Design Calculation of Furnace for Drying of Orange Peel

Arpan Phukan[1] Ayush Gupta[2] Mrinmoy Saikia[3] Prof. Sangmeshwar S. Lohar[4]
[1,2,3,4]Department of Mechanical Engineering
[1,2,3,4]D.Y Patil College of Engineering, Ambi, Talegaon-410507, Pune, India

Abstract— Convective heat transfer, often referred to simply as convection, is the transfer of heat from one place to another by the movement of fluids. Convection is usually the dominant form of heat transfer in liquids and gases. Although often discussed as a distinct method of heat transfer, convective heat transfer involves the combined processes of unknown conduction (heat diffusion) and advection (heat transfer by bulk fluid flow). Convection can be "forced" by movement of a fluid by means other than buoyancy forces (for example, a water pump in an automobile engine). Thermal expansion of fluids may also force convection. In other cases, natural buoyancy forces alone are entirely responsible for fluid motion when the fluid is heated, and this process is called "natural convection". An example is the draft in a chimney or around any fire. In natural convection, an increase in temperature produces a reduction in density, which in turn causes fluid motion due to pressures and forces when fluids of different densities are affected by gravity (or any g-force). For example, when water is heated on a stove, hot water from the bottom of the pan rises, displacing the colder denser liquid, which falls. After heating has stopped, mixing and conduction from this natural convection eventually result in a nearly homogeneous density, and even temperature. Without the presence of gravity (or conditions that cause a g-force of any type), natural convection does not occur, and only forced-convection modes operate.

Keywords: Free or Natural Convection, Forced Convection, Reynolds Number, Prandtl Number, Nusselt Number

I. INTRODUCTION

Many efforts have been made on heat transfer enhancement according to the progress of thermal systems. The recent researches in heat transfer enhancement lead to the development of currently used heat transfer techniques. We can classify the flow of a fluid in a straight circular tube into either laminar or turbulent flow. It is assumed from here on that we assume fully developed incompressible, Newtonian, steady flow conditions. Fully developed flow implies that the tube is long compared with the entrance length in which the velocity distribution at the inlet adjusts itself to the geometry and no longer changes with distance along the tube. It is generally accepted that the heat transfer from the fluid tube of a fin absorber to the fluid itself, depends to a great extent on the flow conditions. Laminar flow, as compared to turbulent flow, can result in a considerable loss in efficiency which could often be avoided by the appropriate design of the absorber and the absorber arrangement in coordination with the actual use.

II. HEAT TRANSFER IN FLOW THROUGH CONDUITS

Convection is the mechanism of heat transfer through a fluid in the presence of bulk fluid motion. Convection is classified as natural (or free) and forced convection depending on how the fluid motion is initiated. In natural convection, any fluid motion is caused by natural means such as the buoyancy effect, i.e. the rise of warmer fluid and fall the cooler fluid. Whereas in forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or fan.

A. Mechanism of Forced Convection

Convection heat transfer is complicated since it involves flui d motion as well as heat conduction. The fluid motion enhan ces heat transfer (the higher the velocity the higher the heat transfer rate).

The rate of convection heat transfer is expressed by Newton' s law of cooling:

$$q_{CONV} = h\ (T_S - T_\infty)\ \ldots\ldots\ldots(W/m^2)$$

$$Q_{CONV} = h\ A\ (T_S - T_\infty)\ \ldots\ldots\ldots\ldots(W)$$

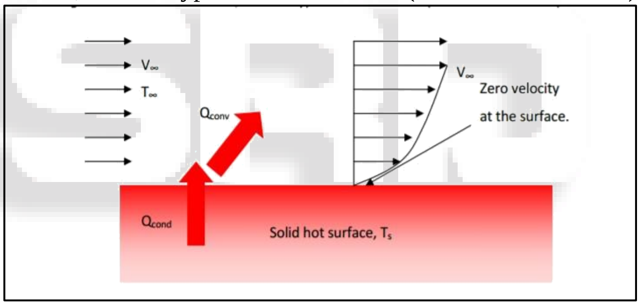
Where,

h = Heat transfer coefficient of

T_S = Surface temperature

T_∞ = Surrounding temperature

The convective heat transfer coefficient (h) strongl y depends on the fluid properties and roughness of the solid surface, and the type of the fluid flow (laminar or turbulent).



It is assumed that the velocity of the fluid is zero at the wall, this assumption is called no-slip condition. As a result, the heat transfer from the solid surface to the fluid layer adjacent to the surface is by pure conduction, since the fluid is motionless. The convection heat transfer coefficient, in general, varies along the flow direction. The mean or average convection heat transfer coefficient for a surface is determined by (properly) averaging the local heat transfer coefficient over the entire surface.

B. Forced Convection Heat Transfer

A common situation encountered by the chemical engineer is heat transfer to fluid flowing through a tube. This can occur in heat exchangers, boilers, condensers, evaporators, and a host of other process equipment. Therefore, it is useful to know how to estimate heat transfer coefficients in this situation.

We can classify the flow of a fluid in a straight circular tube into either laminar or turbulent flow. It is assumed from here on that we assume fully developed incompressible, Newtonian, steady flow conditions. Fully developed flow implies that the tube is long compared with the entrance length in which the velocity distribution at the inlet adjusts itself to the geometry and no longer changes with distance along the tube.

C. Forced Convection Correlations

Our objective is to determine heat transfer coefficients (local and average) for different flow geometries and this heat transfer coefficient (h) may be obtained by experimental or theoretical methods. Theoretical methods involve solutions of the boundary layer equations to get the Nusselt number. On the other hand the experimental methods involve performing heat transfer measurement under controlled laboratory conditions and correlating data in based on experiment data which needs uncertainty analysis, although the experiments are performed under carefully controlled conditions. The causes of the uncertainty are many. Actual situations rarely conform completely to the experimental situations for which the correlations are applicable. Hence one should not expect the actual value of the heat transfer coefficient to be within better than 10% of the predicted value.

III. ANALYSIS ON DIMENSIONAL LESS NUMBER

A. Reynolds Number

The value of the Reynolds number permits us to determine whether the flow is laminar or turbulent. We define the Reynolds number as follows.

$$V = \frac{Vin + Vout}{2}$$

$$Re = \frac{\rho\, V\, d}{\mu} = \frac{V\, d}{\nu}$$

Here, d is the inside diameter of the tube (or pipe), V is the average velocity of the fluid, ρ is the density of the fluid and μ is its dynamic viscosity. ν is the kinematic viscosity

$$\nu = \frac{\mu}{\rho}$$

A good example of laminar and turbulent flow is the rising smoke from a cigarette. The smoke initially travels in smooth, straight lines (laminar flow) then starts to “wave” back and forth (transition flow) and finally seems to randomly mix (turbulent flow). These ranges are discussed below.

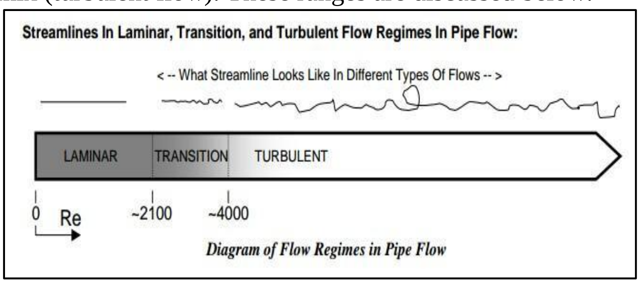


Diagram of Flow Regimes in Pipe Flow

For flow through circular pipes

1) RANGE 1 : Laminar Flow (0-2300) : Generally, a fluid flow is laminar from Re = 0 to some critical value at which transition flow begins.
2) RANGE 2 : Transition Flow (2300-4000) : Flows in this range may fluctuate between laminar and turbulent flow. The fluid flow is on the verge of becoming turbulent.
3) RANGE 3 : Turbulent Flow (4000 <)

The fluid flow has become unstable. In turbulent flow, there is increased mixing that results in viscous losses which are generally much higher than in those in laminar flow.

NOTE: The Re at which turbulent flow begins depends on the geometry of the fluid flow. The value is different for pipe flow and external flow (i.e. over/outside and object). Since we are studying fluid flow in hydraulic systems, we will consider only internal flows (pipe flows).

The flow in a commercial circular tube or pipe is usually laminar when the Reynolds number is below 2,300. In the range, the status of the flow is in transition and for, flow can be regarded as turbulent. Results for heat transfer in the transition regime are difficult to predict, and it is best to avoid this regime in designing heat exchange equipment. Turbulent flow is inherently unsteady, being characterized by time-dependent fluctuations but here we only focus on steady conditions when we discuss either laminar or turbulent flow.

B. Prandtl Number

The Prandtl number plays an important role in heat transfer. It is the ratio of ability of fluid to transfer momentum by molecular means to the ability of fluid to transfer energy by molecular means. The correlation of Prandtl number is given below.

$$Pr = \frac{\nu}{\alpha} = \frac{\text{Ability of a fluid to transport momentum by molecular means}}{\text{Ability of that fluid to transport energy by molecular means}}$$

$$Pr = \frac{\mu\, C_P}{k} = \frac{\nu}{\alpha}$$

Where,
k= Thermal conductivity of the fluid
C_P = Specific heat of the fluid at constant pressure
μ = Dynamic viscosity of fluid
ν = Kinematic viscosity of fluid
α = Thermal diffusivity of the fluid

Gases typically have Prandtl numbers in the range 0.7 -1, while the Prandtl number for most liquids is much larger than unity. The Prandtl number for water ranges from 4-7, while that for an oil might be of the order of 50-100. It is not uncommon to encounter Prandtl numbers for viscous liquids that are of the order of several thousand or even larger.

C. Nusselt Number

In the heat transfer at a boundary (surface) within a fluid, the Nusselt number (Nu) is the ratio of convective to conductive heat transfer across (normal to) the boundary. A larger Nusselt number corresponds to more active convective, with turbulent flow typically in the 100-1000 range.

$$Nu = \frac{h\, x}{k}$$

Here,
h = Heat transfer coefficient
x = Distance from free edge
k = Thermal conductivity

IV. RELATION BETWEEN RE, PR AND NU

In boundary layer flow over a flat plate, experiments confirm that, after a certain length of flow, a laminar boundary layer will become unstable and turbulent. This instability occurs across different scales and with different fluids, usually when $Re_X \approx 5\times10^5$, where x is the distance from the leading edge of the flat plate, and the flow velocity is the freestream velocity of the fluid outside the boundary layer.

These transition Reynolds numbers are also called critical Reynolds numbers, and were studied by Osborne Reynolds around 1895. The critical Reynolds number is different for every geometry. For flow in circular pipes the following relations are known.

A. Laminar Flow ($Re_X < 3\times10^5$)

The average correlation for laminar flow over a flat plate is given by

$$Nu = 0.664\,(Re)^{0.5}\,(Pr)^{(1/3)}$$

B. Turbulent Flow ($Re_X > 5\times10^5$)

The average correlation for turbulent flow over a flat plate is given by

$$Nu = 0.023\,(Re_X)^{(0.8)}\,(Pr)^{(0.4)}$$

V. Drying of Orange Peels

If at all possible, use organic navel oranges. Their thick peels make them the easiest to work with and there is no need to worry about any pesticide residue.

1) Using a good, sharp vegetable peeler (or manually by hand) strips of orange peels are taken out from top to bottom. Scrap as much of the white pith from the underside of the peels as you can because it won't dry properly and it has a bitter flavor.
2) Stack a few strips together at a time and slice them crosswise into thin (about 1/8-inch) pieces.
3) Spread the orange peel in a single layer with the peel side down on baking sheet and bake in a 200°F = 93.33°C oven until they curl and harden slightly - 25 to 30 minutes. Check at regular intervals (about 5 minutes) that the peels do not burn due to overheating.

Remove from the oven and let them cool completely. Store in an airtight container in the refrigerator for up to 3 months. The same method can be used for lemons, tangerines and limes.

VI. Data for Experiment

For our experiment we have taken the drying conditions for 4 kg orange peels to be as follows.

So,

1) Mass of orange peel used is (M) = 4 kg
2) Amount of moisture content in peels(x) = 80 %
3) Specific heat of orange peels (c_{ps}) = 3.81 KJ/Kg.
4) Specific heat of air (c_{pg}) = 1.005 KJ/Kg.
5) Latent heat of water (h_w) = 2257 KJ/Kg.
6) Now, the initial temperature of peels is measured to be (T_3) = 30 ^{0}C.
7) Temperature of peels for drying purpose is (T_4) = 90 ^{0}C. (as earlier suggested value of 200^0 F)
8) Initial temperature of air (T_S) = 33 ^{0}C.
9) NOTE**:** Temperature of air at inlet of compartment is taken to be higher than 90^{0}C since it is required to raise the temperature of water content in peels to about 200^0 F. So, compartment inlet temperature is taken to be (T_1) - 130 ^{0}C
10) The exhaust of the furnace compartment is to be used for preheating of peels at about 50^0 C before being admitted to the furnace. So, temperature of compartment at outlet (T_2) - 50 ^{0}C.

VII. Calculations

A. Mass Flow Rate of Air Required (G) - (REF7,REF8)

By Energy balance

$G^* c_{pg} * (T_1 – T_2) = S^*[\, c_{ps} *(T_3 – T_4) + (x^* h_w)\,]$

G * 1.005 * (130 – 50) = 4 * 3.81 * (90 – 30) + (4 * 0.8 * 2257)

G = 101.203 Kg/ hr

= 0.0281 Kg/sec

Finally mass flow rate of air 0.1 Kg/ sec selected.

Density of air at 130 degree= ρ = 0.876 Kg/ m^3

Volume flow rate of air = Mass flow rate / density of air

= 0.1 / 0.876

= 0.1142 m^3/sec

= 0.1142 * 2118.88 CFM (Where, CFM = cubic feet per minute)

= 241.97 CFM ≈ 242 CFM

So exhaust fan of minimum discharge 242 CFM must be selected.

For the fan of 242 cfm discharge, 160 mm diameter fan is used. Therefore the duct is considered to be cylindrical in shape with diameter (d) = 0.16 m

Now for the velocity of air flowing in the duct,

Air velocity (v) = Air discharge/ area of duct

= 0.1142 / (π * 0.16 ^{2} /4)

= 5.678 m/s

Temperature - t - (°C) (deg F)	Density - ρ - (kg/m^3) (slugs/cu.ft)	Specific Heat - c_p - (kJ/(kg K)) (Btu/lb F)	Thermal Conductivity - k - (W/(m K))	Kinematic Viscosity - ν - x 10^{-6} (m^2/s)	Expansion Coefficient - b - x 10^{-3} (1/K)	Prandtl's Number - P_r -
-150	2.793	1.026	0.0116	3.08	8.21	0.76
-100	1.980	1.009	0.0160	5.95	5.82	0.74
-50	1.534	1.005	0.0204	9.55	4.51	0.725
0	1.293	1.005	0.0243	13.30	3.67	0.715
20	1.205	1.005	0.0257	15.11	3.43	0.713
40	1.127	1.005	0.0271	16.97	3.20	0.711
60	1.067	1.009	0.0285	18.90	3.00	0.709
80	1.000	1.009	0.0299	20.94	2.83	0.708
100	0.946	1.009	0.0314	23.06	2.68	0.703
120	0.898	1.013	0.0328	25.23	2.55	0.70
140	0.854	1.013	0.0343	27.55	2.43	0.695
160	0.815	1.017	0.0358	29.85	2.32	0.69
180	0.779	1.022	0.0372	32.29	2.21	0.69
200	0.746	1.026	0.0386	34.63	2.11	0.685
250	0.675	1.034	0.0421	41.17	1.91	0.68
300	0.616	1.047	0.0454	47.85	1.75	0.68
350	0.566	1.055	0.0485	55.05	1.61	0.68
400	0.524	1.068	0.0515	62.53	1.49	0.68

Table 1: Data for air properties at different temperatures.

From the above table we get values of important parameters for the calculation.

B. Convective Heat Transfer Coefficient of Air (h) for Heating by Coil

By interpolation air properties at 130 ^{0}C are:

Kinematic viscosity (ν) = 26.39 * 10^{-6} (m^3/sec)

Density (ρ) = 0.876 Kg/m^3

Prandtl's number = 0.6975

Thermal conductivity (k) = 0.03355 W/ m K

Specific heat (C_p) = 1.013 KJ/Kg K

NOTE**:** For the conducted experiment we have taken a coil of 180 W rating with thickness of 0.0152 m. This thickness of the coil is taken as characteristic length (L)
So, characteristic length (L) = 0.0152 m
Diameter of coil is (D) = 0.16 m
Internal temperature of coil is about 130 ^{0} C
And surrounding temperature is taken as 33^0 C
So, in this process air at 33^0 C is being heated to 130^0 C and heat transfer Q is given by

Q = mass flow rate(m_a) × change in enthalpy(h_2–h_1)

$= m_a C_p (T_1\text{-}T_s)$

$= \rho A V C_p (T_1\text{-}T_s)$

$= \rho (\pi/4 * D^2) V C_p (T_1\text{-}T_s)$

$= 0.876 * (\pi/4) * 0.16^2 * 5.678 * 1.013 * (130\text{-}33)$

$= 9.83$ W

Also,

$Q = h$ * perimetric area * $[(T_1\text{-}T_s)/2 - T_s]$

$h = Q / \{ \pi D * L * [(T_1\text{-}T_s)/2 - T_s]\}$

$h = 26.528$ W/m^2 K

C. Convective Heat Transfer Coefficient of Air (h) Considering Nu Number Correlation:

Here we have considered a flat plate of length 0.0152 m & width (π * 0.160) m.
So, from above data,

Reynolds number (Re) $= \frac{\rho V d}{\mu} = \frac{V d}{\nu}$

$= \frac{5.678 * 0.0152}{26.39 * 10^{-6}}$

= 3270.39 (> 2000 so flow is turbulent)
Correlation used is,

$Nu = 0.023 * (Re)^{0.8} * (Pr)^{0.4}$ (since flow is turbulent)

$Nu = 0.664 * (3270.39)^{0.5} * (0.6975)^{(1/3)}$

$Nu = 33.676$

We know that,

$Nu = (h * L) / k$

Or, $h = (Nu * k) / L$

Or, $h = 74.33$ W/m^{2}K

D. Electric Heater Selection

Heater is required to heat the supplied air upto 130 ^{0}C from the atmospheric temperature of about
33 ^{0}C.
So,

Wattage Q = h * area of plate * $(\frac{T_1 + T_4}{2} - T_s)$

$= 74.33 * \pi * (0.1)^2 * (\frac{130+90}{2} - 33)$

W = 179.8 Watt ≈ 180 Watt

Hence we used the standard rated heater of 180 Watts.
Minimum power of electric heater must be greater than 180 Watts.

E. Furnace Capacity

4 kg of peels having volume approximately 10000 cm^3 or .01 m^3 .So volume of furnace is for compatibility is taken as 1ft * 1ft * 1ft

V= 1* 1 * 1 ft^3 = 28316.85 cm^3

= 0.02831685 m^3

VIII. Conclusion

Air drying temperature had an important influence on the drying rate. The drying temperature affected significantly the drying time of samples, but this response was not affected significantly by air flow rate. Higher drying temperature promoted shorter drying times, whereas longer drying periods were required at lower drying temperatures.
The following results were obtained from the above experiment

1) Mass flow rate of air required = 0.0281 Kg/sec
2) Power rating of heater required = 180 Watt
3) Volume capacity of furnace = 0.02831685 m^3

A. Improvements

During the optimization of processes, several response variables describing the quality characteristics and performance of the systems are usually optimized. Some of these variables are to be maximized while some are to be minimized. In many cases, these responses are competing, i.e., improving one response may have an opposite effect on another one, which further complicates the situation. It can be seen that there are areas to improve and optimize the drying process as a whole.

1) Work can be done on reducing the power consumption of the machine per Kg of peel dried.
2) The heat lost due to radiation, through open spaces can be minimized to conserve electrical energy used.
3) Work can be done to incorporate solar energy for drying purpose. This would in turn save much of the energy needed for drying.
4) The machine can be modified to intake higher volumes and thus allow mass production.

References

[1] Experimental Prediction of Heat Transfer Correlations in Heat Exchangers Tomasz Sobota Cracow University of Technology Poland.
[2] The effect of prandtl number and viscosity variable on free convection boundary layer flow of a viscoelastic fluid past an elliptic cylinder 1basuki widodo, 2annisadwi sulistyaningtyas, 3chairul imron, International Journal of Mechanical And Production Engineering, ISSN: 2320-2092, Issue-8, Aug.-2015
[3] Simulations of the Near-Wall Heat Transfer at Medium Prandtl Numbers Robert Bergant, Iztok Tiselj "Jožef Stefan" Institute Jamova 39, SI-1000 Ljubljana, Slovenia robert.bergant@ijs.si, iztok.tiselj@ijs.si
[4] Experimental study of Nusselt number and Friction factor in solar air heater duct with Diamond shaped rib roughness on absorber plate S. S. Pawar1, Dr. D. A. Hindolia2 , Dr. J. L. Bhagoria3 1Research scholar, Rajeev Gandhi Prodhyogiki Vishwvidhyalaya, Bhopal (M.P.) 462026, India. 2Principal, Jai Narayan College of Technology, Bhopal (M.P.) 402046, India. 3Professor, Mechanical Engineering department, M.A.N.I.T., Bhopal, (M.P.) 462046, India.
[5] Numerical Study of the Effect of the Reynolds Numbers on Thermal and Hydrodynamic Parameters of Turbulent Flow Mixed Convection Heat Transfer in an Inclined Tube Vahidinia. F. – Miri, M. Farhad Vahidinia* –

Mohadeseh Miri University of Zabol, Department of Mechanical Engineering, Iran
[6] Comparative Study between Heat Transfer through Laminar Flow and Turbulent Flow Pranit M. Patil1 , Amol P. Yadav2, Dr. P. A. Patil3 Assistant Professor, Department of Mechanical Engineering, G.I.T, Lavel, Khed (M.S.) India1 Assistant Professor, Department of Mechanical Engineering, RMCOE, Pedhambe (M.S.) India 2 Professor, Department of Mechanical Engineering, J.S.C.O.E, Hadapsar, Pune (M.S.) India3
[7] Wilhelm, Luther R., Dwayne A. Suter, and Gerald H. Brusewitz. 2004. Drying and Dehydration. Chapter 10 in Food & Process Engineering Technology, 259-284. St. Joseph, Michigan: ASAE. Ðu169© American Society of Agricultural Engineers. (Rev. Aug. 2005.)
[8] Thermodynamic model of microwave drying of citrus peel with different treatments: Clara Talensa, Marta Castro-Giráldezb, Carlos Balda, Pedro J. Fitob AZTI-Tecnalia, Unidad de Investigación Alimentaria, Derio, Spain Laboratorio de Propiedades Dieléctricas, Instituto Universitario de Ingeniería de Alimentos para el Desarrollo, IIAD, Universidad Politécnica de Valencia, Spain.
[9] Sagar V. R. and Suresh Kumar P. Association of Food Scientists and Technologists (India), Mysore: Recent advances in drying and dehydration of fruits and vegetables: a review