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AN EXPERIMENTAL STUDY ON COOLING BY AN IMPINGING AIR JET

MASTER THESIS

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LIST OF SYMBOLS

A	:	Surface area [m^2]
C_p	:	Specific heat at constant pressure [J kg^{-1} K^{-1}]
D	:	Jet diameter [m]
h	:	Jet-to-plate distance [m]
h	:	Heat transfer coefficient [W m^{-2} K^{-1}]
H	:	Dimensionless jet-to-plate spacings, h/D
k	:	Thermal conductivity [W $m^{-1}K^{-1}$]
Nu	:	Local Nusselt number , hD/k
$\overline{Nu}$	:	Average Nuselt number
P	:	Electrical power to heaters [W]
Pt	:	Total pressure [Pa]
Ps	:	Static pressure [Pa]
Pr	:	Prandtl number, $\mu C_p/k$
q	:	Surface heat flux [W m^2]
r	:	Distance from stagnation point [m]
R	:	Dimensionless radial distance from stagnation point, r/R
Re	:	Reynolds number based on the diameter of the jet, VD/ν
T_j	:	Air jet temperature [K]
T_w	:	Surface temperature [K]
T_{ref}	:	Referance temperature [K]

ΔT	:	Difference between local temperature of surface and air jet temperature, $T_w - T_j$ [K]
V	:	Jet velocity [m sec^{-1}]
V	:	Voltage [V]
z	:	Distance normal to the target surface [m]
α	:	Seebeck coefficient [VK^{-1}]
ρ	:	Air density [kg m^{-3}]
ν	:	Kinematic viscosity of air [m^2 sec^{-1}]
μ	:	Dynamic viscosity of air [kg $m^{-1}sec^{-1}$]

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SUMMARY

This study consists of the experimental investigation of the hot surface cooling phenomenon by an impingement air jet at Reynolds numbers 5842, 8763, and 11684 and nozzle-plate spacings,H =2, 4, and 6.

The compressed air coming from the compressor, goes to nozzle after it was set to the desired total pressure level by a regulator. This air jet impinges to the copper plates which has heated by four cylindirical heaters. There were sixteen channels ot measure temperature. One of them was benefited as a reference channel and two thermocouples were used to monitor air jet and the connection of the amplifier-thermocouples temperatures. Thirteen thermocople wires were set up closely to the surface to measure surface temperature. The temperature changes are monitored by a data acquisition card and a computer.

As a result, local Nusselt number and average Nusselt number are presented in therms of Reynolds number and nozzle surface spacings.

ÖZET

ÇARPAN HAVA JETi YARDIMIYLA DÜZ BiR YÜZEYiN SOGUTULMASININ DENEYSEL OLARAK iNCELEMESi

Mühendislikte yüksek sıcaklıklara ulaşan sistemlerin soğutulması probleminin verimli ve amaca uygun şekilde çözülebilmesi temel araştırma konularindan biridir. Bu problemin çözümlenebilmesi için sistemin yapısına uygun olan çok çeşitli soğutma yöntemleri mevcuttur. Örneğin otomobil motorlarında yüksek sıcaklıklara ulaşan motorların vantilatör yardımı ile sıcaklığının makul değerlerde tutulması, klimalar yardımı ile konutların soğutulması ve benzerleri.

Bu çözümlerden biride jet yardımı ile soğutma işlemidir. Bu yöntemin temeli, bir veya daha fazla sayıdaki lülelerden çıkan jetler yardımı ile soğutulması istenen yüzeye çarptırılarak sicakliğının düşürülmesi temeline dayanmaktadır.

Çarpan jet yardımı ile soğutma işlemi bir çok mühendislik alanında uygulanmaktadır. Örneğin jet motorlu modern uçaklarda yüksek sıcaklıklara sahip yanma gazlarının türbin pallerinin aşırı ısınmasını önlemek için soğutulması, cam levhaların temperlenmesi, tekstil ve kağıt ürünlerinin kurutulması, elektonik devrelerin soğutulması sayılabilir.

Çarpan jetin yapısı, serbest jet, durma veya çarpma ve duvar jet bölgeleri olmak üzere üç ana kısımdan oluşur [1].

Serbest jet, temelde iki tip lüleden açığa çıkar. Ortam ile jetin karışımı büyük derecede olan tüp tipi lüleler ve karışımın sınırlı olduğu çarpma yüzeyine paralel olan basit bir yarık şeklindeki lülelerdir [2]. Lüleden uzaklıktıkça jet ile çevre arasındaki momentum transferi nedeniyle, jetin serbest sınırı genişler. Maksimum hız profili lüle çıkısından itibaren artan mesafe ile azalır. Çarpma yüzeyının konumu bu jet alanını etkiler.

ikinci kısım olan durma veya çarpma bölgesinde, akim alanı çarpmadan etkilenir akımın hızı normal doğrultuda lüle ile hızlı bir şekilde azalırken yatay doğrultuda ivme kazanır. Nusselt sayısı, Lüle ile yüzey arasindaki ve yüzey üzerindeki mesafeye kuvvetle bağlıdır. Düşük Reynolds sayilarındakı veya yüksek lüle ile yüzey arasindaki mesafe değerlerinde,

maksimum Nusselt sayısı daima durma noktasýnda meydana gelir ve lüle ile yüzey arasındaki mesafe, lüle çapından büyük değerler için Nusselt sayısı, Reynolds sayısı ile beraber artar fakat yüzey üzerinde jetin çarptığı noktadan itibaren azalır [3].

Son kısım olan duvar jeti bölgesinde, çevrenin sıfır momentuma sahip olması nedeniyle yatay ivmelenme sürekli olamaz ve durma bölgesinde ivmelenen akış yavaşlayan duvar jetine dönüşür. Böylece jetin yüzeye çarptığı noktadan itbaren mesafenin artması ile yüzeye paralel hız bileşeni sıfırdan itibaren bir maksimuma ulaştıktan sonra tekrar sıfır değerine düşer [1]. Konveksiyonla ısı transferi lüle ile hem durma bölgesinde hem de duvar jeti bölgesinde meydana gelir. Yüksek Reynolds sayılarında çarpma noktasından uzakta jetin yüzeye tekrar yapışabilme eğilimi nedeniyle ısı transferi levhanın ucuna doğru lokal bir artış gösterebilir [4]. Genelde durma noktası civarında akış türbülanslı yapıdadır. Bu durum lokal ısı transferi katsayısının artmasına neden olur. Lüle ile yüzey arasındaki mesafenin küçük değerleri için yüzeye paralel akışın ivmelendiği belirli bir bölge vardır.

Soğutulması istenen yüzey geniş olduğu zaman birden fazla lüle amaca uygun olarak kullanılır. Tek bir lüle ile birden fazla lüle arasında temel farklılıklar bulunmaktadır [1]. Bunlardan birincisi jetin yüzey üzerine çarpmasından önce jetler arasındaki etkileşimdir. Jetlerin birbirlerine yakın olmaları ve jet lüleleri ile çarpma levhası ile yüzey arasındaki mesafe arasındaki mesafe oldukça geniş ise böyle bir etkileşimin olma olasılığını arttırmaktadır. İkinci etkileşim ise duvar jetleri arasındaki birleşimdir. Jetler arasındaki mesafe oldukça az ve jet lüle ile hızı büyük olduğu zaman bu birleşimler artan öneme sahip olurlar.

Isı transferi çalışmalarında son zamanlarda sıvı kristal ve kızılötesi imaj kameralarinin (Infrared Imaging Camera) yüzey üzerindeki sıcaklik dağılımlarının bulunmasında [5,6,7] kullanılmasına rağmen ısı transferi çalışmalarında genelde dört tip transdüser kullanılmaktadır, bunlar;

- Isılçiftler (Thermocouples),
- Sıcaklık Direnç Dedektörü (RTD),
- Thermistör,
- Entegre Devre Sensörü (I.C-Integrated Circuit Sensor) dür.

Bunlar arasında ısıçiftler sıkça kullanılmaktadır. Tercih sebebleri basit yapıda olmaları, dayanıklıkları, ekonomik oluşları, çeşitlerinin bol olması ve geniş sıcaklık aralıklarında kullanilabilmeleri şeklinde sıralanabilir.

Isıl çiftlerde farklı metallerden yapılan iki tel bulunmaktadır ki bu teller, her iki ucundan birleştirildiği zaman sürekli bir elektrik akımi elde edilir. İlk olarak bu olay 1821 yılında Thomas Seebeck tarafindan gözlemlenmiştir [8]. Eğer bu devre ortadan kesilirse, net açık devre voltajı (Seebeck voltajı) birleşme noktası sıcakiliğının ve iki metalin yapısının bir fonksıyonu olur. Bu voltaj birbirinden farklı tüm metaller birleştiğinde görülmektedir.

Seebeck voltajı doğrudan ölçülememektedir, çünkü ilk olarak ısılçiftler bir voltmetreye bağlandığı zaman voltmetre uçları yeni bir elektrik devre oluşturmaktadır. Yeni oluşan bu gerilimi hesaplayabilmek için bu bağlantı noktası bir buz banyosuna daldırılır. Dolayısı ile bu noktanın sıcaklığı 0°C değerinde olur ve referans bağlantı noktası olarak isimlendirilir. Bu bağlantı noktası termal elektromotor kuvveti oluşturmaz.

Eğer amplikatör ile ısılçift tellerinin bağlantı noktaları aynı sıcaklık değerlerinde değillerse, ölçümlerde bir hata meydana gelecektir. Hassas bir ölçüm için bu bağlantı noktalarındaki sıcaklıkların da bilinmesi ve hesaplara katılması gerekmektedir. Dolayısı ile bu bağlanti noktaları bir izotermal blok içine yerleştirilir ve buradaki sıcaklık ölçülür. Bu bloğun sıcaklığı önemli değildir çünkü bu blokta akım ters yönde oluşur. Voltmetrenin ölçtüğü voltaj değeri

$$V = \alpha \left(T - T_{izo}\right), \qquad (1)$$

şeklinde olmaktadır.

Çarpan jet yardımı ile ısıtılmış bir yüzeyin soğutulması deneylerine başlamadan önce jet hızının belirlenmesi gerekmektedir. Pitot ve statik tüpleri ile ayrı ayrı toplam basınç ve statik basınçlar bir basınç transduseri yardımı ile jet hızı eksenel ve radyal dogrultularda ölçülmüştür. Ayrıca maksımum hız değerinin değişimi ölçülmüştür. Ortalama hız 20 m/sn, 15 m/sn ve 10 m/sn olarak alınmış, boru çapı 8.5 mm ve kinematik viskozite, ν, 1.455×10^{-3} m^2/sn olmak üzere deneylerin yapıldıgı Reynolds sayıları 11684, 8763 ve 5642 değerlerindedir. Sıcaklık ölçümü deneylerinde hava jetinin çarptırıldığı yüzey olarak kolay işlenilmesi, paslanmaya karşı dayanıklı oluşu nedeniyle bakır malzeme kullanılmıştır.

Bakır levhanın içine 4 tane silindirik ısıtıcı, üniform bir ısı alanı elde etmek için birbirine paralel yerleştirilmiştir. Isıtıcılar dimerler sayesinde gerilimleri ayarlanıp bir voltmetre ile verilen voltaj kontrol edilerek bakır levha üzerindeki sıcaklık dağılımı uniform duruma getirilmiştir. Bu dağılımın hızlı bir şekilde üniform hale gelebilmesi için ısıtıcılar birbirlerine seri olarak bağlanmalıdır.

Isılçıftlerde meydana gelen voltajın değeri doğrudan ölçümler için çok ufak değerlerdedir. Bu nedenle deneylerde 16 kanallı bir çoğaltıcı (amplificator) kullanılmıştır. Çoğaltıcı-ısılçift bağlantı noktaları stroafomdan yapılan bir kutucuk ile muhafaza edilerek bu bağlantı noktalarındaki sıcaklık ayrıca kontrol edilmiştir. Deneylerde kullanılan çoğaltıcı, ısılçiftlerdeki voltaj değerini 50 kat arttırılabilmektedir.

Deneylerde analog-dijital dönüşüm için PCL-818H kontrol kartı kullanılmıştır. Voltaj değerleri ± 0.625 volt arasında ölçeklendirilmiştir. Dolayısı ile 1 bit`lik değişim 0.305 mmVolt değerine karşılık gelmektedir.

Elektronik çoğaltıcıdan (amplifier) gelen voltaj değerlerini algılayabilmek için kullanılan kartın örnek programlarından bir takım modifikasyonlarla yararlanılmıştır. Program, 16 kanaldan okumayı yazılım tetiklemesi (software trigger) ile yapmaktadir.

Sıcaklık değişimlerini ölçmek için K tipi (Ni-Cr) ısılçiftler kullanilmıştır. Ölçülen voltaj degerlerine karşılık sıcaklık değerleri IPTS-1968 tablosundan yararlanarak 0°C-200°C değerleri arasında 5°C`er derece arasında, bu değerlere karşılık gelen voltaj degerleri arasında bağıntı regresyon analizi sonucunda

$$T = 0.0233V^3 - 0.239V^2 + 24.996V + 0.151 \, , \qquad (2)$$

şeklinde bulunmuştur.

Levhanın yüzey sıcaklık dağılımı üniform bir hale geldikten sonra kompresörden gelen sıkıştırılmış hava basınç regülatöründe toplam basıncı ayarlanarak lüleye gönderilir. Lüle 8.5 mm çapında ve 486 mm uzunluğundadır.

Jet yardımı ile soğutma veya ısıtma araştırmalarında Nusselt sayısının belirlenmesi öncelikli problemlerden birisidir. Nusselt sayısı karakteristik x uzunluğunun $\delta = k/h$ sınır tabaka kalınlığına oranı olup,

$$Nu = \frac{x}{k/h} = \frac{h\,x}{k} \qquad (3)$$

şeklinde formüle edilir. Nusselt sayısı yüzeyde taşınım yolu ile ısı geçişinin iletim yolu ile ısı geçisine oranı olmaktadır. Eger farklı sistemlerde Nusselt sayıları eşit ise sistemlerin yüzeylerinde ısı taşınımı aynı olmaktadır.

Çarpan jet yardımı ile soğutma araştırmalarında Nusselt sayısı genelde, Reynolds sayısina, Prandtl sayısına, lüle sayısina, lüle ile yüzey arasındaki mesafeye, lüle ile yüzey arasındaki açıya, yüzeyin pürüzlülüğüne bağlı olmaktadır.

Akışkanin ısi geçişi özelliğini gösteren Prandtl sayısi kinematik viskozitenin ısısal yayılma katsayısına oranı olup

$$Pr = \frac{\mu/\rho}{k/\rho c_p} = \frac{\mu\, c_p}{k} \quad , \qquad (4)$$

şeklinde gösterilebilir. Eğer farklı sistemlerde Prandtl sayıları aynı değerde ise bu sistemlerdeki sıcaklık dağılımları da ayni olmaktadır.

Sıcaklık ölçümlerinden yüzey sicaklıkları, T_W, jet sicaklığı, T_j ve verilen enerji q belli olduğundan ısı transfer katsayısı h

$$h = q/(T_W - T_j) \qquad (5)$$

her bir istasyon için bulunur. Isı transfer katsayısı Nusselt sayısı formülünde yerine konularak o noktada jet etkisi ile oluşturulan konveksiyon ısı transferinin şiddeti lüle ile hesaplanmış olur.

Deney sonuçlarına göre maksimum Nusselt sayısı durma noktasında meydana gelmektedir. Reynolds sayılarının küçük değerlerde olmasından Nusselt sayılarında ikincil maksimum noktaları gözlenmemiştir. Reynolds sayısının arttırılması ile yerel Nusselt sayılarında artış olur. Ancak Reynolds ve lüle ile yüzey arasındaki mesafeden bağımsız olan bir bölge bulunmaktadır. Bu nokta yaklaşık olarak çapın 6.5 katı değerinde başlamakta ve bu noktadan sonra tüm Reynolds sayıları için yaklaşık olarak aynı değeri almaktadır.

Reynolds sayılari ve lüle-yüzey arasındaki boyutsuz mesafeye göre yerel Nusselt ve durma noktasındaki Nusselt sayilari arasındaki oran aynı değerdeki Re sayıları için uyum içinde olduğu gözlemlenmiştir.

CHAPTER 1

INTRODUCTION

1.1 The Jet Impingement Phenomenon

In many fields of industry, making convenient and productive conditions for the cooling processes some of the main engineering problems. Cooling process can be accomplished by various ways that are appropriate for the goals of engineering design. One of the ways is the jet impingement method. In this technique a jet of liquid or gas, impinges on a surface. For an appropriate design, the impinging jet can have various jet configurations, temperatures, angles, and various nozzle numbers. This is a very active method so that its applications can be found in many areas of engineering. Cooling turbine blades and vanes, tempering glass plates, annealing metals shuts, drying textile and paper products, cooling electronic components are some of them

The phenomenon of jet impingement consists of three parts; free jet, stagnation or impingement, and wall jet regions [1]. A typical jet impingement is shown in Fig. 1.1.

First region is the free jet zone. There are mainly two type nozzles used to obtain jets [2]. One of these is the tube-type nozzles, which can make large degree of mixing with surrounding air and the other is a simple orifice placed to a wall parallel to the target (or impingement) surface. In the second type, mixing of ambient air is restricted. At increased distances from the nozzle, the momentum transfer between the jet and ambient causes the free boundary of the jet to broaden. Downstream of the potential core, the velocity profile is non-uniform and the maximum velocity decreases with increasing distance from the nozzle exit. This region is affected by the impingement surface [1]. Averaged heat transfer, while inside of the potential core, is essentially independent of the nozzle-to-plate spacing, but is influenced by the condition of the target plate`s surface and is inversely proportional to the target plate [2].

In the second region, that is stagnation or impingement zone, flow is influenced by the impingement and rapidly decelerated in the normal (z) and the transverse (x) directions, respectively. Average Nusselt and maximum Nusselt numbers are strongly functions of nozzle-plate spacings and lateral distance. At lower Reynolds numbers or at higher nozzle-plate spacings, maximum Nusselt number always occurred at the stagnation point [3]. When nozzle-plate spacings are greater than the nozzle diameter, both average Nusselt and maximum Nusselt numbers increase with increasing Reynolds numbers, but decrease with lateral distance. Maximum Nusselt number is strongly dependent on nozzle-plate spacings and lateral distance. The value of nozzle-plate spacings corresponding to maximum Nusselt number is a strong function of lateral distance [3].

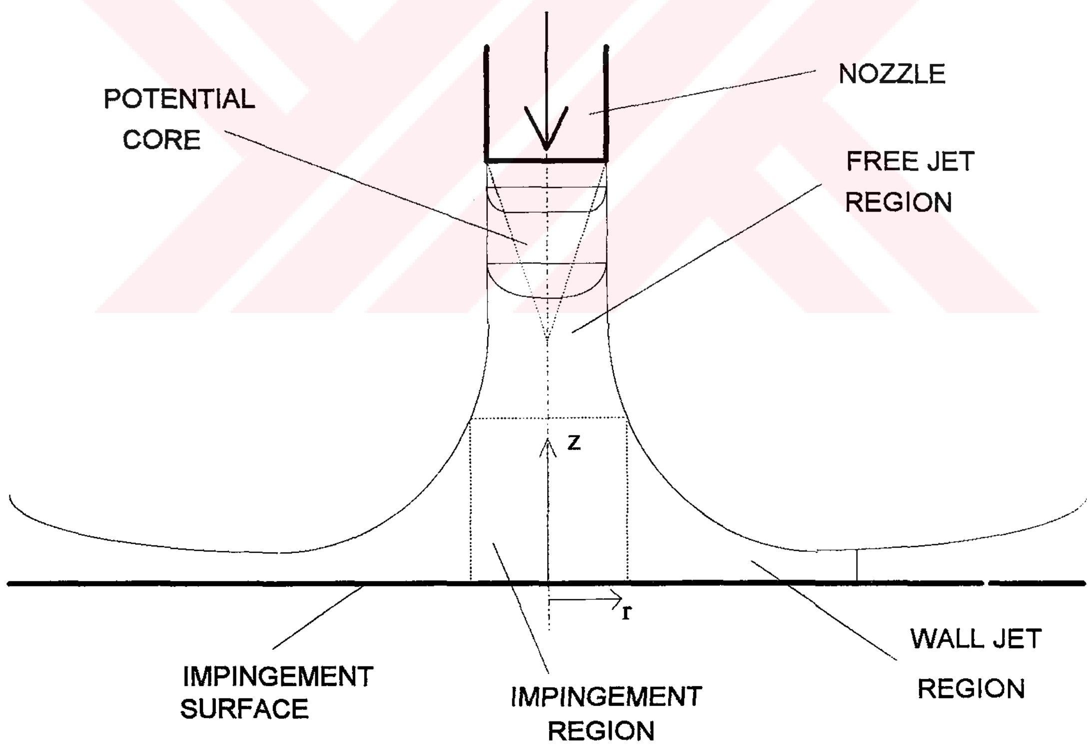

Fig. 1.1 The Structure of a Typical Jet Impingement

In the wall jet zone, since the flow continues to entrain zero momentum fluid from the ambient, horizontal acceleration can not continue indefinitely and thus acceleration flow in the stagnation zone is transformed to a decelerating wall jet. Hence, with increasing r, velocity components parallel to the surface increase from a value of zero to some maximum and subsequently decay to zero value again [1]. Convection heat transfer occurs surely in both stagnation and wall jet regions. In these zone, for high Reynolds numbers, the heat transfer rate exhibits a local peak towards the plate edge because of the tendency of the jet to reattach far from the point of impingement [4]. In general, the flow near the stagnation point is turbulent, resulting large local heat transfer coefficient. For small nozzle-plate spacings, there is a certain region in which the flow parallel to the surface is accelerated. At the end of the accelerated flow region, the disappearance of the stabilizing streamwise pressure gradient leads to a sudden rise in the turbulence level, causing a sharp increase in the heat transfer coefficient, which may be higher than the one at the stagnation point [3].

Multiple nozzles may be used when larger surface is to be cooled. There are basic differences between a single and multiple jets [1]. The first of these is the possible interference between adjacent jets prior to their impingement on surface. The likelihood of such interference effects is enhanced when the jets are closely spaced and when the separation distance between the jet orifices and the impingement plate is relatively large. The second difference is the collision of the surface flows (i.e. wall jets) associated with adjacent impinged jets. These collisions are expected to be of increased importance when the jets are closely spaced, the jet-orifice impingement plate separation is small, and the jet velocity is large.

One way to increase thrust per unit air flow is to increase the turbine inlet temperature. Fuel consumption also improves if the increased temperature is accompanied by an increased compressor pressure ratio. However there is limit about the turbine inlet temperature because of the insufficient thermal resistance of the turbine blades and vanes material. Over the years, there has been a gradual improvement in materials, permitting small increases in temperature in new engines but it is not sufficient. As a result, this problem has been solved by cooling turbine blades and vanes with impinging air jets and film jets. Fig. 1.2 shows the trend of turbine inlet temperature over the years for the air cooled and uncooled types.

As shown in Fig.1.2, the air-cooled turbines were introduced around 1960 with a small initial increase in turbine inlet temperature, T_{t4} and increased the rate of improvement with time. Todays commercial transport engines such as PW 4000, GE 90 can have turbine inlet temperatures at takeoff power near 1650°K [9]. The turbine inlet temperature is lower during cruise. Military engines may use significantly higher values certainly.

Air, which cools the turbine blades and vanes, is bled from the compressor and carried the aft and introduced into the turbine blades and vanes through their roots, as shown in Fig.1.3. Impinging air cools the rim of the turbine disc as skecthed in Fig.1.4. Impingement cooling applied to the leading edge of a turbine blades is sketched in Fig 1.5.

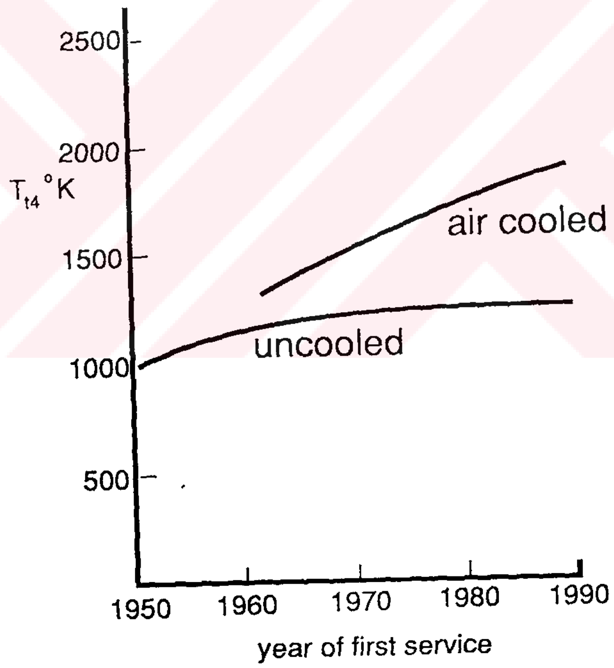

Fig.1.2 Trend of turbine inlet temperature with time [9]

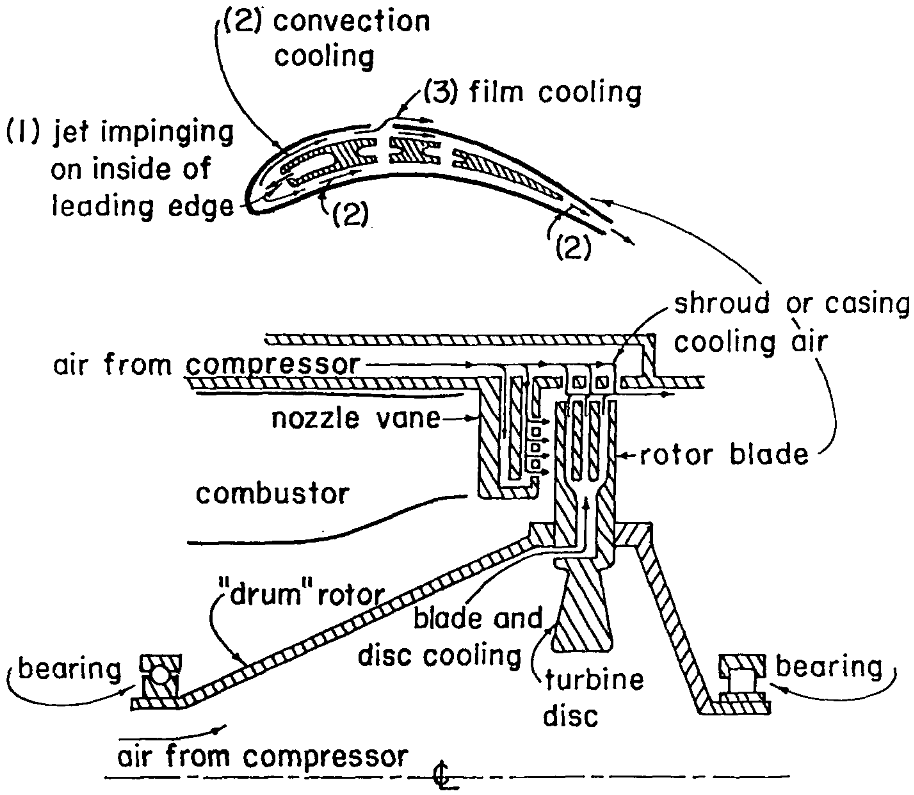

Fig.1.3 Schematic of air-cooled turbine, with cross section of cooled airfoil section at top [9]

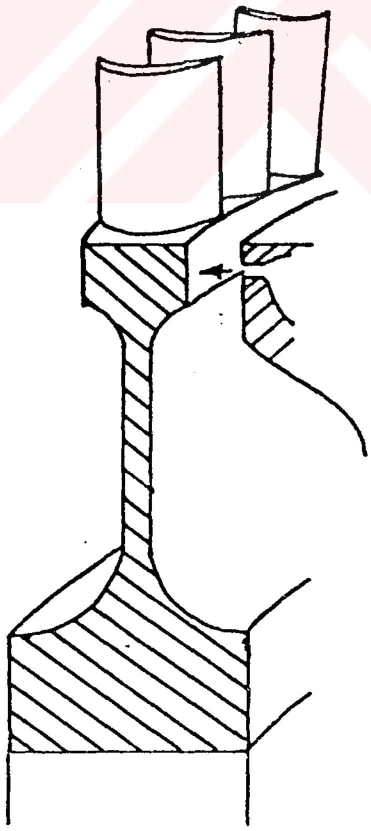

Fig.1.4 The cooling of disk rim [10]

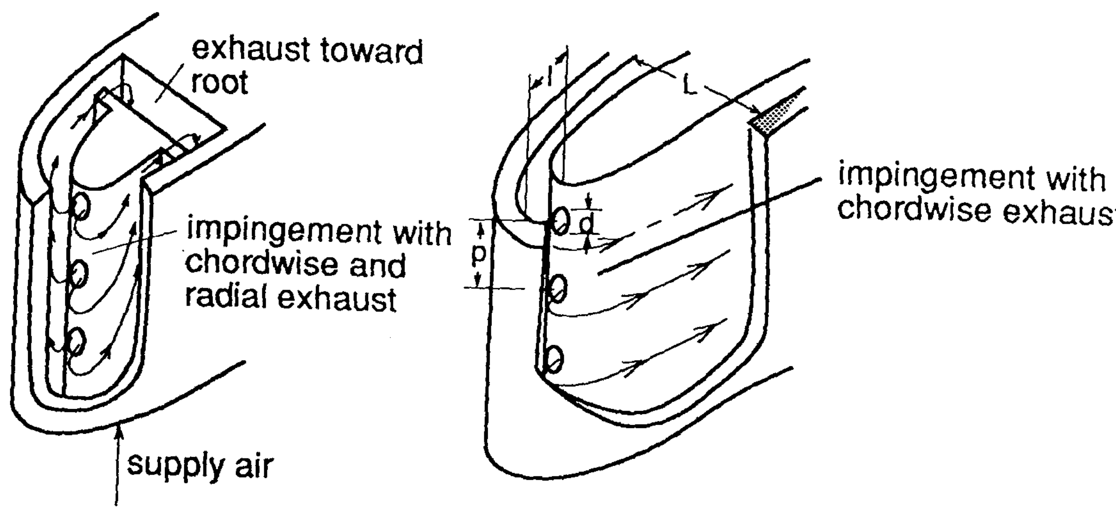

Fig 1.5 Schematics of impingement cooling applied to the leading edge of a turbine blades [9]

Cooling of the first stage turbine vanes and blades are sufficient. In the phenomenon of the cooling turbine blades and vanes cold air coming from the compressor, impinges to leading edge in order to decrease to higher temperatures there. After the air being impinged, it goes to internal blade surface chanells so that convective cooling can occur. After this cooling, the attemp is to sheath the blade with a film of cool air which is introduced through holes.

1.2 Basis of Temperature Measurements by Thermocouples

The temperature measurements in the researches of heat transfer is very important. There are mainly four common temperature transducers available; i.e

i) The Thermocouples

ii) The Resistance Temperature Dedectors (RTD)

iii) The Thermistors

iv) The Integrated Circuit Sensors (IC).

The advantages and disadvantages of these temperature sensors are presented in Table 1.1.

When two wires composed of dissimilar metals are joined from their ends and one of the end is heated, there occurs a continuos current which flows through circuit as shown in Fig. 1.6.

If this circuit is broken at the center, the net open circuit voltage (the Seebeck voltage) is a function of the junction temperature and the composition of the two metals.

Table 1.1 Advantages and disadvantages of temperature sensors

THERMOCOUPLE	RTD	THERMISTOR	I.C
advantages			
self-powered	most stable	high output	most linear
simple	most accurate	fast	highest output
rugged	more linear than thermocouple	two wire ohms measurement	inexpensive
inexpensive			
wide variety			
wide temperature range			
disadvantages			
non linear	expensive	non linear	T<200°C
low voltagesupply required	current source required	limited temperature range	power supply required
reference required	small difference of resistance	fragile	slow
	low absolute resistence	current source required	self heating
	self heating	self heating	limited configuration

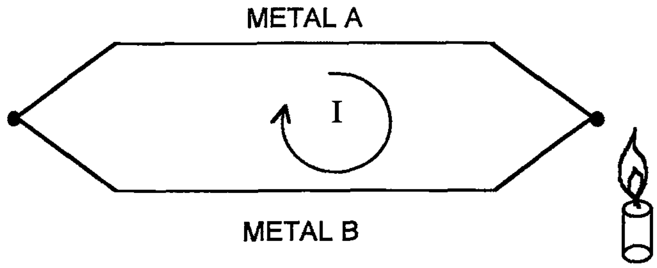

Fig.1.6 The Seebeck Effect

All dissimilar metals have this effect. For small changes in temperature, the Seebeck voltage, Δe_{AB} is linearly proportional to the temperature difference,

$$\Delta e_{AB} = \alpha \Delta T \tag{1.1}$$

where α is the Seebeck coefficient.

It is very difficult to measure the Seebeck voltage directly because the voltmeter creates a thermoelectric circuit when the thermocouple connected to the voltmeter. Voltmeter-thermocouple connection is shown in Fig. 1.7.

Measuring voltage at the junction J_1 is the main objective, but two more metallic junctions J_2 and J_3 have to be considered. Since J_3 is a copper-to-copper junction, it creates no thermal electro-motor-force, J_2 is a copper-to-constantan junction which will add an electro-motor-force to oppose V_1. The resultant voltmeter reading V is proportional to the temperature difference between J_1 and J_2. The temperature at J1 can not be determined before the temperature at J_2 is found.

In order to determine temperature at J_2 one can put the junction into an ice bath, forcing its temperature to be 0°C and establishing J_2 as a reference junction. The voltmeter reading can be written as

$$V = \alpha T_{j1}. \tag{1.2}$$

This method is very accurate since the ice point temperature controlled precisely.

Thermocouples are much more rugged than thermistors. They can be manufactured on the spot, either by soldering or welding.

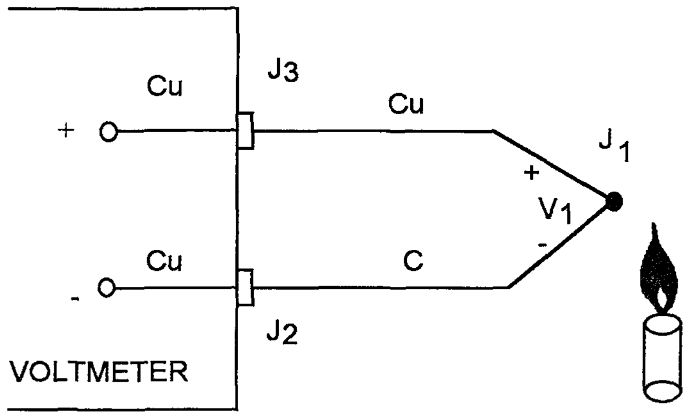

Fig 1.7 Measuring junction voltage with a digital voltmeter

Thermocouple output voltages are very small as shown at Table 1.2

Table 1.2 The Seebeck coefficient and digital voltmeter sensitivities required of thermocouples

Thermocouple Type	Seebeck coefficient (μV/°C) at 20°C	Digital voltmeter sensitivity for 0.1° C (μV)
E	62	6.2
J	51	5.1
K	40	4.0
R	7	0.7
S	7	0.7
T	40	4.0

1.3 Literature Survey

Hrycak [2] presented experimental results for heat transfer between a flat plate and impinging turbulent air jets, for conditions applicable both inside and outside of the potential core, for the stagnation point heat flux as well as for the average heat transfer obtained through integration of local heat fluxes. The stagnation point results showed that the Nusselt number has a tendency to peak out about seven diameters away from the target plate, for tube-type nozzles.

Huang and El-genk [3] measured the radial variation in the local and average Nu numbers for a uniformly heated flat plate impinged by a circular air jet. In their experiments, Reynolds number varies from 6.10^3 to 6.10^4, dimensionless radial distance from stagnation point, R, from 0 to 10, and dimensionless jet-plate distance, H from 1 to 12. They showed that the maximum stagnation point Nusselt number occurred at H=4.7 and both Nusselt and stagnation point Nusselt numbers very strong functions of H and R. For H equals to 1 and Re is greater than $1.3.10^3$, maximum Nusselt number occurred at R=1.0-2.0 while at lower Reynolds number and higher H, maximum Nusselt number occurred at the stagnation point. For H greater than 1, both Nusselt and maximum Nusselt numbers increased with Re but decreased with R.

Lyttle and Webb [5] examined experimentally the local transfer characteristicsof air jet impingement at nozzle-plate spacings of less than one nozzle diameter using an infrared thermal imaging technique. They found that the outer peak in local Nusselt number move radially outward for larger nozzle-plate spacings and higher jet Reynolds numbers

Some proturberances can be used to increase cool of the heated plate. Hrycak [11] carry out heat transfer enhancing proturberances. His results showed a substantial increase in stagnation point heat transfer. As proturberances, modified surfaces can be used to increase cool. Hansen and Webb [12] performed to characterize heat transfer to a normally impinging jet from surfaces modified with fin type extensions.

CHAPTER 2

EXPERIMENTAL INVESTIGATION

2.1 Experimental Equipments

Experiments were carried out at Istanbul Technical University Gümüssuyu Aeronautical Laboratory. Experimental set up is shown in Fig.2.1. The ýnstruments used were explained at below sections.

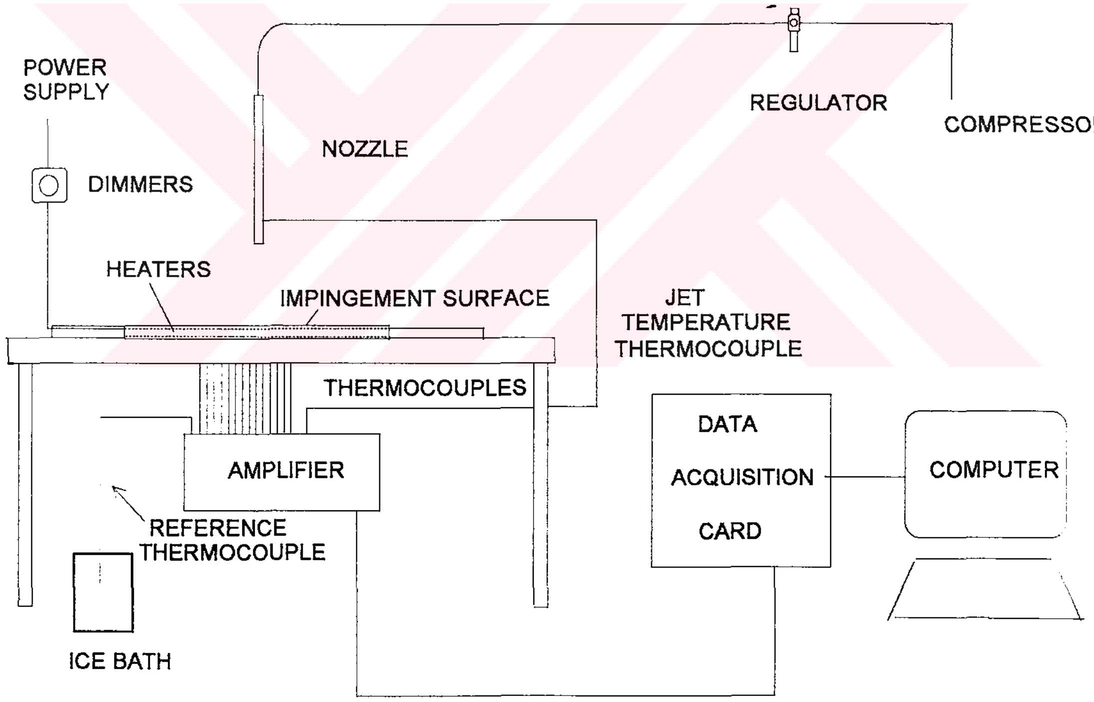

Fig. 2.1 Experimental set up.

2.1.1 Compressor and Pressure Regulator

A piston type compressor is used in order to supply continuos air and its features are given in Table 2.1

Table 2.1 Features of the compressor

Commercial name	Komsan
Pressure	1.4 bar
Stroke volume	0.015 m^3/sec
Revolution	1000 rpm
Engine power	7457 W

The compressed air is transferred to the pressure regulator by hose-pipe whose length is approximately twenty meters. The pressure losses is not important because the pressure at the regulator is taken into consideration. Because of the long experimental time (about three hours) pressure in the regulator is controlled continuously.

The regulator is used also to control the air coming from the compressor. Also it filters particles and water from the air.

2.1.2 Nozzle

Schematic diagram of the nozzle used in the experiments is shown in Fig. 2.2. It is made from a water pipe and is connected to the pressure regulator by an apparatus. The nozzle was placed perpendicularly to the heated plate by a supports made by L-profiles

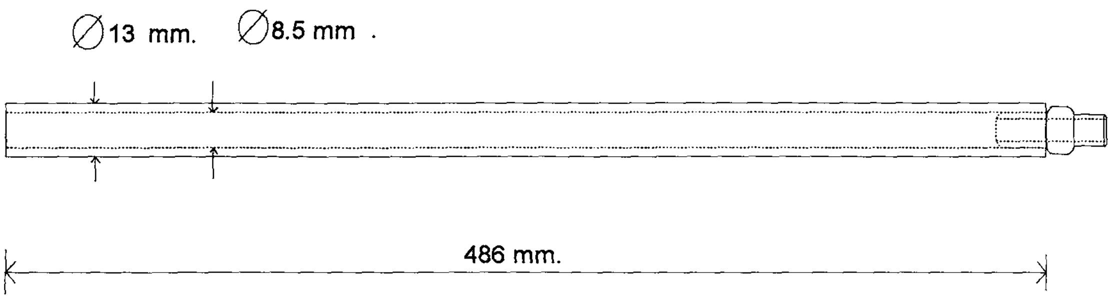

Fig. 2.2 The nozzle used in the experiments

2.1.3 Impingement Surface

Impingement surface was made of copper material. The reason of selecting this material is due to its resistance to corrosion. Also, it can be shaped very easily.

Ten holes were drilled to the bottom of plate surface until 1.5 mm. left to the top in order to measure surface temperature via thermocouple wires. The four sides and the bottom were ýsolatated with bricks. Thermal conductivity of the isolation material was 0.5 kcal/(m2h°C). There are also four holes for heaters. The dimensions of the plate are shown in Fig.2.3 and the positions of the thermocouple holes are given in Table 2.2.

2.1.4 Heaters

The geometrical values of the heaters used in the experiments are presented in Fig. 2.4.

The electrical resistance of the heaters are choosed differently. Heaters close to surface were selected 197Ω and others were selected 195Ω. The heaters were connected in a series in order to reach uniform temperature distribution quickly. The voltage of the heaters was controlled by two dimmers. The heat given to surface is balanced by controlling heaters voltage.

2.1.5 Thermocouples

The K-type thermocouple wires used in this study. These wires were made up of Nickel and Chromium. The ends of the wires were welded carefully by on arc in a mercury bath. The diameter of the thermocouples is 1mm. The connection points of the thermocouples were sticked on the plate by epoxy-glass adhesive.

2.1.6 Amplifier

The voltage occurred on thermocouples is very low to measure directly. For example this value is 8.137 mm Volt for 200 °C [8]. To increase this low voltages to perceive electronic equipments, an amplifier unit is used. The connections between the amplifier and thermocouple wires are closed with the Styrofoam box to control temperature at these points.

In this study, the 16 channels thermocouple amplifier was used. Their duties were presented in Table 2.3. Its operational amplifier capability is 50. For this reason, the voltage values read are divided by 50 to read from the International Pressure Temperature Scale (IPTS-1968) table. The relation between the voltages and temperatures for the range of 0 °C and 200 °C is calculated from the relation

$$T=0.0233V^3-0.239V^2+24.996V+0.151, \tag{2.1}$$

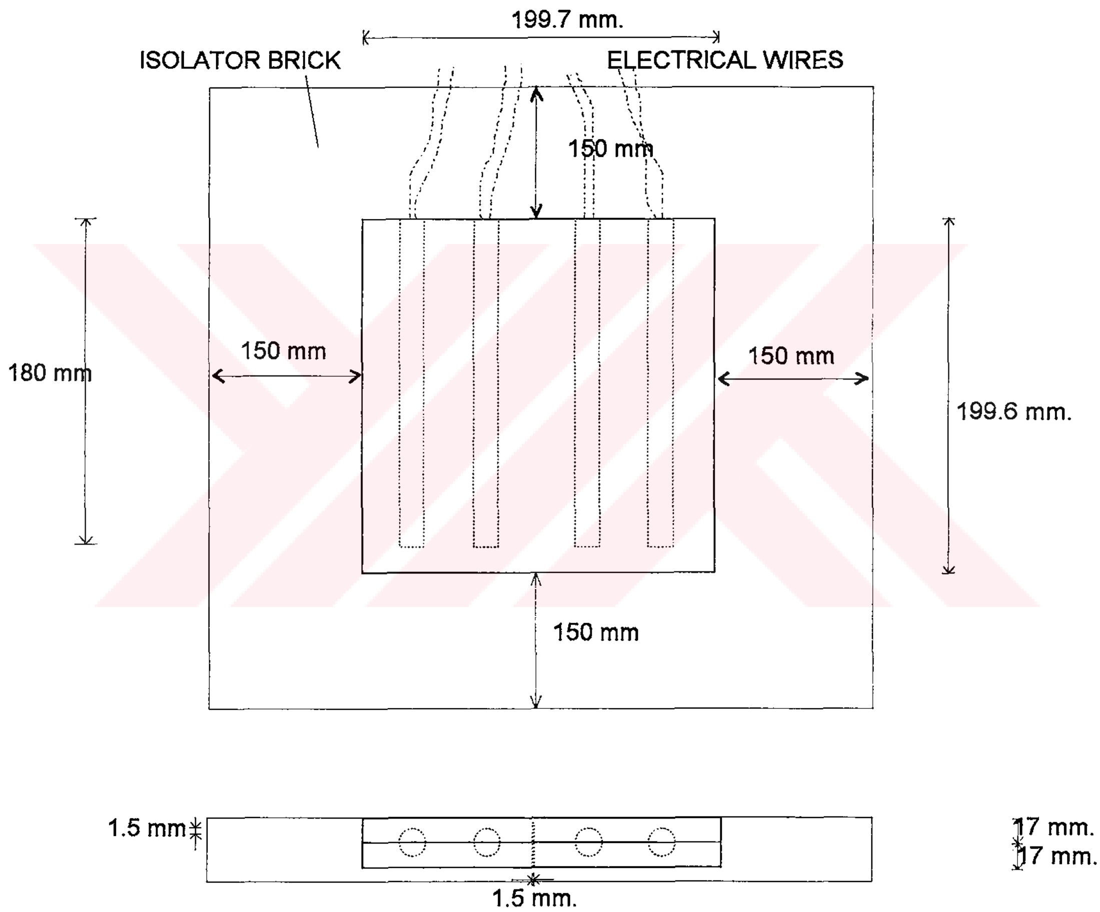

Fig.2.3 Impingement Surface

Table 2.2. Channel Positions on the Surface

POSITION (in mm.)	0	7.4	13.7	27	33	40.5	47	54	60	68	87
CHANNEL NUMBER	3	4	5	6	7	8	9	10	11	12	13

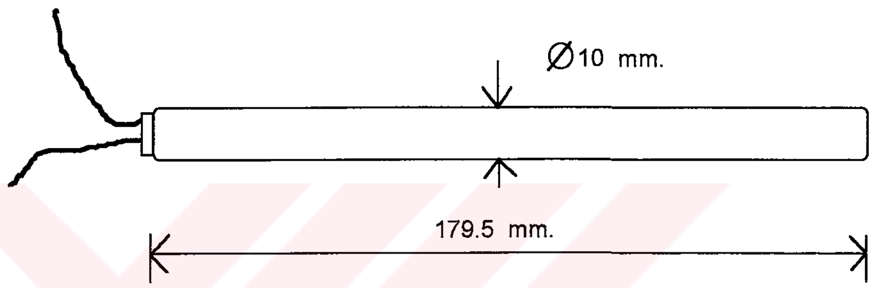

Fig. 2.4 Heaters

Table 2.3 Channel numbers and their duties

CHANNEL NO.	DUTIES
0	REFERENCE CHANNEL
1	JET TEMPERATURE
2	ATMOSPHERE TEMP.
3 - 13	SURFACE TEMP.
14	ISOTHERMAL TEMP.
15	CONTROLLING OF DISTRIBUTION TEMP. OR ISOLATION TEMP.

2.1.7 Data Acquisition Card and Computer

As a data acquisition system, PCL-818H card was used. This card has high performance quantities such as, high speed, and is appropriate for multi-functional data acquisition and can be used with IBM compatible computers. The computer used in this study was 486-DX-66 .

2.2 Velocity Experiments

At first, measurements of maximum nozzle exit velocity were carried out. The pressure regulator was adjusted to different pressure levels and thus nozzle exit velocities can be measured using a manometer.

In the experiments pressure transducer, pitot and static tubes were used for measuring jet velocities. Static pressure was approximately takenas atmospheric pressure.

$$P - P_{atm} = \rho V^2 / 2 \quad . \tag{2.2}$$

2.3 Temperature Experiments

The heat flux given to plate, surface and jet temperatures, and heat losses must be known to research experimentally for cooling of heated surface by impingement jet. Thus effect of the jet on cooling of the plate can be determined.

2.3.1 Measurement of the Surface Heat Flux

The impingement surface is heated by four cylindrical heaters. Heaters are positioned parallel to the each other to obtain a uniform surface temperature distribution. Heaters with same electrical resistance are selected and connected as a serial to reach quickly to the uniform surface temperature. The electrical connections are shown in Fig. 2.6. The voltages are balanced by two dimmers and controlled by a voltmeter. Because the heaters are serially connected, the resistance must be added. The total electrical resistance of first part is 197Ω +197Ω = 394Ω and second is 195Ω +

195Ω = 390Ω. The power to the each heater systems can be calculated using the relation:

$$P = V_1^2/R_1 + V_2^2/R_2 \qquad (2.3)$$

where P is the power given to system, V is the voltage measured by the voltmeter and R is the electrical resistance. As a result, the heat flux can be obtained as

$$q = P/A , \qquad (2.4)$$

where A is the impingement surface area.

At first, the surface temperature has to come to uniform temperature distribution . A thermocouple wire touched to the different locations on the plate to control whether it has come to the uniform distribution. The uniform temperature was selected as 120°C.

After the surface temperature uniformity were obtained, the jet was impinged to the plate. Approximately three hours later, surface temperature has come to uniform distribution and that time the voltage values were read from the digital voltmeter. The inner and outer voltage values were measured as V_1=120 volt, V_2=100 volt. As a result, heat flux given to the system is q=1.6 kw/m2 using eqs. 2.3 and 2.4.

There were heat losses due to conduction and radiation. Because of the maximum surface temperature is approximately 40 °C at the uniform condition, the radiation heat loss was insignificant. To calculate the conduction heat loss, the isolator temperatures were measured two points whose interval is ten milimeter for four sides of the isolator as shown in Fig. 2.6.

From the equation of heat conduction,

$$q_{cond} = -k\, \Delta T/\Delta x \qquad (2.5)$$

can be written. Using eq.(2.5) conduction heat loss is calculated as 0.5 kw/m^2.

As a result, the net heat flux given to the system is 1.1 kw/m^2.

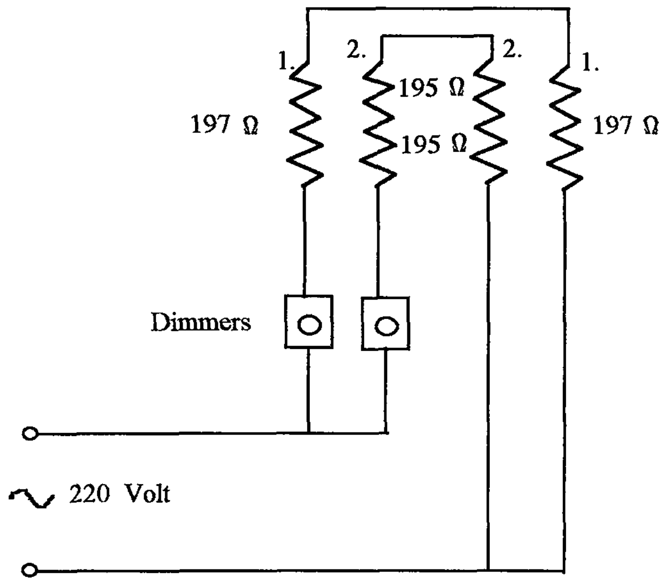

Fig. 2.5 Electrical connection of heaters

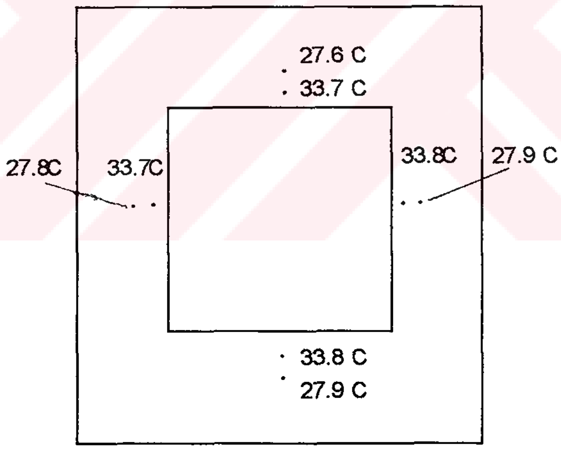

Fig. 2.6 Temperatures on the isolator

2.3.2 Measurement of Wall Temperature

Until the surface temperature reaches to the uniform distribution, the voltage given the heaters was balanced by the dimmers. This process was accomplished by a computer software. Program was prepared using PCL-818H demonstration programs. In this program trigger mode is software type and collection time intervals are 10 min., while coming to the uniform surface temperature distribution.

The first channel was taken as reference and during the experiments the thermocouple in this channel was put to an ice bath. Values which were read from the other channels were removed from the value read from this reference channel.

2.3.3 Measurement of the Jet Temperature

A thermocouple was placed to 10 cm distance from the nozzle exit in order to measure jet temperature. Since the velocities were low, static temperature was approximately equal to the free-stream temperature.

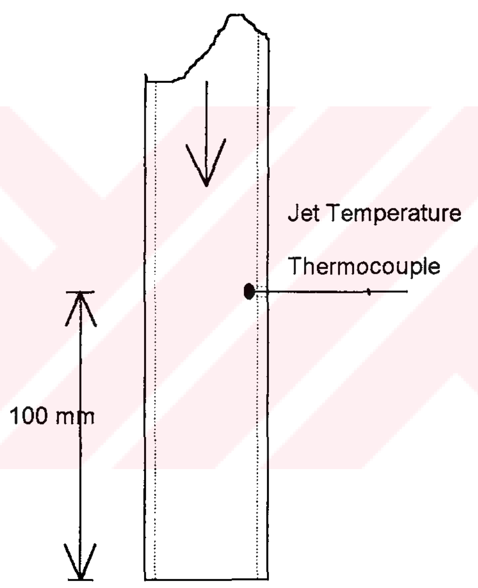

Fig.2.7 The position of the Jet Temperature Thermocouple

2.3.4 Calculation of Nusselt Number

The Nusselt number is very important dimensionless parameter for convection heat transfer phenomena Nusselt number can be defined as

$$Nu = h\, D/k_{air} \tag{2.6}$$

where D and k show the diameter of the nozzle and the thermal conductivity, respectively.

To acquit the voltage values coming from the amplifier, the program uses the software trigger. The plate temperature was controlled using this program. The program acquits voltage values by 10 minutes interval. Whenever the uniform surface temperature distribution was reached, pressure regulator was opened in accordance with the previously scaled pressure levels.

CHAPTER 3

RESULTS AND CONCLUSION

3.1 Results

Experiments were performed at three Reynolds numbers (5842, 8763 and 11684) and dimensionless nozzle-plate distance was 2, 4, and 4.

As a result the local Nusselt numbers are presented with respect to dimensionless locations, R, in Fig. 3.1, 3.2 and 3.3.

In Fig.3.1, the characteristics of the local Nusselt number curves with respect to the diamensionless locations look like bell-curves. With increasing Reynolds number, the local Nusselt number is also increases. At Reynolds number, 5842 and 8763, the local Nusselt number is almost equal each other at dimensionless locations, 4.8. At dimensionless locations, 6.4, all local Nusselt numbers are equal too each other and their values are approximately as 41.

When the nozzle-plate distance is 4 times the nozzle diameter (Fig. 3.2) stagnation Nusselt number begin to decrease. That decreasing is lower at 5842 Reynolds number. At Reynolds number, 5842 and 8763, Nusselt numbers are equal at 3.8 and Nusselt numbers at all Reynolds numbers are equal to each other at 7.2 .

At Fig.3.3, although increased nozzle-plate spacings, Nusselt number is not effected significantly at Reynolds number, 5842 and 8763. Their Nusselt numbers are equal appoximately at the same dimensionless location, 3.8.

If the dimensionless nozzle-plate spacings is increased, Nusselt numbers at the low Reynolds numbers (5842 and 8763), did not change significantly. At low Reynolds numbers and high nozzle-plate distances, these curves were broaden.

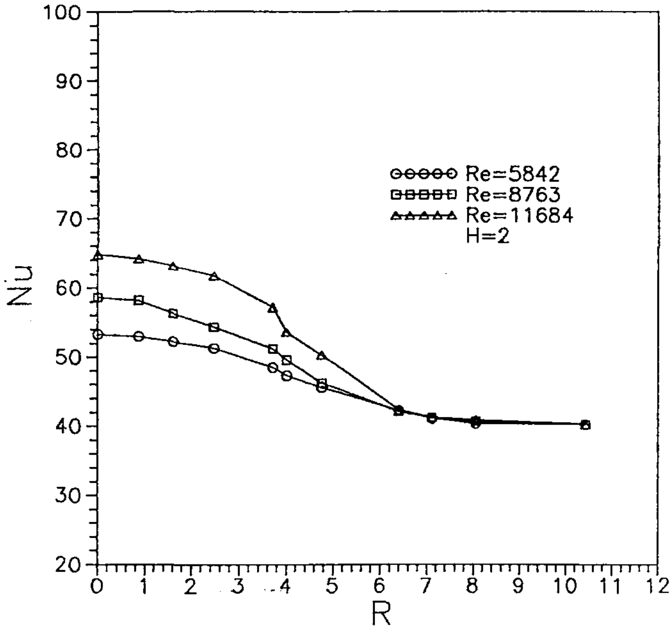

Fig. 3.1 The local Nusselt number variations with respect to R at H=2

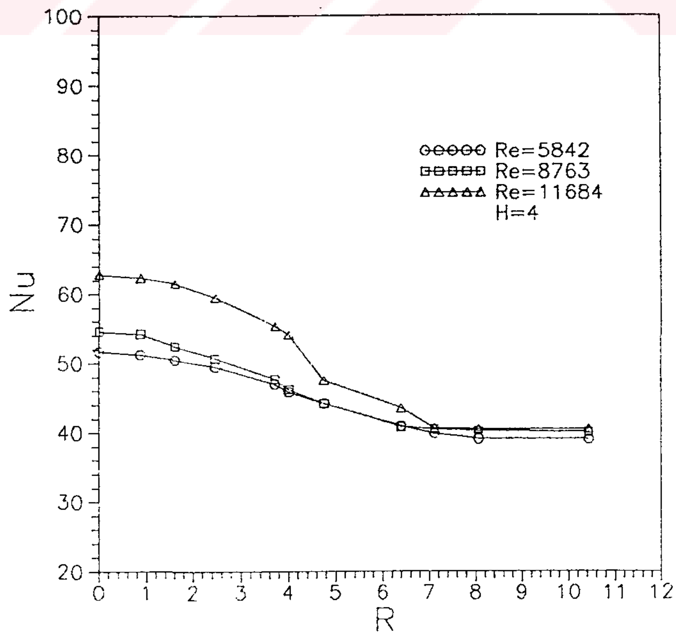

Fig. 3.1 The local Nusselt number variations with respect to R at H=2

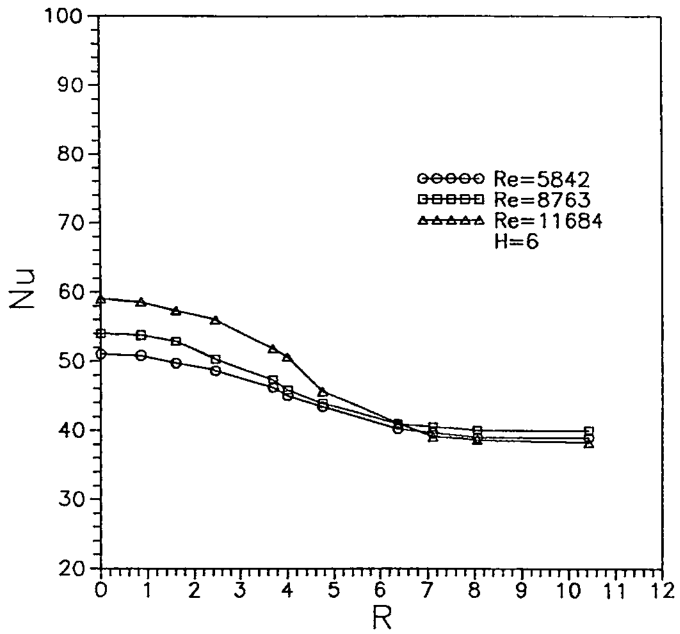

Fig. 3.3 The local Nusselt number variations with respect to R at H=6

Because of low Re numbers, there is no significant change of the stagnation Nu number as shown in Table 3.1. For example between the Reynolds number 5842 and 11684 the change is 22 %.

Table 3.1 The values of stagnation point Nusselt number

	Dimensionless nozzle-plate distance,H		
Re	***2***	***4***	***6***
5842	53.3	51.5	51.0
8763	58.6	54.6	54.0
11684	64.8	62.8	59.1

In Fig. 3.4, variation of the averaged Nusselt numbers with respect to dimensionless nozzle-plate spacing,H is given. At Re=11684, average Nusselt number decay is higher according to H.

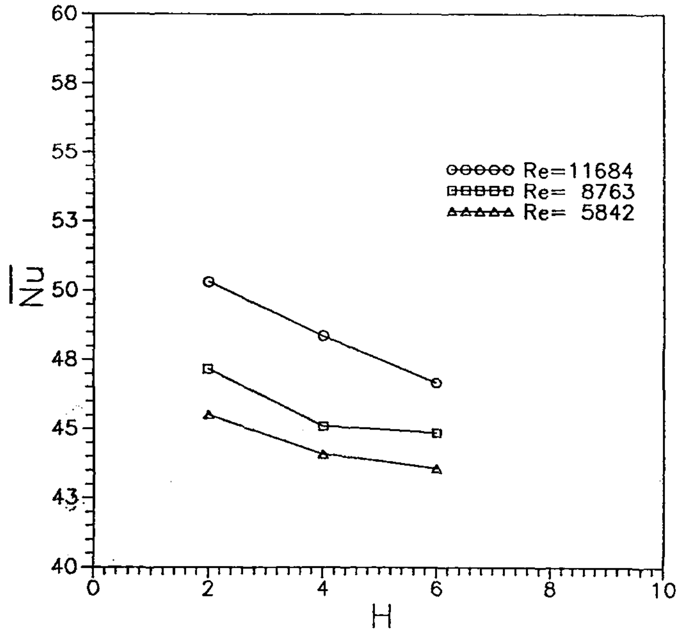

Fig. 3.4 Average Nusselt number variations with dimensionless nozzle-plate spacings,H

As it can be seen, average Nusselt number decreases with increasing H. The value of average Nusselt numbers is presented in Table 3.2.

Table 3.2 The values of the average Nusselt number

	Re		
H	***5842***	***8763***	***11684***
2	45.52	47.20	50.32
4	44.07	45.10	48.37
6	43.57	44.86	46.67

In fig. 3.5, the ratio of local Nu to the stagnation point Nu numbers with respect to R is presented. There is a well adaptation for same Re numbers for H=2, 4, and 6 but for different Re numbers this adaptation take place until R=1.6. After this point there is no adaptation to cover all values. This can be explained that at low Re, the effect of the change of Re numbers is more dominate than that of H.

In Fig.3.6., with increased H, the stagnation point Nusselt number decreases. This is minimum for Re=5842. If Re number is increased, it is higher.

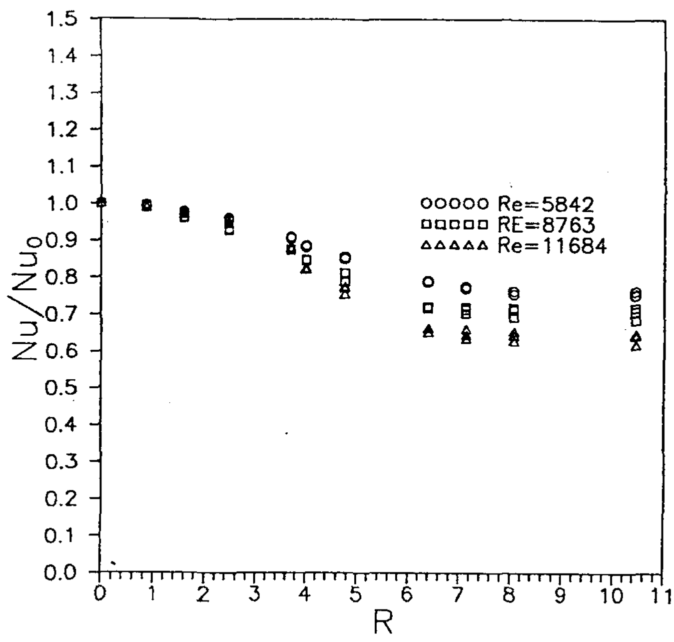

Fig. 3.5. The ratio Nu to the stagnation point Nusselt numbers with respect to R.

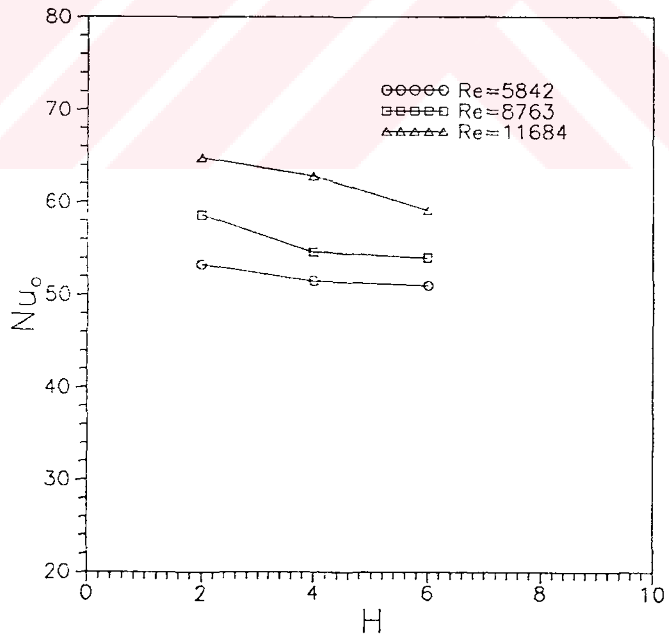

Fig. 3.6. Stagnation Nusselt number variations with respect to H

3.2 Conclusion

In this study, cooling of the air impingment phonemenon was studied experimentally. Experiments were performed at the Re=5842, 8763, and 11684 based on the average jet velocity and dimensionless nozzle-plate spacings,H, as 2,4, 6. Reynolds numbers were selected at low values.

As a result , for low Reynolds numbers and high nozzle-plate spacings, the little change of the local Numbers were observed. When R is higher than 6.5, the local Nusselt number is approximately constant and its value is about 41. The average Nu number decreased when H was increased.

In the cooling processes at low Reynolds numbers, the Nusselt number is much move important than nozzle to plate spacings.

The ratio of the loical Nusselt number to the staghation point Nusselt number is identical variations at the same Reynolds numbers and different H

Although the flow structure is important. It could not be measured because of the lack of the experimental equipments.

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APPENDIX A

THE TEMPERATURE DATA

Experiments were carried out at Re=5842, 8763, and 11684 and dimensionless nozzle-to-plate spacings,H=2, 4, and 6.

The power to the each heater systems can be calculated using the eq. (2.3). The heat flux can be obtained from eq. (2.4).

At first, the surface temperature has to come to uniform temperature distribution . A thermocouple wire touched to the different locations on the plate to control whether it has come to the uniform distribution. The uniform temperature was selected as 120°C.

After the surface temperature uniformity were obtained, the jet was impinged to the plate. Approximately three hours later, surface temperature has come to uniform distribution and that time the voltage values were read from the digital voltmeter. The inner and outer voltage values were measured as V_1=120 volt, V_2=100 volt. As a result, the net heat flux given to the system is q=1.1 kw/m^2 using eqs. 2.3 and 2.4. Heat transfer coefficient ,h, and Nusselt number, Nu, are calculated from eq. (2.5).

Table A.1 The values of Nu, h and ΔT variations with Re=5842, Tj=20.4°C, Tatm=23.5°C, and H=2

R	Nu	h	ΔT
0	53.29	146.10	7.50
0.87	53	145.28	7.54
1.61	52.23	143.17	7.65
2.47	51.29	140.62	7.79
3.71	48.5	132.94	8.24
4.01	47.29	129.65	8.45
4.76	45.59	124.99	8.77
6.39	42.20	115.67	9.47
7.12	41.09	112.66	9.73
8.06	40.29	110.46	9.92
10.44	40.20	110.19	9.95

Table A.2 The values of Nu, h and ΔT variations with Re=5842, Tj=20.4°C, Tatm=24.5°C, and H=4

R	Nu	h	ΔT
0	51.64	141.57	7.74
0.87	51.20	140.34	7.81
1.61	50.5	138.42	7.92
2.47	49.5	135.68	8.08
3.71	46.99	128.82	8.51
4.01	45.83	125.63	8.72
4.76	44.18	121.12	9.05
6.39	40.89	112.09	9.78
7.12	39.82	109.16	10.04
8.06	39.05	107.04	10.24
10.44	39.10	108.22	10.25

Table A.3 The values of Nu, h and ΔT variations with Re=5842, Tj=20.6°C, Tatm=23.5°C, and H=6

R	Nu	h	ΔT
0	51	139.80	7.84
0.87	50.79	139.25	7.87
1.61	49.70	136.23	8.04
2.47	48.59	133.22	8.23
3.71	46.20	126.64	8.65
4.01	45.04	123.48	8.87
4.76	43.42	119.04	9.21
6.39	40.20	110.19	9.95
7.12	39.59	108.55	10.10
8.06	39	106.90	10.25
10.44	39	106.90	10.25

Table A.4 The values of Nu, h and ΔT variations with Re=8763, Tj=20.4°C, Tatm=22.5°C, and H=2

R	Nu	h	ΔT
0	58.59	160.63	6.82
0.87	58.20	159.53	6.87
1.61	56.29	154.32	7.10
2.47	54.40	149.12	7.35
3.71	51.20	140.34	7.81
4.01	49.59	135.96	8.06
4.76	46.20	126.64	8.65
6.39	42.09	115.40	9.50
7.12	41.20	112.93	9.70
8.06	40.59	111.29	9.85
10.44	40.20	110.19	9.95

Table A.5 The values of Nu, h and ΔT variations with Re=8763, Tj=21.4°C, Tatm=21.9°C, and H=4

R	Nu	h	ΔT
0	54.59	149.66	7.32
0.87	54.22	148.64	7.37
1.61	52.45	143.79	7.62
2.47	50.68	138.93	7.89
3.71	47.70	130.76	8.38
4.01	46.21	126.67	8.65
4.76	44.20	121.16	9.05
6.39	40.79	111.83	9.80
7.12	40.5	111.01	9.87
8.06	40.20	110.19	9.95
10.44	40	109.64	10.00

Table A.6 The values of Nu, h and ΔT variations with Re=8763, Tj=19.5°C, Tatm=22.5°C, and H=6

R	Nu	h	ΔT
0	53.96	147.93	7.41
0.87	53.77	147.40	7.43
1.61	52.88	144.98	7.56
2.47	50.28	137.85	7.95
3.71	47.33	129.74	8.45
4.01	45.85	125.68	8.72
4.76	43.87	120.26	9.11
6.39	40.90	112.11	9.78
7.12	40.5	111.01	9.87
8.06	40	109.64	10.00
10.44	40	109.64	10.00

Table A.7 The values of Nu, h and ΔT variations with Re=11684, Tj=18.8°C, Tatm=22.5°C, and H=2

R	Nu	h	ΔT
0	64.80	177.62	6.17
0.87	64.19	175.98	6.23
1.61	63.20	173.24	6.32
2.47	61.79	169.40	6.47
3.71	57.20	156.79	6.99
4.01	53.59	146.92	7.46
4.76	50.29	137.88	7.95
6.39	42.29	115.95	9.45
7.12	41.20	112.93	9.70
8.06	40.79	111.83	9.80
10.44	40.20	110.19	9.95

Table A.8 The values of Nu, h and ΔT variations with Re=11684, Tj=19.4°C, Tatm=23.9°C, and H=4

R	Nu	h	ΔT
0	62.79	172.12	6.37
0.87	62.29	170.77	6.42
1.61	61.5	168.58	6.50
2.47	59.5	163.10	6.72
3.71	55.29	151.58	7.23
4.01	54.09	148.29	7.39
4.76	47.5	130.20	8.42
6.39	43.5	119.24	9.19
7.12	40.59	111.29	9.85
8.06	40.5	111.01	9.87
10.44	40.5	111.01	9.87

Table A.9 The values of Nu, h and ΔT variations with Re=11684, Tj=18.9°C, Tatm=23.5°C, and H=6

R	Nu	h	ΔT
0	59.04	161.85	6.77
0.87	58.57	160.55	6.82
1.61	57.32	157.13	6.97
2.47	56.05	153.65	7.13
3.71	51.88	142.21	7.70
4.01	50.66	138.88	7.89
4.76	45.62	125.06	8.76
6.39	41.04	112.51	9.74
7.12	39.04	107.03	10.24
8.06	38.66	105.99	10.34
10.44	38.28	104.94	10.44

CURRICULUM VITAE

Murat Kızılırmak was born in Edirne-TURKEY on July 24, 1971. He graduated from Edirne high school in 1988. He obtained his B.Sc.degree from Istanbul Technical University, Faculty of Aeronautics and Astraunatics, Department of Aeronautical Engineering in 1992. Same year he enrolled to M.Sc. program in Aeronautical Engineering of the Institute of Science and Technology at Istanbul Technical University.