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Transfert de chaleur (3L)

Module : .... **Année :** 21/20 **Spécialité :** .... **Groupe :**………….. **Durée :** ….. min
Nom et prénom : ....:الاسم واللقب:.................................. **Matricule:**……………….....

Heat *Conduction* Rate Equations (Fourier's Law)

- Heat Flux: $q''_x = -k\frac{dT}{dx}$ $\frac{W}{m^2}$ k: *Thermal Conductivity* $\frac{W}{m \cdot k}$
- Heat Rate: $q_x = q''_x A_c$ W A_c: Cross-Sectional Area

Heat *Convection* Rate Equations (Newton's Law of Cooling)

- Heat Flux: $q'' = h(T_s - T_\infty)$ $\frac{W}{m^2}$ h: *Convection Heat Transfer Coefficient* $\frac{W}{m^2 \cdot K}$
- Heat Rate: $q = hA_s(T_s - T_\infty)$ W A_s: Surface Area m^2

Heat *Radiation* emitted ideally by a blackbody surface has a surface *emissive power:* $E_b = \sigma T_s^4$ $\frac{W}{m^2}$

- Heat Flux emitted: $E = \varepsilon\sigma T_s^4$ $\frac{W}{m^2}$ where ε is the *emissivity* with range of $0 \leq \varepsilon \leq 1$

  and $\sigma = 5.67 \times 10^{-8}$ $\frac{W}{m^2 K^4}$ is the *Stefan-Boltzmann constant*
- Irradiation: $G_{abs} = \alpha G$ but we assume small body in a large enclosure with $\varepsilon = \alpha$ so that $G = \varepsilon\,\sigma\,T_{sur}^4$
- Net Radiation heat flux from surface: $q''_{rad} = \frac{q}{A} = \varepsilon E_b(T_s) - \alpha G = \varepsilon\sigma(T_s^4 - T_{sur}^4)$
- Net radiation heat exchange rate: $q_{rad} = \varepsilon\sigma A_s(T_s^4 - T_{sur}^4)$ where for a real surface $0 \leq \varepsilon \leq 1$

This can ALSO be expressed as: $q_{rad} = h_r A(T_s - T_{sur})$ depending on the application

where h_r is the *radiation heat transfer coefficient* which is: $h_r = \varepsilon\sigma(T_s + T_{sur})(T_s^2 + T_{sur}^2)$ $\frac{W}{m^2 \cdot K}$

- TOTAL heat transfer from a surface: $q = q_{conv} + q_{rad} = hA_s(T_s - T_\infty) + \varepsilon\sigma A_s(T_s^4 - T_{sur}^4)$ W

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Transfert thermique et Echangeurs de chaleur & Phénomène de Transfert II

Module : **UEM1.1** **Année :** 21/20 **Spécialité :** ………………… **Groupe :**………….. **Durée :** 60 min

Nom et prénom : ……………………………………………… **:الاسم واللقب**:…………………………… **Matricule:**……………………

Heat Equation (used to find the temperature distribution)

Heat Equation (Cartesian): $\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{q}=\rho c_p\frac{\partial T}{\partial t}$

If k is constant then the above simplifies to: $\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}+\frac{\dot{q}}{k}=\frac{1}{\alpha}\frac{\partial T}{\partial t}$ where $\alpha=\frac{k}{\rho c_p}$ is the *thermal diffusivity*

Heat Equation (Cylindrical): $\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial}{\partial \phi}\left(k\frac{\partial T}{\partial \phi}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{q}=\rho c_p\frac{\partial T}{\partial t}$

Heat Eqn. (Spherical): $\frac{1}{r^2}\frac{\partial}{\partial r}\left(kr^2\frac{\partial T}{\partial r}\right)+\frac{1}{r^2\sin\theta^2}\frac{\partial}{\partial \phi}\left(k\frac{\partial T}{\partial \phi}\right)+\frac{1}{r^2\sin\theta}\frac{\partial}{\partial \theta}\left(k\sin\theta\frac{\partial T}{\partial \theta}\right)+\dot{q}=\rho c_p\frac{\partial T}{\partial t}$

Thermal Circuits

Plane Wall: $R_{t,cond}=\frac{L}{kA}$ Cylinder: $R_{t,cond}=\frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi kL}$ Sphere: $R_{t,cond}=\frac{\left(\frac{1}{r_1}-\frac{1}{r_2}\right)}{4\pi k}$

$R_{t,conv}=\frac{1}{hA}$ $R_{t,rad}=\frac{1}{h_r A}$

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Convection Heat Transfer

$$Re = \frac{\rho V L_c}{\mu} = \frac{V L_c}{\nu} \quad \text{[Reynolds Number]} \quad ; \quad \overline{Nu} = \frac{\bar{h} L_c}{k_f} \quad \text{[Average Nusselt Number]}$$

where ρ is the density, V is the velocity, L_c is the characteristic length, μ is the dynamic viscosity, ν is the kinematic viscosity, $\dot{m}$ is the mass flow rate, $\bar{h}$ is the average convection coefficient, and k_f is the fluid thermal conductivity.

Internal Flow

$$Re = \frac{4\,\dot{m}}{\pi D \mu} \quad \text{[For Internal Flow in a Pipe of Diameter } D\text{]}$$

For Constant Heat Flux $[q_s'' = constant]$: $\quad q_{conv} = q_s''(P \cdot L)$; where P = Perimeter, L = Length

$$T_m(x) = T_{m,i} + \frac{q_s'' \cdot P}{\dot{m} \cdot c_p} x$$

For Constant Surface Temperature $[T_s = constant]$:

If there is only convection between the surface temperature, T_s, and the mean fluid temperature, T_m, use

$$\frac{T_s - T_m(x)}{T_s - T_{m,i}} = exp\left(-\frac{P \cdot x}{\dot{m} \cdot c_p}\bar{h}\right)$$

If there are multiple resistances between the outermost temperature, T_∞, and the mean fluid temperature, T_m, use

$$\frac{T_\infty - T_m(x)}{T_\infty - T_{m,i}} = exp\left(-\frac{P \cdot x}{\dot{m} \cdot c_p} U\right) = exp\left(-\frac{1}{\dot{m} \cdot c_p \cdot R_t}\right)$$

Total heat transfer rate over the entire tube length:

$$q_t = \dot{m} \cdot c_p \cdot (T_{m,o} - T_{m,i}) = \bar{h} \cdot A_s \cdot \Delta T_{lm} \; or \; U \cdot A_s \cdot \Delta T_{lm} \quad ; \quad T_s = constant$$

Log mean temperature difference: $\quad \Delta T_{lm} = \dfrac{\Delta T_o - \Delta T_i}{\ln\left(\frac{\Delta T_o}{\Delta T_i}\right)} \quad ; \quad \Delta T_o = T_s - T_{m,o} \quad ; \quad \Delta T_i = T_s - T_{m,i}$

Free Convection Heat Transfer

$$Gr_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} \qquad \text{[Grashof Number]}$$

$$Ra_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu\alpha} \qquad \text{[Rayleigh Number]}$$

Vertical Plates: $$\overline{Nu}_L = \left\{0.825 + \frac{0.387\, Ra_L{}^{1/6}}{\left[1+\left(\frac{0.492}{Pr}\right)^{9/16}\right]^{8/27}}\right\}^2$$; [Entire range of Ra_L; properties evaluated at T_f]

- For better accuracy for *Laminar Flow*: $\overline{Nu}_L = 0.68 + \dfrac{0.670\, Ra_L^{1/4}}{\left[1+\left(\frac{0.492}{Pr}\right)^{9/16}\right]^{4/9}}$; $Ra_L \lesssim 10^9$ [Properties evaluated at T_f]

Inclined Plates: for the *top* and *bottom* surfaces of *cooled* and *heated* inclined plates, respectively, the equations of the vertical plate can be used by replacing (g) with ($g\cos\theta$) in Ra_L for $0 \le \theta \le 60°$.

Horizontal Plates: use the following correlations with $L = \dfrac{A_s}{P}$ where A_s = Surface Area and P = Perimeter

- Upper surface of Hot Plate or Lower Surface of Cold Plate:

$$\overline{Nu}_L = 0.54\, Ra_L^{1/4} \quad (10^4 \le Ra_L \le 10^7, Pr \ge 0.7)\ ;\ \overline{Nu}_L = 0.15\, Ra_L^{1/3} \quad (10^7 \le Ra_L \le 10^{11}, all\ Pr)$$

- Lower Surface of Hot Plate or Upper Surface of Cold Plate:

$$\overline{Nu}_L = 0.52\, Ra_L^{1/5} \qquad (10^4 \le Ra_L \le 10^9, Pr \ge 0.7)$$

Vertical Cylinders: the equations for the *Vertical Plate* can be applied to vertical cylinders of height L if the following criterion is met: $\dfrac{D}{L} \ge \dfrac{35}{Gr_L^{1/4}}$

Long Horizontal Cylinders: $$\overline{Nu}_D = \left\{0.60 + \frac{0.387\, Ra_D^{1/6}}{\left[1+\left(\frac{0.559}{Pr}\right)^{9/16}\right]^{8/27}}\right\}^2$$; $Ra_D \lesssim 10^{12}$ [Properties evaluated at T_f]

Spheres: $$\overline{Nu}_D = 2 + \frac{0.589\, Ra_D^{1/4}}{\left[1+\left(\frac{0.469}{Pr}\right)^{9/16}\right]^{4/9}}$$; $Ra_D \lesssim 10^{11}$; $Pr \ge 0.7$ [Properties evaluated at T_f]

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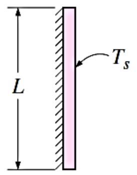
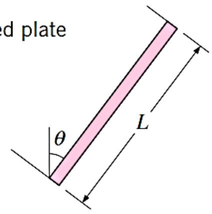
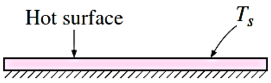
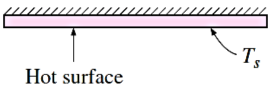
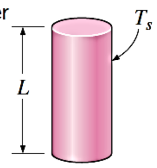
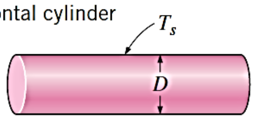
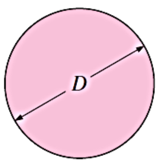
Transfert thermique et Echangeurs de chaleur & Phénomène de Transfert II

Module : **UEM1.1** **Année :** 21/20 **Spécialité :** **…………………Groupe :**………….. **Durée :** 60 min

Nom et prénom : ……………………………………………………:الاسم واللقب:……………………………… **Matricule:**……………………

Geometry	Characteristic length L_c	Range of Ra	Nu
Vertical plate 	L	10^4–10^9 10^9–10^{13} Entire range	$Nu = 0.59Ra_L^{1/4}$ $Nu = 0.1Ra_L^{1/3}$ $Nu = \left\{0.825 + \frac{0.387Ra_L^{1/6}}{[1+(0.492/Pr)^{9/16}]^{8/27}}\right\}^2$ (complex but more accurate)
Inclined plate 	L		Use vertical plate equations for the upper surface of a cold plate and the lower surface of a hot plate Replace g by $g\cos\theta$ for $Ra < 10^9$
Horizontal plate (Surface area A and perimeter p) (*a*) Upper surface of a hot plate (or lower surface of a cold plate)  (*b*) Lower surface of a hot plate (or upper surface of a cold plate) 	A_s/p	10^4–10^7 10^7–10^{11} 10^5–10^{11}	$Nu = 0.54Ra_L^{1/4}$ $Nu = 0.15Ra_L^{1/3}$ $Nu = 0.27Ra_L^{1/4}$
Vertical cylinder 	L		A vertical cylinder can be treated as a vertical plate when $D \geq \frac{35L}{Gr_L^{1/4}}$
Horizontal cylinder 	D	$Ra_D \leq 10^{12}$	$Nu = \left\{0.6 + \frac{0.387Ra_D^{1/6}}{[1+(0.559/Pr)^{9/16}]^{8/27}}\right\}^2$
Sphere 	D	$Ra_D \leq 10^{11}$ $(Pr \geq 0.7)$	$Nu = 2 + \frac{0.589Ra_D^{1/4}}{[1+(0.469/Pr)^{9/16}]^{4/9}}$

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Transfert thermique et Echangeurs de chaleur & Phénomène de Transfert II

Module : **UEM1.1** **Année :** 21/20 **Spécialité :** ………………… **Groupe :**………….. **Durée :** 60 min

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Résumé des corrélations de transfert de chaleur par convection pour le flux externe

Correlation	Geometry	Conditions[c]
$\delta = 5x\, Re_x^{-1/2}$	Flat plate	Laminar, T_f
$C_{f,x} = 0.664\, Re_x^{-1/2}$	Flat plate	Laminar, local, T_f
$Nu_x = 0.332\, Re_x^{1/2}\, Pr^{1/3}$	Flat plate	Laminar, local, T_f, $Pr \gtrsim 0.6$
$\delta_t = \delta\, Pr^{-1/3}$	Flat plate	Laminar, T_f
$\overline{C}_{f,x} = 1.328\, Re_x^{-1/2}$	Flat plate	Laminar, average, T_f
$\overline{Nu}_x = 0.664\, Re_x^{1/2}\, Pr^{1/3}$	Flat plate	Laminar, average, T_f, $Pr \gtrsim 0.6$
$Nu_x = 0.564\, Pe_x^{1/2}$	Flat plate	Laminar, local, T_f, $Pr \lesssim 0.05$, $Pe_x \gtrsim 100$
$C_{f,x} = 0.0592\, Re_x^{-1/5}$	Flat plate	Turbulent, local, T_f, $Re_x \lesssim 10^8$
$\delta = 0.37x\, Re_x^{-1/5}$	Flat plate	Turbulent, T_f, $Re_x \lesssim 10^8$
$Nu_x = 0.0296\, Re_x^{4/5}\, Pr^{1/3}$	Flat plate	Turbulent, local, T_f, $Re_x \lesssim 10^8$, $0.6 \lesssim Pr \lesssim 60$
$\overline{C}_{f,L} = 0.074\, Re_L^{-1/5} - 1742\, Re_L^{-1}$	Flat plate	Mixed, average, T_f, $Re_{x,c} = 5 \times 10^5$, $Re_L \lesssim 10^8$
$\overline{Nu}_L = (0.037\, Re_L^{4/5} - 871)Pr^{1/3}$	Flat plate	Mixed, average, T_f, $Re_{x,c} = 5 \times 10^5$, $Re_L \lesssim 10^8$, $0.6 \lesssim Pr \lesssim 60$
$\overline{Nu}_D = C\, Re_D^m\, Pr^{1/3}$ (Table 7.2)	Cylinder	Average, T_f, $0.4 \lesssim Re_D \lesssim 4 \times 10^5$, $Pr \gtrsim 0.7$
$\overline{Nu}_D = C\, Re_D^m\, Pr^n (Pr/Pr_s)^{1/4}$ (Table 7.4)	Cylinder	Average, T_∞, $1 \lesssim Re_D \lesssim 10^6$, $0.7 \lesssim Pr \lesssim 500$
$\overline{Nu}_D = 0.3 + [0.62\, Re_D^{1/2}\, Pr^{1/3} \times [1 + (0.4/Pr)^{2/3}]^{-1/4}] \times [1 + (Re_D/282{,}000)^{5/8}]^{4/5}$	Cylinder	Average, T_f, $Re_D\, Pr \gtrsim 0.2$
$\overline{Nu}_D = 2 + (0.4\, Re_D^{1/2} + 0.06\, Re_D^{2/3})Pr^{0.4} \times (\mu/\mu_s)^{1/4}$	Sphere	Average, T_∞, $3.5 \lesssim Re_D \lesssim 7.6 \times 10^4$, $0.71 \lesssim Pr \lesssim 380$, $1.0 \lesssim (\mu/\mu_s) \lesssim 3.2$

Ferhat-mohammedfouad@univ-eloued.dz

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Résumé des corrélations de convection pour l'écoulement dans un tube circulaire

Correlation	Conditions
$f = 64/Re_D$	Laminar, fully developed
$Nu_D = 4.36$	Laminar, fully developed, uniform q''_s
$Nu_D = 3.66$	Laminar, fully developed, uniform T_s
$\overline{Nu}_D = 3.66 + \dfrac{0.0668\, Gz_D}{1 + 0.04\, Gz_D^{2/3}}$	Laminar, thermal entry (or combined entry with $Pr \gtrsim 5$), uniform T_s, $Gz_D = (D/x)\, Re_D\, Pr$
$\overline{Nu}_D = \dfrac{\dfrac{3.66}{\tanh[2.264\, Gz_D^{-1/3} + 1.7\, Gz_D^{-2/3}]} + 0.0499\, Gz_D \tanh(Gz_D^{-1})}{\tanh(2.432\, Pr^{1/6}\, Gz_D^{-1/6})}$	Laminar, combined entry, $Pr \gtrsim 0.1$, uniform T_s, $Gz_D = (D/x)\, Re_D\, Pr$
$\dfrac{1}{\sqrt{f}} = -2.0 \log\left[\dfrac{e/D}{3.7} + \dfrac{2.51}{Re_D \sqrt{f}}\right]$	Turbulent, fully developed
$f = (0.790 \ln Re_D - 1.64)^{-2}$	Turbulent, fully developed, smooth walls, $3000 \lesssim Re_D \lesssim 5 \times 10^6$
$Nu_D = 0.023\, Re_D^{4/5}\, Pr^n$ $Nu_D = 0.027\, Re_D^{4/5}\, Pr^{1/3}\left(\dfrac{\mu}{\mu_s}\right)^{0.14}$ $Nu_D = \dfrac{(f/8)(Re_D - 1000)\, Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$	Turbulent, fully developed, $0.6 \lesssim Pr \lesssim 160$, $Re_D \gtrsim 10{,}000$, $(L/D) \gtrsim 10$, $n = 0.4$ for $T_s > T_m$ and $n = 0.3$ for $T_s < T_m$ Turbulent, fully developed, $0.7 \lesssim Pr \lesssim 16{,}700$, $Re_D \gtrsim 10{,}000$, $L/D \gtrsim 10$ Turbulent, fully developed, $0.5 \lesssim Pr \lesssim 2000$, $3000 \lesssim Re_D \lesssim 5 \times 10^6$, $(L/D) \gtrsim 10$
$Nu_D = 4.82 + 0.0185(Re_D\, Pr)^{0.827}$	Liquid metals, turbulent, fully developed, uniform q''_s, $3.6 \times 10^3 \lesssim Re_D \lesssim 9.05 \times 10^5$, $3 \times 10^{-3} \lesssim Pr \lesssim 5 \times 10^{-2}$, $10^2 \lesssim Re_D\, Pr \lesssim 10^4$
$Nu_D = 5.0 + 0.025(Re_D\, Pr)^{0.8}$	Liquid metals, turbulent, fully developed, uniform T_s, $Re_D\, Pr \gtrsim 100$

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Useful conversion factors

Physical quantity	Symbol	SI to English conversion	English to SI conversion
Length	L	1 m = 3.2808 ft	1 ft = 0.3048 m
Area	A	1 m^2 = 10.7639 ft^2	1 ft^2 = 0.092903 m^2
Volume	V	1 m^3 = 35.3134 ft^3	1 ft^3 = 0.028317 m^3
Velocity	v	1 m/s = 3.2808 ft/s	1 ft/s = 0.3048 m/s
Density	ρ	1 kg/m^3 = 0.06243 lb_m/ft^3	1 lb_m/ft^3 = 16.018 kg/m^3
Force	F	1 N = 0.2248 lb_f	1 lb_f = 4.4482 N
Mass	m	1 kg = 2.20462 lb_m	1 lb_m = 0.45359237 kg
Pressure	p	1 N/m^2 = 1.45038×10^{-4} lb_f/in^2	1 lb_f/in^2 = 6894.76 N/m^2
Energy, heat	q	1 kJ = 0.94783 Btu	1 Btu = 1.05504 kJ
Heat flow	q	1 W = 3.4121 Btu/h	1 Btu/h = 0.29307 W
Heat flux per unit area	q/A	1 W/m^2 = 0.317 $Btu/h \cdot ft^2$	1 $Btu/h \cdot ft^2$ = 3.154 W/m^2
Heat flux per unit length	q/L	1 W/m = 1.0403 $Btu/h \cdot ft$	1 $Btu/h \cdot ft$ = 0.9613 W/m
Heat generation per unit volume	$\dot{q}$	1 W/m^3 = 0.096623 $Btu/h \cdot ft^3$	1 $Btu/h \cdot ft^3$ = 10.35 W/m^3
Energy per unit mass	q/m	1 kJ/kg = 0.4299 Btu/lb_m	1 Btu/lb_m = 2.326 kJ/kg
Specific heat	c	1 $kJ/kg \cdot °C$ = 0.23884 $Btu/lb_m \cdot °F$	1 $Btu/lb_m \cdot °F$ = 4.1869 $kJ/kg \cdot °C$
Thermal conductivity	k	1 $W/m \cdot °C$ = 0.5778 $Btu/h \cdot ft \cdot °F$	1 $Btu/h \cdot ft \cdot °F$ = 1.7307 $W/m \cdot °C$
Convection heat-transfer coefficient	h	1 $W/m^2 \cdot °C$ = 0.1761 $Btu/h \cdot ft^2 \cdot °F$	1 $Btu/h \cdot ft^2 \cdot °F$ = 5.6782 $W/m^2 \cdot °C$
Dynamic		1 $kg/m \cdot s$ = 0.672 $lb_m/ft \cdot s$	
Viscosity	μ	= 2419.2 $lb_m/ft \cdot h$	1 $lb_m/ft \cdot s$ = 1.4881 $kg/m \cdot s$
Kinematic viscosity and thermal diffusivity	ν, α	1 m^2/s = 10.7639 ft^2/s	1 ft^2/s = 0.092903 m^2/s

Important physical constants

Avogadro's number	$N_0 = 6.022045 \times 10^{26}$ molecules/kg mol
Universal gas constant	$\mathcal{R} = 1545.35$ $ft \cdot lbf/lbm \cdot mol \cdot °R$
	$= 8314.41$ $J/kg\ mol \cdot K$
	$= 1.986$ $Btu/lbm \cdot mol \cdot °R$
	$= 1.986$ $kcal/kg\ mol \cdot K$
Planck's constant	$h = 6.626176 \times 10^{-34}$ $J \cdot sec$
Boltzmann's constant	$k = 1.380662 \times 10^{-23}$ $J/molecule \cdot K$
	$= 8.6173 \times 10^{-5}$ $eV/molecule \cdot K$
Speed of light in vacuum	$c = 2.997925 \times 10^{8}$ m/s
Standard gravitational acceleration	$g = 32.174$ ft/s^2
	$= 9.80665$ m/s^2
Electron mass	$m_e = 9.1095 \times 10^{-31}$ kg
Charge on the electron	$e = 1.602189 \times 10^{-19}$ C
Stefan-Boltzmann constant	$\sigma = 0.1714 \times 10^{-8}$ $Btu/hr \cdot ft^2 \cdot R^4$
	$= 5.669 \times 10^{-8}$ $W/m^2 \cdot K^4$
1 atm	$= 14.69595$ lbf/in^2 = 760 mmHg at 32°F
	= 29.92 inHg at 32°F = 2116.21 lbf/ft^2
	$= 1.01325 \times 10^5$ N/m^2