECOMMENDATIONS FOR FUTURE WORK

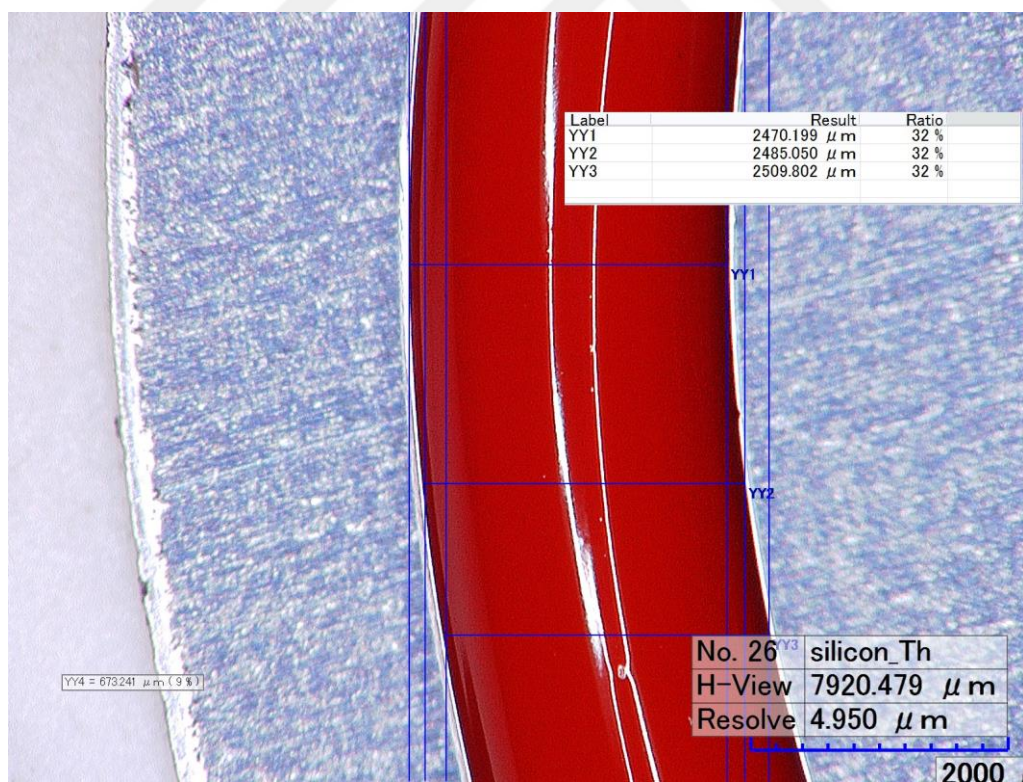
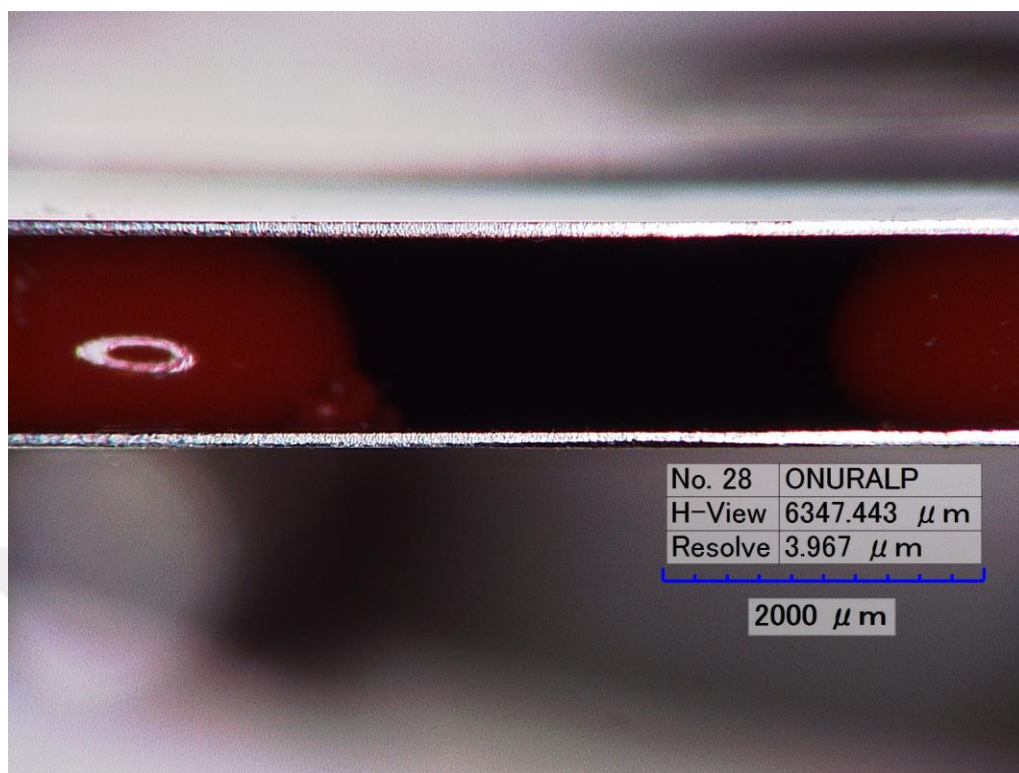
This thesis focused on the main parameters affecting the thermal and mechanical performance on synthetic jet actuators. As a conclusion, a novel product design has been proposed and validated using the approaches taken from recent studies and literature sources. From this point, further analysis on the different aspects should be investigated such as jet flow analysis. This can be performed with Schlieren photography, PIV experiments, and of course computational fluid mechanics (CFD) analysis, focusing on the characteristics of the cyclical jet flow. Furthermore, air velocity measurements can be performed in order to validate and derive the heat transfer calculations, along with the correlations that can be pointed out.

From the manufacturing standpoint, pick and place mechanism can be upgraded into fully automated system that can include a software that can be parametrized with boundary conditions, and produce the jet actuators automatically, with the help of servo axis linear motion system. This implementation can substantially increase the manufacturing accuracy. To expand even more on the manufacturing capability, piezo disks can also be assembled in-house, with the similar application that applies the piezo material into any surface. This can open up new areas in synthetic jet manufacturing capability such as geometry variations in diaphragm. As seen from the commercial jet study, lower driving frequencies with better performance could be achieved because of the geometry of the diaphragm, having a resonant frequency of much lower than 500 Hz. Lowering the driving frequency while preserving the cooling capacity is crucial, because of the acoustic comfort of the synthetic jet, especially for the end-user applications.

APPENDIX A

**DIMENSIONAL INSPECTION OF MANUFACTURED
SYNTHETIC JET ACTUATORS**





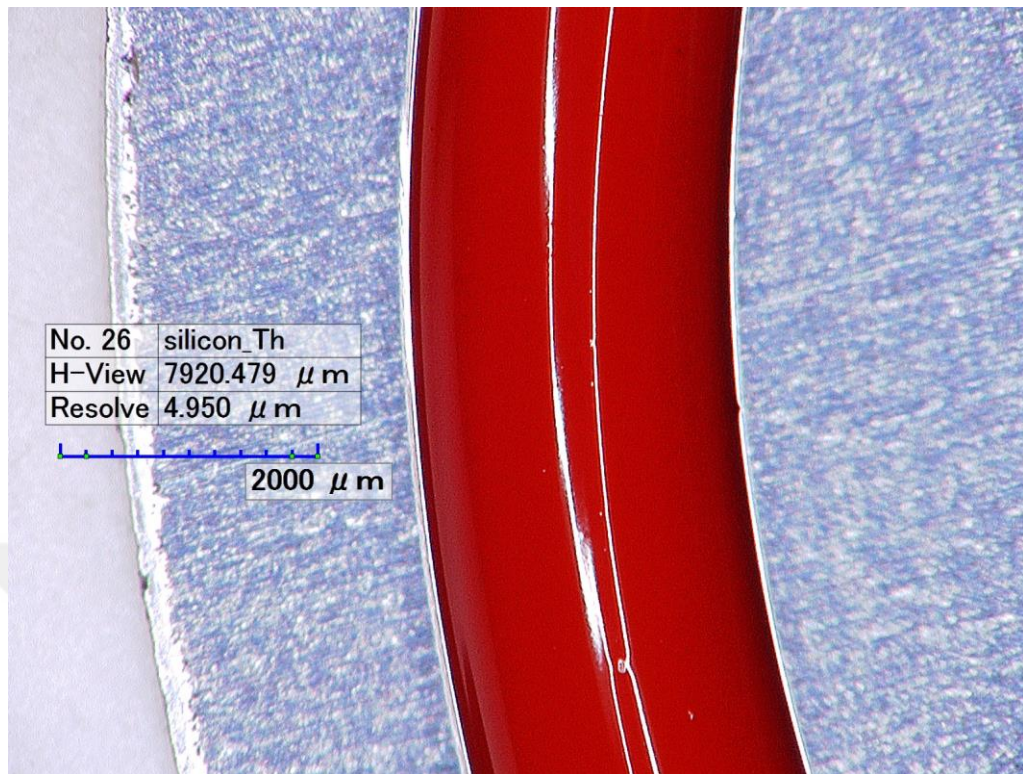


Figure A.1, 2, 3, 4: Microscopic measurements of side orifice and applied wall thickness

APPENDIX B

PARAMETRIC STUDY FOR HOLDER DESIGN

Table B.1: Parametric study results table

Width Study	Arm Length Study	Holder Thickness	Holder Material Study	Resonant Frequencies	
W (°)	L (mm)	t (mm)	-	1st	2nd
30	4	0.1	Copper	696.4	846.5
30	4	0.1	Stainless Steel	738	847.9
30	3	0.1	CK45 Quenched Steel	739.3	852.1
45	4	0.1	Copper	739.3	847.9
30	4	0.1	CK45 Quenched Steel	740.8	848
45	4	0.1	Stainless Steel	766.2	849.5
60	4	0.1	Copper	766.7	849.4
60	4	0.1	Stainless Steel	784.7	851.3
45	3	0.1	Copper	785.8	851.6
60	4	0.1	CK45 Quenched Steel	786.6	851.6
30	3	0.1	Stainless Steel	787.9	851.8
45	4	0.1	CK45 Quenched Steel	788.6	849.7
45	3	0.1	Stainless Steel	798.4	853.8
60	3	0.1	Copper	799.3	853.8
45	3	0.1	CK45 Quenched Steel	799.7	854
30	4	0.3	Copper	806.1	855.2
60	3	0.1	Stainless Steel	806.4	855.8
60	3	0.1	CK45 Quenched Steel	807.4	856.1
60	4	0.3	Copper	808.5	857.3
45	4	0.3	Copper	815.5	857.3
30	4	0.3	Stainless Steel	816.4	857.8
45	2	0.1	Copper	816.43	860.3
30	4	0.3	CK45 Quenched Steel	817	858
60	4	0.3	Stainless Steel	817.3	860.5
60	4	0.3	CK45 Quenched Steel	817.5	860.7
60	2	0.1	Copper	817.8	861.1
45	2	0.1	Stainless Steel	819.1	862
45	2	0.1	CK45 Quenched Steel	819.3	862.2
60	2	0.1	Stainless Steel	819.8	862.6
60	2	0.1	CK45 Quenched Steel	820	862.7
30	3	0.3	Copper	820.6	859.4
30	2	0.1	Copper	820.8	863.5
30	2	0.1	Stainless Steel	821	864
30	2	0.1	CK45 Quenched Steel	821	864
45	4	0.3	Stainless Steel	821.6	859.7
45	4	0.3	CK45 Quenched Steel	822	859.9
45	3	0.3	Copper	823.2	860.5

30	3	0.3	Stainless Steel	824.3	861.3
30	3	0.3	CK45 Quenched Steel	824.6	861.5
30	4	0.5	Copper	825.2	861.2
60	3	0.3	Copper	825.3	861.8
45	3	0.3	Stainless Steel	826	862.26
45	3	0.3	CK45 Quenched Steel	826.2	862.4
60	3	0.3	Stainless Steel	827.2	863.2
60	3	0.3	CK45 Quenched Steel	827.4	863.3
45	4	0.5	Copper	827.5	862.6
30	4	0.5	Stainless Steel	827.8	862.9
30	4	0.5	CK45 Quenched Steel	828	863
30	3	0.5	Copper	828.1	863.2
45	2	0.3	Copper	828.1	864
60	2	0.3	Copper	828.5	864.3
60	4	0.5	Copper	828.6	863.4
30	2	0.3	Copper	828.7	864.6
45	2	0.3	Stainless Steel	828.9	864.7
45	2	0.3	CK45 Quenched Steel	829	864.7
45	4	0.5	Stainless Steel	829.1	863.9
60	2	0.3	Stainless Steel	829.2	864.9
30	2	0.3	CK45 Quenched Steel	829.2	865
60	2	0.3	CK45 Quenched Steel	829.2	865
30	2	0.3	Stainless Steel	829.2	865
45	4	0.5	CK45 Quenched Steel	829.3	864
30	3	0.5	Stainless Steel	829.4	864.1
30	3	0.5	CK45 Quenched Steel	829.5	864.2
45	3	0.5	Copper	829.5	864.2
60	4	0.5	Stainless Steel	829.7	864.4
60	4	0.5	CK45 Quenched Steel	829.8	864.5
60	3	0.5	Copper	830	864.6
45	3	0.5	Stainless Steel	830.3	864.9
45	3	0.5	CK45 Quenched Steel	830.3	865
60	3	0.5	CK45 Quenched Steel	830.6	865.2
60	3	0.5	Stainless Steel	830.6	865.2
30	2	0.5	Copper	830.6	865.3
30	3	0.1	Copper	830.7	865.3
45	2	0.5	Copper	830.7	865.3
60	2	0.5	Copper	830.8	865.4
30	2	0.5	Stainless Steel	830.8	865.5
30	2	0.5	CK45 Quenched Steel	830.9	865.5
45	2	0.5	CK45 Quenched Steel	831	865.6
45	2	0.5	Stainless Steel	831	865.6
60	2	0.5	CK45 Quenched Steel	831.1	865.7
60	2	0.5	Stainless Steel	831.1	865.7

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VITA

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