Air-Cooled Exchanger: Air-Side Coefficient, Pressure Drop, Fan Power

Air-side film coefficient (Briggs–Young, Ganguli), bundle pressure drop (Robinson–Briggs, Ganguli) and fan and motor power for a finned-tube air cooler.

Input parameters

Unit system: SI

Parameter	Value	Unit
Fin root diameter (tube OD)	31.750	mm
Fin height	12.700	mm
Fin thickness	0.40000	mm
Fins per unit length	11.000	1/in
Tube pitch (equilateral triangular)	76.200	mm
Tube rows	6.0000	
Air mass flow	1,080,000	kg/h
Bundle face area	80.000	m^2
Air density at the mean temperature	1.1300	kg/m^3
Air viscosity	0.018800	mPa·s
Air thermal conductivity	0.026800	W/(m·K)
Air specific heat	1.0070	kJ/(kg·K)
Other air-side losses as a share of the bundle loss	15.000	%
Number of fans	3.0000	
Fan diameter	4.2700	m
Air density at the fan	1.1800	kg/m^3
Fan total efficiency	75.000	%
Speed reducer efficiency	95.000	%

Calculated results

Parameter	Value	Unit
Face velocity	3.3186	m/s
Maximum velocity in the bundle	6.3140	m/s
Reynolds number (root diameter)	12,050	
Finned-to-bare area ratio	16.540	
Briggs–Young coefficient	49.510	$W/(m^2·K)$
Ganguli (HTRI) coefficient	52.682	$W/(m^2·K)$
Robinson–Briggs friction factor	0.43163	
Bundle pressure drop, Robinson–Briggs	233.33	Pa
Ganguli friction factor	0.20904	
Bundle pressure drop, Ganguli	113.01	Pa
Total air-side loss (fan static pressure)	129.96	Pa
Air flow per fan	84.746	m^3/s

Parameter	Value	Unit
Velocity pressure in the fan ring	20.663	Pa
Fan brake power, each	17.019	kW
Motor output power, each	17.915	kW

Notes for this case

- Ganguli et al.
- Fan static pressure taken equal to the air-side loss (induced draft without diffusers); fans should cover at least 40 % of the face area.

Formula and method

$$V_{max} = \frac{P_T V_{face}}{P_T - D_r - 2 n_f b \tau}, \quad Nu = 0.38 Re^{0.6} Pr^{1/3} \left(\frac{A_{Tot}}{A_o}\right)^{-0.15}, \quad Nu = 0.134 Re^{0.681} Pr^{1/3} \left(\frac{s}{b}\right)^{0.2} \left(\frac{s}{\tau}\right)^{0.1134}$$

$$\Delta P = \frac{2 f N_r G^2}{\rho}, \quad f_{RB} = 18.93 Re^{-0.316} \left(\frac{P_T}{D_r}\right)^{-0.927}, \quad W_{fan} = \frac{(\Delta P_o + \rho V_{fr}^2/2)\dot{v}_{fan}}{\eta_{fan}}$$

Variables

D_r	Fin root diameter (tube OD)	m
b	Fin height	m
τ	Fin thickness	m
n_f	Fins per unit length	1/m
P_T	Tube pitch (equilateral triangular)	m
N_r	Tube rows	-
$\dot{m}_a$	Air mass flow	kg/s
A_{face}	Bundle face area	m²
ρ	Air density at the mean temperature	kg/m³
μ	Air viscosity	Pa·s
k	Air thermal conductivity	W/(m·K)
c_p	Air specific heat	J/(kg·K)
χ	Other air-side losses as a share of the bundle loss	-
n_{fan}	Number of fans	-
D_{fan}	Fan diameter	m
ρ_{fan}	Air density at the fan	kg/m³
η_{fan}	Fan total efficiency	-
η_{sr}	Speed reducer efficiency	-
V_{face}	Face velocity	m/s
V_{max}	Maximum velocity in the bundle	m/s
Re	Reynolds number (root diameter)	-
A_{Tot}/A_o	Finned-to-bare area ratio	-
h_{BY}	Briggs–Young coefficient	W/(m²·K)
h_G	Ganguli (HTRI) coefficient	W/(m²·K)
f_{RB}	Robinson–Briggs friction factor	-
ΔP_{RB}	Bundle pressure drop, Robinson–Briggs	Pa
f_G	Ganguli friction factor	-

ΔP_G	Bundle pressure drop, Ganguli	Pa
ΔP_o	Total air-side loss (fan static pressure)	Pa
$\dot{v}_{fan}$	Air flow per fan	m³/s
$\rho V_{fr}^2/2$	Velocity pressure in the fan ring	Pa
W_{fan}	Fan brake power, each	W
W_{motor}	Motor output power, each	W

Method and standards

Finned-tube bank correlations for air-side heat transfer and friction, with fan total-pressure power.

Assumptions and limits

- Equilateral triangular pitch, high-fin tubes, three or more rows.

Report details

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Results are for engineering estimation. Verify critical designs against the referenced standards and your own data.

This calculator online

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Reopens the calculator with the inputs of this report.