 Convection heat transfer is a fundamental process that plays a critical role in various engineering and scientific applications. Understanding and optimizing convective heat transfer are essential for improving energy efficiency, thermal management, and system performance across a wide range of industries. This article provides an in-depth exploration of convection heat transfer, covering its fundamental principles, calculation methods, measurement techniques, and strategies for enhancement. Additionally, the article discusses the application of Computational Fluid Dynamics (CFD) in simulating convective heat transfer, as well as the different types of convection heat transfer mechanisms. By examining practical examples and strategies for improving convective heat transfer, this article aims to provide valuable insights into the complex interplay between fluid dynamics and heat transfer phenomena. What is convective heat transfer?Convection is the process of heat transfer through the movement and mixing of molecules within a liquid or gas. Convection heat transfer involves two mechanisms: energy transfer through random molecular motion (diffusion) and through the bulk motion of the fluid (advection). The latter occurs because large groups of molecules move collectively, contributing to heat transfer when theres a temperature gradient. This cumulative transport, involving both molecular and bulk motion, is commonly referred to as convection. In this article the term convection heat transfer is used to describe thermal energy transfer between a surface and a fluid moving over the surface. What is the formula of convection heat transfer?The rate of heat transfer q [W] between a surface and a fluid moving over it is calculated using equation 1. h is the convective heat transfer coefficient [W.m-2.K-2], A is the surface area through which heat is transferred [m2], and T the temperature difference between the surface and the fluid [K]. Convective heat transfer coefficient unitsThe units of convective heat transfer coefficient expressed in various systems:International System of Unit (SI): Watts per square meter per Kelvin (W.m-2.K-1)Imperial Units (British Engineering Unit): BTU per hour per square foot per degree Fahrenheit (BTU.h-1.ft-2.F-1)Metric Unit: Joule per second per square meter per Kelvin (J.s-1.m-2.K-1) or Watt per square meter per degree Celsius (W.m-2.C-1)International Unit: Kilocalorie (IT) per hour per square meter per degree Celsius (kcal.h-1.m-2.C-1)How to measure the convective heat transfer coefficient?Engineers, scientists, and physicists employ experimental methods to determine the convective heat transfer coefficient (h) in various situations. They measure the heat transferred between a surface and a fluid and then use appropriate equations to calculate h. At each specific point on the surface, the amount of h is referred to as the local convective heat transfer coefficient, while the average h across the entire surface is called the average convective heat transfer coefficient. By measuring local h at multiple points and utilizing numerical methods and computational fluid dynamics (CFD) simulations, equations can be derived for h that are specific to the fluid and surface conditions under consideration.Once sufficient data is gathered to formulate the appropriate equation for h, dimensionless numbers are employed to derive an equation applicable across different fluid types and surface geometries. The average Nusselt number serves as a crucial parameter in this process and is expressed in equation 2. where [W.m-2.K-1] is the averaged convection heat transfer coefficient, L [m] is a characteristic length or characteristic dimension associated with the flow geometry, and k is the thermal conductivity of the fluid [W.m-1.K-1].For instance, the average Nusselt number for flow over a cylinder is obtained from an equation proposed by Churchill and Bernstein as: Re and Pr are the Reynolds number and the Prandtl number, respectively. Therefore for flow over a cylinder, average convective heat transfer coefficient is calculated using equations 2 and 3.In heat transfer handbooks and scientific papers, equations and tables are often provided to determine Nusselt numbers for different surface geometries and fluid flow conditions. These resources offer valuable tools for engineers and researchers to analyze convective heat transfer across various scenarios. Convective heat transfer coefficient tableConvective heat transfer coefficient values are sometimes presented in tables within scientific literature and books. Its important to recognize that these tables are tailored to specific conditions of surface and flow geometry. For accurate interpretation, its essential to consider the precise details of the heat transfer scenario described in those tables.To estimate or gain an understanding of the magnitude of the convective heat transfer coefficient, refer to the numbers in Table 1.Table 1. Typical values of the convection heat transfer coefficient. From Incropera et al., Fundamentals of Heat and Mass Transfer, 6th edition, Wiley, 2007.Processh (W.m-2.K-1)Free convection Gases225 Liquids501000Forced convection Gases25250 Liquids10020,000Convection with phase change Boiling or condensation2500100,000 Convective heat transfer coefficients of water and airWhen seeking the convective heat transfer coefficient of water or air, its crucial to specify the geometry of the surface and flow. Reference articles and books provide values tailored to specific conditions. If your scenario matches a specific case in those references, you can find formulas and tables for it. Remember, the convective heat transfer coefficient is a function of various variables, so ensure the validity of the reference used, especially if it provides a single numerical value rather than a function. Convective heat transfer coefficient in CFD simulationsIn Computational Fluid Dynamics (CFD) simulations, convective heat transfer is directly calculated without the need to explicitly define the convective heat transfer coefficient. Through the solution of governing equations such as the Navier-Stokes equations and the energy equation, CFD software computes fluid flow characteristics and temperature distributions, allowing for the direct calculation of convective heat transfer rates at each point in the domain. This process inherently accounts for convective heat transfer, eliminating the requirement for specifying h separately. However, the accuracy of convective heat transfer predictions in CFD simulations depends on factors such as the numerical methods, turbulence modeling approaches, and boundary conditions used. Validation against experimental data remains crucial to ensure the reliability of CFD predictions. Types of convection heat transferConvection heat transfer can be classified into two primary types according to the driving forces and mechanisms involved:Forced convection: It occurs when fluid motion is initiated by external forces, such as fans, pumps, or atmospheric winds. Examples of forced convection include the use of fans to cool electronic devices like computers or refrigerators, the circulation of coolant in car engines driven by a water pump, and the movement of air in HVAC (heating, ventilation, and air conditioning) systems to regulate indoor temperature.Free (or natural) convection: Flow is induced by buoyancy forces, resulting from density variations caused by temperature differences within the fluid. Examples of this include the rising of warm air above a heated surface, such as in natural ventilation systems, the upward movement of hot water in a pot heated on a stove, and the circulation of air in a room due to a temperature gradient between indoor and outdoor environments. Convection heat transfer examplesConvection heat transfer is a common phenomenon encountered in various everyday situations as well as in industrial processes. Here are some examples:Cooking: Convection heat transfer plays a crucial role in numerous cooking techniques. For example, in an oven, hot air circulates around food, transferring heat and cooking it evenly.Cooling of Electronic Devices: Convection serves a vital role in cooling electronic devices like computers and servers. Fans or heatsinks aid in convection by directing airflow over heated components, enabling the dissipation of heat.Air Conditioning Systems: Air conditioners utilize convection in various ways to regulate indoor temperature and humidity.Atmospheric Circulation: Convection drives large-scale atmospheric circulation patterns, such as the Hadley, Ferrel, and Polar cells, which influence weather systems and climate around the globe.Boiling Water: When water is heated in a pot, convection currents form as hotter water rises to the surface, cools, and then sinks back down. This circulation helps distribute heat throughout the liquid. How to improve convection heat transfer?Here are some strategies to improve convection heat transfer:Increase Fluid Velocity: Increasing fluid velocities enhances convective heat transfer rates by encouraging more fluid mixing and minimizing thermal boundary layer thickness. Achieving this can involve employing fans, pumps, or other mechanical devices to augment fluid flow.Use Turbulent Flow: Turbulent flow boosts convective heat transfer coefficients relative to laminar flow by improving fluid mixing and disrupting thermal boundary layers. Promoting turbulent flow can be accomplished by increasing flow velocities or introducing obstacles.Enhance Surface Roughness: Surface roughness can disrupt laminar boundary layers and promote turbulence, thereby improving convective heat transfer. Textured or rough surfaces are utilized to increase heat transfer coefficients.Optimize Surface Geometry: Surface geometry is pivotal in convective heat transfer. Shapes and configurations that encourage fluid mixing while minimizing flow separation can enhance heat transfer coefficients. Incorporating surface features such as dimples, grooves, or other structures can significantly improve heat transfer performance.Any approach that amplifies turbulence in the flow, enlarges the contact surface, and prevents the formation of separation zones on the surface is effective. ConclusionIn conclusion, convection heat transfer is a fundamental process essential for various engineering and scientific endeavors, impacting energy efficiency, thermal management, and system performance. This article has comprehensively covered its underlying principles, calculation methods, measurement techniques, and enhancement strategies. By highlighting the role of Computational Fluid Dynamics (CFD) in predicting convective heat transfer rates and discussing the different types of convection mechanisms, including forced and natural convection, it provides valuable insights for optimizing convective heat transfer processes. Through strategies such as increasing fluid velocity, promoting turbulent flow, and optimizing surface geometry, engineers and researchers can effectively enhance convective heat transfer rates, leading to improved performance and efficiency in a wide range of applications.Click to access the heat transfer CFD Related Resources: heat transfer Heat Transfer Engineering Thermodynamics Convective Heat Transfer Coefficients Table Chart The following table charts of typical convective convection heat transfer coefficients for fluids and specific applications Typical values of heat transfer coefficient Flow type (W/m2 K) Forced convection; low speed flow of air over a surface 10 Forced convection; moderate speed flow of air over a surface 100 Forced convection; moderate speed cross- flow of air over a cylinder 200 Forced convection; moderate flow of water in a pipe 3000 Forced Convection; molten metals 2000 to 45000 Forced convection; boiling water in a pipe 50,000 Forced Convection - water and liquids 50 to 10000 Free Convection - gases and dry vapors 5 to 37 Free Convection - water and liquids 50 to 3000 Air 10 to 100 Free convection; vertical plate in air with 30C temperature difference 5 Boiling Water 3.000 to 100.000 Water fowing in tubes 500 to 1200 Condensing Water Vapor 5.0 - 100.0 Water in free convection 100 to 1200 Oil in free convection 50 to 350 Gas flow on tubes and between tubes 10 to 350 Typical values of heaters heat transfer coefficients Heaters (no phase change) heat transfer coefficients Hot Fluid Cold Fluid (BTU/hr-ft2-F) Steam Air 10 20 Steam Water 250 750 Steam Methanol 200 700 Steam Ammonia 200 700 Steam Aqueous solutions 100 700 Steam Light hydrocarbons (viscosity < 0.5 cP) 100 200 Steam Medium hydrocarbons (0.5 cP < viscosity < 1 cP) 50 100 Steam Heavy hydrocarbons (viscosity > 1) 6 60 Steam Gases 5 50 Dowtherm Gases 4 40 Dowtherm Heavy oils 8 60 Flue gas Aromatic hydrocarbon and steam 5 10 Typical values of Evaporator heat transfer coefficients Evaporators heat transfer coefficients Hot Fluid Cold Fluid Overall U (BTU/hr-ft2 -F) Steam Water 350 750 Steam Organic solvents 100 200 Steam Light oils 80 180 Steam Heavy oils (vacuum) 25 75 Water Refrigerant 75 150 Organic solvents Refrigerant 30 100 Typical values of Coolers heat transfer coefficients Coolers (no phase change) heat transfer coefficients Cold Fluid Hot Fluid Overall U (BTU/hr-ft2 -F) Water Water 150 300 Water Organic solvent 50 150 Water Gases 3 50 Water Light oils 60 160 Water Heavy oils 10 50 Light oil Organic solvent 20 70 Brine Water 100 200 Brine Organic solvent 30 90 Brine Gases 3 50 Organic solvents Organic solvents 20 60 Heavy oils Heavy oils 8 50 Typical values of Condensers heat transfer coefficients Condensers heat transfer coefficients Cold Fluid Hot Fluid Overall U (BTU/hr-ft2 -F) Water Steam (pressure) 350 -750 Water Steam (vacuum) 300 600 Water or brine Organic solvent (saturated, atmospheric) 100 200 Water or brine Organic solvent (atmospheric, high non-condensables) 20 80 Water or brine Organic solvent (saturated, vacuum) 50 120 Water or brine Organic solvent (vacuum, high non-condensables) 10 50 Water or brine Aromatic vapours (atmospheric with non-condensables) 5 30 Water Low boiling hydrocarbon (atmospheric) 80 200 Water High boiling hydrocarbon (vacuum) 10 30 Related: Heat energy transferred between a surface and a moving fluid with different temperatures - is known as convection.In reality this is a combination of diffusion and bulk motion of molecules. Near the surface the fluid velocity is low, and diffusion dominates. At distance from the surface, bulk motion increases the influence and dominates.Convective heat transfer can beforced or assisted convectionnatural or free convection Forced or Assisted ConvectionForced convection occurs when a fluid flow is induced by an external force, such as a pump, fan or a mixer.Natural or Free ConvectionNatural convection is caused by buoyancy forces due to dens ity differences caused by temperature variations in the fluid. At heating the density change in the boundary layer will cause the fluid to rise and be replaced by cooler fluid that also will heat and rise. This continues phenomena is called free or natural convection. Boiling or condensing processes are also referred to as a convective heat transfer processes.The heat transfer per unit surface through convection was first described by Newton and the relation is known as the Newton's Law of Cooling.The equation for convection can be expressed as:q = hc A dT (1)whereq = heat transferred per unit time (W, Btu/hr)A = heat transfer area of the surface (m2, ft2)hc = convective heat transfer coefficient of the process (W/(m2oC, Btu/(ft2 h oF))dT = temperature difference between the surface and the bulk fluid (oC, F) Heat Transfer Coefficients - Units1 W/(m2K) = 0.85984 kcal/(h m2 oC) = 0.1761 Btu/(ft2 h oF)1 Btu/(ft2 h oF) = 5.678 W/(m2 K) = 4.882 kcal/(h m2 oC)1 kcal/(h m2 oC) = 1.163 W/(m2K) = 0.205 Btu/(ft2 h oF)Overall Heat Transfer CoefficientsConvective Heat Transfer CoefficientsConvective heat transfer coefficients - hc - depends on type of media, if its gas or liquid, and flow properties such as velocity, viscosity and other flow and temperature dependent properties.Typical convective heat transfer coefficients for some common fluid flow applications:Free Convection - air, gases and dry vapors : 0.5 - 1000 (W/(m2K))Free Convection - water and liquids: 50 - 3000 (W/(m2K))Forced Convection - air, gases and dry vapors: 10 - 1000 (W/(m2K))Forced Convection - water and liquids:50 - 10000 (W/(m2K))Forced Convection - liquid metals:5000 - 40000 (W/(m2K))Boiling Water : 3.000 - 100.000 (W/(m2K))Condensing Water Vapor: 5.000 - 100.000 (W/(m2K))Heat Exchanger Heat Transfer CoefficientsConvective Heat Transfer Coefficient for AirThe convective heat transfer coefficient for air flow can be approximated tohc = 10.45 - v + 10 v1/2 (2)where hc = heat transfer coefficient (kCal/m2hC) v = relative speed between object surface and air (m/s)Since1 kcal/m2hC = 1.16 W/m2Cor 1 W/m2C = 0.862 kcal/m2hC(2) can be modified tohcW = 1.16 (10.45 - v + 10 v1/2) (2b)where hcW =heat transfer coefficient (W/m2C) Note! - this is an empirical equation and can be used for velocities 2 to 20 m/s.Convective Air Flow from a single Heat SourceExample - Convective Heat TransferA fluid flows over a plane surface 1 m by 1 m. The surface temperature is 50oC, the fluid temperature is 20oC and the convective heat transfer coefficient is 2000 W/m2oC. The convective heat transfer between the hotter surface and the colder air can be calculated asq = (2000 W/m2oC) ((1 m) (1 m)) ((50 oC) - (20 oC)) = 60000 (W) = 60 (kW) Convective Heat Transfer Calculator Convective Heat Transfer ChartConvective Heat Transfer Chart (pdf) Arithmetic Mean Temperature Difference in Heat Exchangers - AMTD - and Logarithmic Mean Temperature Difference - LMTD - formulas with examples - Online Mean Temperature Calculator. Heat transfer when steam condensates. Conductive heat transfer takes place in a solid if there is a temperature gradient. Calculate the vertical air flow and air velocity generated by a single heat source. Convective air flows from typical heat sources like people, computers, radiators and more. Hot or cold vertical surfaces generates vertical air flows - calculate air velocity and volume flow. Heat-transfer in heat exchangers are reduced by fouling. Overall heat transfer coefficients in common heat exchanger designs - tubular, plate or spiral. Average overall heat transmission coefficients for fluid and surface combinations like Water to Air, Water to Water, Air to Air, Steam to Water and more. Walls or heat exchangers - calculate overall heat transfer coefficients. Effective Temperature of air in movement.Convection, in general, can be defined as the process by which heat is transferred by movement of a heated fluid such as air or water. Natural Convection (a specific variety of convection) can be defined as the movement of a fluid caused by the tendency of hotter and therefore less dense material to rise, and colder, denser material to sink under the influence of gravity, which consequently results in transfer of heat.The purpose of this document is to demonstrate how to calculate heat transfer coefficients for these surfaces surrounded by either air or by water held at constant ambient temperatures and to discuss the theory associated with a few simple types of natural convection (flow over flat vertical or horizontal plates).This Excel spreadsheet calculates the convective heat transfer coefficients based on air and water properties at the defined ambient temperatures while using the formulae described below in the Calculation Details. The advantage of this spreadsheet is that it simplifies two complex tasks:Providing the properties of air and water at the ambient temperature of interest, andPerforming the calculation of natural convective heat transfer coefficients under several typical scenariosHow to Operate Excel Natural Convection WorksheetDownload the Excel spreadsheet to your computer and open.Three tabs are visible: "Natural Convection", "AirProperties" and "WaterProperties". All calculations are performed within the "Natural Convection" worksheet. Input values are entered into the green "Surface Temperature", "Ambient Air/Fluid Temperature" and "Length of Surface" fields. Results are produced in the orange fields.The air and water properties are calculated based on the "Ambient Air/Fluid Temperature" from a collection of curve fitted formulae based on those tabular data embodied within the "AirProperties" and "WaterProperties" worksheets.The "Surface Temperature" may not be known at first, but we make a guess to calculate our first convective heat transfer coefficient. We apply this convective heat transfer coefficient to our model and analyze to solve for temperatures and derive the "average" temperature of that same surface. We enter this "new" average temperature in our spreadsheet to calculate a "new" convective heat transfer coefficient and apply that "new" value to our model and analyze and solve for temperatures. We repeat this process until there is little change between the solved average temperature and the temperature we used to estimate our convective heat transfer coefficient. This temperature often converges after 1 to 3 iterations.Calculations are performed for both "air" (column B) and for "water" (column C), which are presented in columns labeled as such. Obviously the "air" convective heat transfer coefficients are much smaller than those for "water".The "Length of Surface" is also known as the "Length of flow direction along surface". For a rectangular flat horizontal plate, this will be the smaller of the two dimensions. For example, a plate that is 300mm by 600mm, we would use 300mm as our length, since this is the path of least resistance for our fluid flow. However, if our flat plate is inclined, or vertical, then this distance represents the plate length in the direction of inclination. To be clear, if our 300mm by 600mm plate was inclined such that the long direction was lifted at one end, then our length would be 600mm entered into our calculations. For vertical plates, this length is our plate height.Calculation DetailsThe following formulae are sourced from "Introduction to Heat Transfer, Third Edition", (Incropera and DeWitt), "Empirical Correlations: External Free Convection Flows, Chapter 9.6".The basic premise for these calculations is that there is a temperature difference between the object we would like to analyze and the surrounding ambient fluid (gas or liquid). This temperature difference changes the local density of the fluid near our object, and that fluid begins to move, trading places with fluid molecules having a different density. The rate of this movement depends on the fluid properties and the temperature difference. Ultimately, due to this movement, the transfer of heat moves from the hotter heat source to the colder heat sink. The rate of this convective action is directly related to the heat transfer capacity of the surrounding fluid. For example, if the surrounding fluid is water, then more heat transfer will occur compared to the same temperature condition but surrounded by air.These formulae are based on a collection of dimensionless numbers, such as the Nusselt Number, Rayleigh Number and Prandtl Number, as well as various properties of our surrounding fluid. The only property used from our object of heat transfer, is the average temperature across the surface for which we will apply our newly calculated convective heat transfer coefficient. Heat Transfer Engineering | Thermodynamics Example: A flat wall is exposed to the environment. The wall is covered with a layer of insulation 1 in. thick whose thermal conductivity is 0.8 Btu/hr-ft-F. The temperature of the wall on the inside of the insulation is 600F. The wall loses heat to the environment by convection on the surface of the insulation. The average value of the convection heat transfer coefficient on the insulation surface is 950 Btu/hr-ft2 -F. Compute the bulk temperature of the environment (Tb) if the outer surface of the insulation does not exceed 105F. Share copy and redistribute the material in any medium or format for any purpose, even commercially. Adapt remix, transform, and build upon the material for any purpose, even commercially. The licensor cannot revoke these freedoms as long as you follow the license terms. Attribution You must give appropriate credit , provide a link to the license, and indicate if changes were made . You may do so in any reasonable manner, but not in any way that suggests the licensor endorses you or your use. ShareAlike If you remix, transform, or build upon the material, you must distribute your contributions under the same license as the original. No additional restrictions You may not apply legal terms or technological measures that legally restrict others from doing anything the license permits. You do not have to comply with the license for elements of the material in the public domain or where your use is permitted by an applicable exception or limitation . No warranties are given. The license may not give you all of the permissions necessary for your intended use. For example, other rights such as publicity, privacy, or moral rights may limit how you use the material. Thermodynamics Directory Heat Transfer Directory Overall Heat Transfer Coefficient Table Chart: The heat transfer coefficient is the proportionality coefficient between the heat flux and the thermodynamic driving force for the flow of heat (i.e., the temperature difference, T): h = q / (Ts - K) where: q: amount of heat required (Heat Flux), W/m2 i.e., thermal power per unit area, q = d\dot{Q}/dA h: heat transfer coefficient, W/(m2 K) Ts = Solid Surface temperature K = Surrounding fluid area Temperature It is used in calculating the heat transfer, typically by convection or phase transition between a fluid and a solid. The heat transfer coefficient has SI units in watts per squared meter kelvin: W/(m2K). Heat transfer coefficient is the inverse of thermal insulance. This is used for building materials (R-value) and for clothing insulation. Related Resources: Overall Heat Transfer Coefficient Table Chart Pipes and Tubes Types Application Overall Heat Transfer Coefficient - U - W/(m2 K) Btu/(ft2 oF h) Tubular, heating or cooling Gas at atmospheric pressure inside and outside tubes 5 - 35 1 - 6 Gas at high pressure inside and outside tubes 150 - 500 25 - 90 Liquid outside (inside) and gas at atmospheric pressure inside (outside) tubes 15 - 70 3 - 15 Gas at high pressure inside and liquid outside tubes 200 - 400 35 - 70 Liquids inside and outside tubes 150 - 1200 25 - 200 Steam outside and liquid inside tubes 300 - 1200 50 - 200 Tubular, condensation Steam outside and cooling water inside tubes 1500 - 4000 250 - 700 Organic vapors or ammonia outside and cooling water inside tubes 300 - 1200 50 - 200 Tubular, evaporation steam outside and high-viscous liquid inside tubes, natural circulation 300 - 900 50 - 150 steam outside and low-viscous liquid inside tubes, natural circulation 600 - 1700 100 - 300 steam outside and liquid inside tubes, forced circulation 900 - 3000 150 - 500 Air-cooled heat exchangers Cooling of water 600 - 750 100 - 130 Cooling of liquid light hydrocarbons 400 - 550 70 - 95 Cooling of tar 30 - 60 5 - 10 Cooling of air or flue gas 60 - 180 10 - 30 Cooling of hydrocarbon gas 200 - 450 35 - 80 Condensation of low pressure steam 700 - 850 125 - 150 Condensation of organic vapors 350 - 500 65 - 90 Plate heat exchanger liquid to liquid 1000 - 4000 150 - 700 Spiral heat exchanger liquid to liquid 700 - 2500 125 - 500 condensing vapor to liquid 900 - 3500 150 - 700 Overall Heat Transfer Coefficient Table Chart Heat Exchangers Heaters (no phase change) Hot Fluid Cold Fluid Overall U (BTU/hr-ft2 -F) Steam Air 10 20 Steam Water 250 750 Steam Methanol 200 700 Steam Ammonia 200 700 Steam Aqueous solutions 100 700 Steam Light hydrocarbons (viscosity < 0.5 cP) 100 200 Steam Medium hydrocarbons (0.5 cP < viscosity < 1 cP) 50 100 Steam Heavy hydrocarbons (viscosity > 1) 6 60 Steam Gases 5 50 Dowtherm Gases 4 40 Dowtherm Heavy oils 8 60 Flue gas Aromatic hydrocarbon and steam 5 10 Overall Heat Transfer Coefficient Table Chart Industrial Evaporators Evaporators Hot Fluid Cold Fluid Overall U (BTU/hr-ft2 -F) Steam Water 350 750 Steam Organic solvents 100 200 Steam Light oils 80 180 Steam Heavy oils (vacuum) 25 75 Water Refrigerant 75 150 Organic solvents Refrigerant 30 100 Overall Heat Transfer Coefficient Table Chart Industrial Coolers Coolers (no phase change) Cold Fluid Hot Fluid Overall U (BTU/hr-ft2 -F) Water Water 150 300 Water Organic solvent 50 150 Water Gases 3 50 Water Light oils 60 160 Water Heavy oils 10 50 Light oil Organic solvent 20 70 Brine Water 100 200 Brine Organic solvent 30 90 Brine Gases 3 50 Organic solvents Organic solvents 20 60 Heavy oils Heavy oils 8 50 Overall Heat Transfer Coefficient Table Chart Industrial Condensers Condensers Cold Fluid Hot Fluid Overall U (BTU/hr-ft2 -F) Water Steam (pressure) 350 -750 Water Steam (vacuum) 300 600 Water or brine Organic solvent (saturated, atmospheric) 100 200 Water or brine Organic solvent (atmospheric, high non-condensables) 20 80 Water or brine Organic solvent (saturated, vacuum) 50 120 Water or brine Organic solvent (vacuum, high non-condensables) 10 50 Water or brine Aromatic vapours (atmospheric with non-condensables) 5 30 Water Low boiling hydrocarbon (atmospheric) 80 200 Water High boiling hydrocarbon (vacuum) 10 30 Overall Heat Transfer Coefficient Table Chart Various Fluids no phase change Fluid Film Coefficient (BTU/hr-ft2 -F) Water 300 2000 Gases 3 50 Organic Solvents 60 500 Oils 10 120 Overall Heat Transfer Coefficient Table Chart Condensing Fluids Condensing Fluid Film Coefficient (BTU/hr-ft2 -F) Steam 1000 3000 Organic Solvents 150 500 Light Oils 200 400 Heavy Oils (vacuum) 20 50 Ammonia 500 1000 Overall Heat Transfer Coefficient Table Chart Various Fluids (Liquids and Gasses) Heat transfer by the motion and mixing of the molecules of a liquid or gas is called convection. Convection Convection involves the transfer of heat by the motion and mixing of "macroscopic" portions of a fluid (that is, the flow of a fluid past a solid boundary). The term natural convection is used if this motion and mixing is caused by density variations resulting from temperature differences within the fluid. The term forced convection is used if this motion and mixing is caused by an outside force, such as a pump. The transfer of heat from a hot water radiator to a room is an example of heat transfer by natural convection. The transfer of heat from the surface of a heat exchanger to the bulk of a fluid being pumped through the heat exchanger is an example of forced convection. Heat transfer by convection is more difficult to analyze than heat transfer by conduction because no single property of the heat transfer medium, such as thermal conductivity, can be defined to describe the mechanism. Heat transfer by convection varies from situation to situation (upon the fluid flow conditions), and it is frequently coupled with the mode of fluid flow. In practice, analysis of heat transfer by convection is treated empirically (by direct observation). Convection heat transfer is treated empirically because of the factors that affect the stagnant film thickness: Convection involves the transfer of heat between a surface at a given temperature (Ts) and fluid at a bulk temperature (Tb). The exact definition of the bulk temperature (Tb) varies depending on the details of the situation. For flow adjacent to a hot or cold surface, Tb is the temperature of the fluid "far" from the surface. For boiling or condensation, Tb is the saturation temperature of the fluid. For flow in a pipe, Tb is the average temperature measured at a particular crosssection of the pipe. The basic relationship for heat transfer by convection has the same form as that for heat transfer by conduction: $$ \dot{Q} = h ~A ~\Delta T $$ where: \(\dot{Q} \) = rate of heat transfer (Btu/hr) h = convective heat transfer coefficient (Btu/hr-ft2-F) A = surface area for heat transfer (ft2) T = temperature difference (F) The convective heat transfer coefficient (h) is dependent upon the physical properties of the fluid and the physical situation. Typically, the convective heat transfer coefficient for laminar flow is relatively low compared to the convective heat transfer coefficient for turbulent flow. This is due to turbulent flow having a thinner stagnant fluid film layer on the heat transfer surface. Values of h have been measured and tabulated for the commonly encountered fluids and flow situations occurring during heat transfer by convection. Example: A 22 foot uninsulated steam line crosses a room. The outer diameter of the steam line is 18 in. and the outer surface temperature is 280F. The convective heat transfer coefficient for the air is 18 Btu/hr-ft2-F. Calculate the heat transfer rate from the pipe into the room if the room temperature is 72F. Solution: $$ \begin{eqnarray} \dot{Q} &=& h ~A ~\Delta T onumber \\ &=& h ~(2 \pi ~r ~L) ~\Delta T onumber \\ &=& \left(18 ~{\text{Btu} \over \text{hr-ft-}^{\circ}\text{F}} \right) \left[2 (3.14) (0.75 ~\text{ft}) (22 ~\text{ft}) \right] (280^{\circ}\text{F} - 72^{\circ}\text{F}) onumber \\ &=& 3.88 \times 10^5 ~{ \text{Btu} \over \text{hr} } \end{eqnarray} $$ Many applications involving convective heat transfer take place within pipes, tubes, or some similar cylindrical device. In such circumstances, the surface area of heat transfer normally given in the convection equation (\(\dot{Q} = h ~A ~\Delta T \)) varies as heat passes through the cylinder. In addition, the temperature difference existing between the inside and the outside of the pipe, as well as the temperature differences along the pipe, necessitates the use of some average temperature value in order to analyze the problem. This average temperature difference is called the log mean temperature difference (LMTD), described earlier. It is the temperature difference at one end of the heat exchanger minus the temperature difference at the other end of the heat exchanger, divided by the natural logarithm of the ratio of these two temperature differences. The above definition for LMTD involves two important assumptions: (1) the fluid specific heats do not vary significantly with temperature, and (2) the convection heat transfer coefficients are relatively constant throughout the heat exchanger. Overall Heat Transfer Coefficient Many of the heat transfer processes encountered in nuclear facilities involve a combination of both conduction and convection. For example, heat transfer in a steam generator involves convection from the bulk of the reactor coolant to the steam generator inner tube surface, conduction through the tube wall, and convection from the outer tube surface to the secondary side fluid. In cases of combined heat transfer for a heat exchanger, there are two values for h. There is the convective heat transfer coefficient (h) for the fluid film inside the tubes and a convective heat transfer coefficient for the fluid film outside the tubes. The thermal conductivity (k) and thickness (x) of the tube wall must also be accounted for. An additional term (Uo), called the overall heat transfer coefficient, must be used instead. It is common practice to relate the total rate of heat transfer (\(\dot{Q} \)) to the cross-sectional area for heat transfer (Ao) and the overall heat transfer coefficient (Uo). The relationship of the overall heat transfer coefficient to the individual conduction and convection terms is shown in Figure 6. Figure 6: Overall Heat Transfer Coefficient Recalling Equation 2-3: $$ \dot{Q} = U_o ~A_o ~\Delta T_o $$ where Uo is defined in Figure 6. An example of this concept applied to cylindrical geometry is illustrated by Figure 7, which shows a typical combined heat transfer situation. Figure 7: Combined Heat Transfer Using the figure representing flow in a pipe, heat transfer by convection occurs between temperatures T1 and T2; heat transfer by conduction occurs between temperatures T2 and T3; and heat transfer occurs by convection between temperatures T3 and T4. Thus, there are three processes involved. Each has an associated heat transfer coefficient, cross-sectional area for heat transfer, and temperature difference. The basic relationships for these three processes can be expressed using Equations 2-5 and 2-9. $$ \dot{Q} = h_1 ~A_1 ~(T_1 - T_2) $$ $$ \dot{Q} = {k \over \Delta r} ~A_{lm} ~(T_2 - T_3) $$ $$ \dot{Q} = h_2 ~A_2 ~(T_3 - T_4) $$ To can be expressed as the sum of the T of the three individual processes. To = (T1 T2) + (T2 T3) + (T3 T4) If the basic relationship for each process is solved for its associated temperature difference and substituted into the expression for To above, the following relationship results. $$ \Delta T_o = \dot{Q} \left({1 \over h_1 A_1} + { \Delta r \over k ~A_{lm} } + {1 \over h_2 A_2} \right) $$ This relationship can be modified by selecting a reference cross-sectional area Ao. $$ \Delta T_o = { \dot{Q} \over A_o } \left({A_o \over h_1 A_1} + { \Delta r ~A_o \over k ~A_{lm} } + {A_o \over h_2 A_2} \right) $$ Solving for \(\dot{Q} \) results in an equation in the form \(\dot{Q} = U_o ~A_o ~\Delta T_o \). $$ \dot{Q} = { 1 \over \left({A_o \over h_1 A_1} + { \Delta r ~A_o \over k ~A_{lm} } + {A_o \over h_2 A_2} \right) } ~A_o ~\Delta T_o $$ where: $$ U_o = { 1 \over \left({A_o \over h_1 A_1} + { \Delta r ~A_o \over k ~A_{lm} } + {A_o \over h_2 A_2} \right) } $$ Equation 2-10 for the overall heat transfer coefficient in cylindrical geometry is relatively difficult to work with. The equation can be simplified without losing much accuracy if the tube that is being analyzed is thin-walled, that is the tube wall thickness is small compared to the tube diameter. For a thin-walled tube, the inner surface area (A1), outer surface area (A2), and log mean surface area (Alm), are all very close to being equal. Assuming that A1, A2, and Alm are equal to each other and also equal to Ao allows us to cancel out all the area terms in the denominator of Equation 2-11. This results in a much simpler expression that is similar to the one developed for a flat plate heat exchanger in Figure 6. $$ U_o = { 1 \over {1 \over h_1} + { \Delta r \over k } + {1 \over h_2} } $$ The convection heat transfer process is strongly dependent upon the properties of the fluid being considered. Correspondingly, the convective heat transfer coefficient (h), the overall coefficient (Uo), and the other fluid properties may vary substantially for the fluid if it experiences a large temperature change during its path through the convective heat transfer device. This is especially true if the fluid's properties are strongly temperature dependent. Under such circumstances, the temperature at which the properties are "looked-up" must be some type of average value, rather than using either the inlet or outlet temperature value. For internal flow, the bulk or average value of temperature is obtained analytically through the use of conservation of energy. For external flow, an average film temperature is normally calculated, which is an average of the free stream temperature and the solid surface temperature. In any case, an average value of temperature is used to obtain the fluid properties to be used in the heat transfer problem. The following example shows the use of such principles by solving a convective heat transfer problem in which the bulk temperature is calculated. Example: A flat wall is exposed to the environment. The wall is covered with a layer of insulation 1 in. thick whose thermal conductivity is 0.8 Btu/hr-ft-F. The temperature of the wall on the inside of the insulation is 600F. The wall loses heat to the environment by convection on the surface of the insulation. The average value of the convection heat transfer coefficient on the insulation surface is 950 Btu/hr-ft2-F. Compute the bulk temperature of the environment (Tb) if the outer surface of the insulation does not exceed 105F. Solution: Find heat flux (\(\dot{Q}'' \)) through the insulation. $$ \dot{Q} = k ~A \left({ \Delta T \over \Delta x }\right) $$ $$ \begin{eqnarray} { \dot{Q} \over A } &=& 0.8 ~{\text{Btu} \over \text{hr-ft-}^{\circ}\text{F}} \left({ 600^{\circ}\text{F} - 105^{\circ}\text{F} \over 1 ~\text{in} ~{ 1 ~\text{ft} \over 12 ~\text{in} } }\right) onumber \\ &=& 4752 ~{ \text{Btu} \over \text{hr-ft}^2 } \end{eqnarray} $$ Find the bulk temperature of the environment. $$ \begin{eqnarray} \dot{Q} &=& h ~A ~(T_{ins} - T_b) onumber \\ (T_{ins} - T_b) &=& { \dot{Q} \over h ~A } onumber \\ T_b &=& T_{ins} - { \dot{Q}'' \over h } onumber \\ T_b &=& 105^{\circ}\text{F} - { 4752 ~{ \text{Btu} \over \text{hr-ft}^2 } \over 950 ~{\text{Btu} \over \text{hr-ft-}^{\circ}\text{F}} } onumber \\ T_b &=& 100^{\circ}\text{F} \end{eqnarray} $$ The Engineering ToolBox provides a wide range of free tools, calculators, and information resources aimed at engineers and designers. It offers detailed technical data and calculations for various fields such as fluid mechanics, material properties, HVAC systems, electrical engineering, and more.The site includes resources for common engineering tasks, such as calculating physical properties (e.g., density, viscosity, thermal conductivity), converting units, and designing systems like heating and water distribution. With sections on everything from acoustics to hydraulics, it serves as a comprehensive tool for both students and professionals in technical and engineering disciplines.This article is concerned with the transfer of thermal energy by the movement of fluid and, as a consequence, such transfer is dependent on the nature of the flow. Heat transfer by convection may occur in a moving fluid from one region to another or to a solid surface, which can be in the form of a duct, in which the fluid flows or over which the fluid flows. Convective heat transfer may take place in boundary layers, that is, to or from the flow over a surface in the form of a boundary layer, and within ducts where the flow may be boundary-layer-like or fully-developed. It may also occur in flows which are more complicated, such as those which are separated, for example, in the aft region of a cylinder in cross-flow or in the vicinity of a backward-facing step. The flow may give rise to convective heat transfer where it is driven by a pump and is referred to as forced convection, or arise as a consequence of temperature gradients and buoyancy, referred to as natural or free convection. Examples are given later in this Section and are shown in Figure1 to facilitate introduction to terminology and concepts.Figure1.Velocity and temperature profiles in boundary-layer and separated flows. The boundary layer on the flat surface of Figure1 has the usual variation of velocity from zero on the surface to a maximum in the free-stream. In this case, the surface is assumed to be at a higher temperature than the free-stream and the finite gradient at the wall confirms the heat transfer from the surface to the flow. It is also possible to have zero temperature gradient at the wall so there is no heat transfer to or from the surface but heat transfer within the flow. If the flow is laminar, heat transfer from the surface is given by the Fourier flux law, that is:where q represents the rate of heat transfer per unit surface area, is the thermal conductivity, T is the temperature and y is distance measured from the surface. The same expression applies to any region of the flow and also in the case of the adiabatic wall where zero temperature gradient implies zero heat transfer. It should be noted that the surface can be horizontal as shown, with air flow driven by a fan or a liquid flow by a pump, and that it can equally be vertical, with buoyancy providing the driving force for the flow. In the latter case, the free-stream velocity would be zero so that the corresponding profile would have zero values at the wall and far from the wall.The backward-facing step of Figure1 results in a more complicated flow and several boundary layers can be identified within the flow as a consequence of separation and reattachment. The details of flows of this type are not well-understood so it is difficult to identify the characteristics of the boundary layers and it can be imagined that the shapes of the velocity and temperature profiles and therefore of the local heat transfer within the fluid and to the wallwill vary considerably from one location to another. It is known, for example, that the rate of heat transfer can become high at the location of reattachment of the upstream flow on to the surface of the step, as is also the case at the leading edge of a cylinder in cross-flow, but the detailed mechanisms remain incompletely understood and research continues.It is well known that even comparatively simple geometrical configurations, such as those of Figure1, can give rise to heat transfer rates which vary considerably depending on the nature of the flow and of the surface. With laminar flows, heat transfer to or from the wall varies with distance from the leading edge of a boundary layer. Turbulent flows can give rise to heat transfer rates which are much larger than those of laminar flows, and are caused by the manner in which the turbulent fluctuations increase mixing; they also affect the heat transfer to and from the surface, especially where the free-stream fluid is able to penetrate to the wall even for short periods of time. The nature of the surface, for example the degree or type of roughness, usually affects heat transfer to or from it, and in some circumstances to a large extent. It is convenient, therefore, to represent the heat transfer at the wall by the expressionwhereagain represents the rate of heat transfer from the wall, this time over unit area of surface; the temperature difference refers to that between the wall and the free stream; and is the heat transfer coefficient which is a characteristic of the flow and of the surface. The two temperatures can vary with x-distance and it can be difficult to identify a free-stream temperature in some complex flows. Typical values of are shown in Table1, from which it can be seen that increases in velocity generally result in increases in heat transfer coefficient, so that is smallest in natural convection and increases to 100 and more on flat surfaces with air velocities greater than around 50 m/s. The heat transfer coefficient is considerably greater with liquid flows and greater again with two-phase flows.Table1.Typical values of heat transfer coefficient Flow type(W/m2K)Forced convection; low speed flow of air over a surface10Forced convection; moderate speed flow of air over a surface100Forced convection; moderate speed cross- flow of air over a cylinder200Forced convection; moderate flow of water in a pipe3000Forced convection; boiling water in a pipe50,000Free convection; vertical plate in air with 30C temperature difference5It should be noted that the above equations are expressed in terms of dimensional parameters. And it is easy to see that a combination of the two will lead to a nondimensional parameter x/, where is a wall heat transfer coefficient, x is a characteristic distance and is the conductivity of the fluid; this is known as the Nusselt number and can readily be devised from dimensional analysis as well as from nondimensional forms of conservation equations as suggested in the following section. The heat transfer coefficients of Table1 can be expressed in terms of this nondimensional number, the Nusselt number, and analytical and correlation equations are usually expressed in this way as will be shown below.It is also useful to note that the heat transfer coefficient and the Nusselt number can be used to refer to local values at a location x on a surface, or to an integrated value up to the location x.The concept of dimensional analysis gives rise to several nondimensional groups, to which reference will be made in this section, and it is convenient to introduce them here. In addition to the Nusselt number, reference will be made to the following:The Prandtl number is dependent only on fluid properties; the Reynolds number is a ratio of inertial to viscous forces and is relevant throughout the subject of fluid mechanics and convection; the Stanton number is a combination of Nu, Pr, and Re; and the Grashof number characterizes natural convection with the gravitational acceleration, g, and , the coefficient of volumetric thermal expansion, and is a combination of inertial, u2/y, frictional, vu/y2, and buoyancy, gT, scales. These nondimensional groups may be obtained from conservation equations and are convenient in the representation of results and correlations of experimental data.It is useful to examine the equations which represent conservation of mass, momentum and energy and these are written below for rectangular Cartesian coordinates with simplification of uniform properties.whereThe three equations representing conservation of momentum and the equation representing conservation of energy have the same form, with the terms on the left-hand sides representing convection of momentum and energy. It should be noted that these convective terms are nonlinear, thereby presenting difficulties for any solution and that there are four individual parts of convection corresponding to variations in time and in the three directions. The terms on the right-hand side are slightly simplified forms of those representing transport by diffusion together with pressure forces and sources or sinks of thermal energy. Terms for buoyancy may be added as shown in a following section. It is easy to see that nondimensional velocities and distances in the momentum equations will lead to the inverse of Reynolds number and of the temperatures, velocities and distances in the energy equation to a nondimensional group which comprises (1/PrRe). In later sections, these equations will be simplified to deal with convective heat transfer in steady, laminar flows of forced and free convection.It is evident from the above that there is some similarity between the equations for conservation of momentum and thermal energy so that the solutions of the two equations will have similar forms when the source terms are zero, the Prandtl number is unity and the solutions are presented in nondimensional form. The presence of buoyancy is often limited to the second momentum equation into which an additional term of the form g(Tw T) must be added. Where the surface which gives rise to the temperature differenceand therefore to the buoyant forceis not vertical, the angle of the surface to the direction of the gravitational force must be considered. This will lead to the resolution of forces so that part of the buoyancy term will appear in the first momentum equation with that in the second equation, multiplied by the sine of the angle to the vertical. This will give rise to an additional nondimensional group, the Grashof number.In the absence of convection terms, the energy equation reduces to that for heat conduction and the momentum equations are no longer relevant where the conduction takes place in a stationary material. Many other simplifications of the above equations are possible, including those for two-dimensional flows and for boundary-layer flows, as will be seen below. Also, it is possible to integrate the equations and, in their simpler forms, this can have some merit; for example, in the integral momentum and energy equations where the dependent variable is devised so as to be represented in terms of one independent variable, and therefore solvable by simple numerical methods. More complicated forms may also exist as discussed in the following section.Laminar and Turbulent FlowsMost flows in nature and in engineering equipment occur at moderately high Reynolds numbers, so they are described as turbulent. Thus, the properties of the flow at any point are time dependent with scales which vary from very small, the Kolmogorov scale, to that corresponding to the largest possible dimension of the flow. In a room, for example, the Kolmogorov scale may be of the order of a fraction of 1 mm or less than 1 ms time scale if the velocity is of the order of 1 m/s, and the largest, of the order of several meters or more than 103 larger. There are two important implications for this: the first, that the rate of heat transfer from a surface to a flow will be considerably greater than if the flow were laminar at the same Reynolds number; and secondly, that the conservation equations are even more difficult to solve than for the laminar flow since any numerical solution must now consider physical and time scales which encompass three orders of magnitude. The first means that turbulent convection is important, much more important than laminar convection; and the second, that the conservation equations cannot be solved in their general form except where the boundary conditions allow them to be reduced to simpler forms and even then, with additional problems. This conclusion has led to the widespread use of correlation formulas based on measurements and these, of necessity, encompass limited ranges of flow. Some examples are presented and discussed in the following section. It has also led to widespread attempts to solve complicated forms of the conservation equations with assumptions which represent the turbulent aspects of the flow. The following paragraphs provide an introduction to this approach.The introduction of Reynolds averaging, that is, to rewrite the time-dependent variables as sums of mean and fluctuating components, to introduce the new dependent variable into the conservation equations and to average overall time results in equations of the form:where the upper case symbols refer to time-averaged quantities; the lower case, to fluctuating quantities with q, the temperature fluctuations; is /cp; and the overbars, to averages of multiplications of two time-dependent quantities. The equations have been written in tensor notation to render them more compact, but the similarity between the conservation of the time-averaged momentum and energy equations is still evident. The terms representing convection are still on the left-hand side, with diffusion on the right-hand side. There are now two diffusion terms in each equation: one representing laminar diffusion; and the second, the correlations between fluctuating components. There are still five equations, but now there are more than five unknowns since the correlations imply six terms in the momentum equations and three in the energy equation. Thus, it is evident that these equations do not represent a soluble set without assumptions which reduce the number of unknowns to the number of equations. These require models for the Reynolds stresses,, and the turbulent heat fluxes,, and, as shown elsewhere, it is possible to derive equations for these correlation terms. Each gives rise to higher order correlations so that a decision must be made about closure as well as the introduction of model assumptions.By analogy with laminar flow, it is possible to write the turbulent momentum flux and turbulent heat flux in the formsorand nondimensional forms of these expressions with turbulent viscosity and turbulent conductivity will lead to Reynolds and Prandtl numbers, where the latter is frequently referred to as a turbulent Prandtl number.The turbulent Prandtl number has found considerable use in engineering calculations of convective heat transfer since it can be assigned a value of unity. With the laminar Prandtl number also near unity for airand often of secondary importance since laminar diffusion is less important than turbulent diffusionthe momentum and energy equations can be solved once for flows where there is no pressure gradient and no sources or sinks of energy, with similar results if presented in nondimensional variables. This approach applies to complex flows with difficult numerical solutions and to simple boundary-layer flows as will be shown.With assumptions of high Reynolds numbers and local equilibrium, so that the influence of one region of flow on another is small, it is possible to simplify the time-averaged conservation equations. Assuming two-dimensional boundary layers yields:andwhere C and Ct are constants, lm is the mixing length for the transfer of momentum and lt is a corresponding mixing length for the transfer of thermal energy. These equations reduce to the effective viscosity and Prandtl number equations referred to above when the length scales and constants are equal and the Prandtl number is unity. Thus, the concept of a turbulent Prandtl number is limited in its applicability, as is that of a turbulent viscosity. But the range of acceptance for engineering calculations remains large.As will be shown below, the exact solution of the equation appropriate to the laminar flow over a flat plate, where the free-stream and plate temperatures are constant and different, may be written as:which recognizes the importance of the Reynolds and Prandtl numbers and expresses the heat transfer coefficient in terms of the Nusselt number. The corresponding result for laminar natural convection over a vertical plate with similar boundary conditions is:In turbulent flows, approximations appropriate to a flat plate with forced convection have led to expressions of similar form; for example,As a consequence, equations used to represent measurements of complex flowswhere analytic and numerical solutions are either impossible or subject to large inaccuracytend to have this form. Several examples are provided in the following sections.Forced Convective Heat TransferForced convection is associated with flows which are driven by pumps and fans or by the movement of a body through stationary fluids, as in an aeroplane or ship where each has substantial means at its disposal to cause it to move. It is in contrast to natural convection where gravity provides the driving force, although it is possible to have mixed convection in a limited number of flows where the pressure and gravitational forces are of the same order of magnitude, that is Gr/Re2 is approximately unity. All exact analytical solutions are simplified forms of conservation equations and for laminar flows. Some other cases are discussed below.Boundary Layer Heat Transfer is discussed in the relevant article.Heat transfer between parallel platesThe flow between flat plates is portrayed in Figure2. It comprises boundary layers which begin at the leading edges, grow on each of the two surfaces until the potential core narrows to zero and then continues towards a fully-developed laminar flow, after which all gradients in the x-direction become zero.Figure2.Laminar flow between flat plates.The boundary layers are represented by the boundary layer equationswith boundary conditionsandcorresponds to a symmetry boundary condition.In the initial region where the boundary layers are separated by a region of potential flow, the analysis is similar to that for a boundary layer, with the free-stream condition represented by the potential core velocity and temperature. Further downstream, the flow becomes fully-developed so that the velocity and temperature profiles will not change if expressed in terms of appropriate dimensionless quantities. This will be demonstrated below. It is useful to note, however, that there is an intermediate region where there is no potential core and where the flow is not fully-developed. In this region, it is necessary to solve the equations representing conservation of mass and momentum so that each is satisfied; this may require an interactive approach.In the case of fully-developed laminar flow, the convective terms become zero sinceand the momentum equation becomeswith the pressure gradient constant so that integration with the boundary conditions at the wall and on the symmetry line leads to:and, if one plate moves parallel to the other with a constant velocity U, the solution becomesIn the former case, the temperature profile has the simple formThis, too, may be complicated by considering the effect of viscous heating, which requires the addition of a term of the form to the conservation equation for energy, andfor zero pressure gradient and constant values of Uleads toandThis last result must be regarded as an approximation since no account has been taken of variations likely to occur in transport and thermodynamic properties.The velocity profile for fully-developed laminar flow is a parabola when the walls are stationary, provided that the fluid properties are constant and the velocities are low; it is linear when the pressure gradient is absent and the wall moves with constant speed with respect to the other. The effect of a moving surface is to provide a force which can act against or with that exerted by pressure. This is reflected in the velocities which can be in positive and negative directions. The temperature profile is expressed in terms of surface temperatures and it is clear that the bulk temperature will increase if one or both of the walls is hotter than the initial temperature, T1 Thus, the temperature profile is often expressed in terms of the initial temperature and the bulk-mean temperature, defined as:where U is the bulk velocity as discussed further below.Flow and heat transfer in a pipe are of rather more importance than those between parallel plates since they are found more frequently in engineering practice. The flow may again begin at the leading edge so that laminar flow solutions can be obtained as for parallel plates, but this time to equations in cylindrical coordinates and without the prospect of one surface moving with respect to another. At small values of Reynolds number, ud/ , the length required to achieve fully-developed laminar flow may be given by the expressionand originates from asymptotic solutions of the boundary layer equations. The flow in small-diameter pipes required to achieve these small Reynolds numbers comes from larger diameter pipes or from plenum chambers, so it is likely that boundary layers do not have their origins at the beginning of the small diameter pipe. Rather, there is a sudden contraction for which the flow is properly represented by more complete forms of the conservation equations than their boundary-layer forms. Indeed, the flow may separate inside the pipe with a more rapid movement towards fully-devel-oped conditions than would be the case with attached boundary layers.The region of developing flow can be small in many cases and fully-developed flow is usually more important than the developing flow. The conservation equations in cylindrical coordinates may be reduced for fully-developed flow in the same way as between two plates, with the resultand this, with boundary conditionsandmay be integrated to yieldwhereThe Moody friction factor, defined as f = (dp/dx)/0.5U2/D, commonly represents the relationship between pressure drop, geometry and fluid properties, and may be deduced for fully-developed laminar pipe flow as:which is sometimes referred to as the Hagen-Poiseuille friction law.The energy equation in cylindrical coordinates has the formand this reduces for fully-developed flow towhere Tb is the bulk temperature, defined as:Integration of the differential equation with boundary conditions corresponding to symmetry at the centre line, and for the particular condition thatleads toand towhich is independent of the Reynolds and Prandtl numbers, provided the flow remains laminar. An iterative solution is required to solve the equations for the boundary conditionand leads to the resultwhich shows that the solution depends on the thermal boundary condition.Of course, the flow will remain laminar only if the Reynolds number is less than around 2 300 or to larger values if it is so free from disturbances that none are available to propagate and cause turbulent flow, as is usually the case. Where turbulent flow occurs because a disturbance has propagated and led to fluctuations in all regions of flow except in the viscous sublayer, the nature of the flow and of the problem has changed. It is possible to return to considering the consequences of the onset of transition and of the transitional region in the context of the boundary layer in the entrance region of the pipe. But the overall effect will be to induce turbulent flow rapidly so that the emphasis is again, and even more so, on the region of fully-developed flowwhich now corresponds to turbulent and not laminar flow. It is possible to retain a boundary-layer flow, possibly with transitional regions for some distance, but the common shape of slightly-rounded entrance geometry usually leads to a fully-developed turbulent flow in distances not much more than 50 diameters, and in shorter distances for engineering calculations.Correlation of measurements of pressure drop against bulk velocity and diameter has led Blasius to propose the expressionwhich, together with the laminar flow result ofallow Figure3 to be drawn and in which the laminar-flow result can be extended to Reynolds numbers well in excess of 105, provided care is taken with the nature of the initial conditions, with the smooth surface of the pipe and with the absence of disturbances of any kind. More usually, laminar flow does not exist at Reynolds numbers larger than 2,300, above which transition to the turbulent-flow curve takes place with a transitional region which can be short or long depending on the nature of the disturbances. The friction factor thus varies with the Reynolds number, based on the diameter of the pipe, and with laminar, transitional and turbulent regions as shown in Figure3.Figure3.Variation of friction factor with Reynolds number for flow in a pipe.The skin-friction coefficient (or Fanning friction factor) is related to the friction factor byso that the coefficient for turbulent flow may be expressed aswith the constants stemming from consideration of experimental results and are, therefore, of limited applicability. Figure3 shows the variation of friction factor with Reynolds number based on the pipe diameter, and the distinction between those for laminar and turbulent flow. At high Reynolds number, the results become less certain as indicated by the two lines, but the graph is adequate for many design purposes.Consideration of the similar nature of the equations representing conservation of momentum and energy implies that the variation of Nusselt number will also be dependent upon Reynolds number, together with the Prandtl number where it is different from unity. An example of an expression describing the variation of Nusselt number with turbulent flow in a pipe is:As with Figure3 and the friction factor and skin-friction coefficient, uncertainty increases at high Reynolds numbers and also in the transitional region where the difference between the results for laminar and turbulent flows are widely divergent. This may occur over a range of Reynolds numbers depending on the initial and boundary conditions. It should be noted that rough surfaces increase the values of skin-friction coefficient and Nusselt number. Related calculations can be made for noncircular ducts with an hydraulic diameter replacing the geometric diameter.

Convective heat transfer coefficient of still air. What is convective heat transfer coefficient. Heat transfer coefficient of water vs air. Convective heat transfer coefficient calculation. What is the heat transfer coefficient of water. Convection coefficient for air.