1. Introduction

NARSA Heavy Duty Heating and Cooling Conference
Sept 2012 Ann Arbor, MI

Instructor: Joe Borghese
Copyright

• This presentation material presented for the NARSA Education Seminar is copyrighted material

• Original material copyright 2012 © Joseph Borghese
# Course Outline

- Introduction
- Functions and Types of Heat Exchangers
- Heat Exchanger Design Process
- Heat Transfer and Pressure Drop Analyses
- Heat Exchanger Surface Characteristics
- Engine Cooling Systems
- Air Conditioning Systems
- Recent Developments
- Concluding Remarks
References


Acknowledgement

The author would like to acknowledge the support of Ramesh K. Shah who originally presented the SAE course “Compact Heat Exchangers for Automotive Applications”
Compact Heat Exchanger Design, Characteristics and Trends

2. Heat Exchanger Functions and Types

NARSA Heavy Duty Heating and Cooling Conference
Sept 2012 Ann Arbor, MI

Instructor: Joe Borghese
Heat Exchanger Defined

• A device to transfer energy from one fluid mass to another
• A wall must separate the fluids so they do not mix
Why it is not that simple…

• Perform the required heat transfer AND
  – Minimize size and weight
  – Minimize pressure drop
  – Meet required life
  – Be resistant to fouling and contamination
  – Minimize cost
Compact Heat Exchangers

- Compact heat exchangers are a class of heat exchangers that incorporate a large amount of heat transfer surface area per unit volume.

- Most automotive heat exchangers would come into the compact heat exchanger category since space is an extreme constraint for automotive applications.
Classification of Heat Exchangers

Classification according to transfer process
- Indirect contact type
  - Direct transfer type
    - Single-phase
    - Multiphase
  - Storage type
  - Fluidized bed
- Direct contact type
  - Immiscible fluids
  - Gas-liquid
  - Liquid-vapor

Classification according to number of fluids
- Two-fluid
- Three-fluid
- N-fluid ($N > 3$)

Classification according to surface compactness
- Gas-to-fluid
- Liquid-to-liquid and phase-change
  - Compact ($\beta \geq 700 \text{ m}^2/\text{m}^3$)
  - Noncompact ($\beta < 700 \text{ m}^2/\text{m}^3$)
  - Compact ($\beta \geq 400 \text{ m}^2/\text{m}^3$)
  - Noncompact ($\beta < 400 \text{ m}^2/\text{m}^3$)

FROM REF #1
Classification of Heat Exchangers

Classification according to flow arrangements

- Single-pass
  - Counterflow
  - Parallel flow
  - Crossflow
- Multipass
  - Split-flow
  - Divided-flow
  - Extended surface
    - Cross-counterflow
    - Cross-parallellow flow
    - Compound flow
  - Shell-and-tube
    - Parallel counterflow
      - $m$-shell passes
      - $n$-tube passes
    - Split-flow
    - Divided-flow
  - Plate
    - Fluid 1 $m$ passes
    - Fluid 2 $n$ passes

Classification according to heat transfer mechanisms

- Single-phase convection on both sides
- Single-phase convection on one side, two-phase convection on other side
- Two-phase convection on both sides
- Combined convection and radiative heat transfer

FROM REF #1
Classification of Heat Exchangers

Classification according to construction:

- Tubular
  - Double-pipe
    - Crossflow to tubes
    - Parallelflow to tubes
  - Shell-and-tube
- Plate-type
  - PHE
  - Spiral
- Plate coil
  - Printed circuit
- Extended surface
  - Plate-fin
  - Tube-fin
- Regenerative
  - Rotary
  - Fixed-matrix
  - Rotating hoods

FROM REF #1
Exchanger Surface Area Density

\[ \beta = \frac{4\pi}{D_h} \] and \( \sigma = 0.833 \)

\[ \beta = \frac{3333}{D_h} \text{ (mm), m}^2/\text{m}^3 \]

FROM REF #1
Automotive Heat Exchangers

• Coolant heat exchangers (radiators)
  – engine coolant
  – inverter coolant
• Oil coolers (engine, transmission, power steering, hydraulic oil)
• Exhaust Gas Recirculation (EGR) coolers

• Charge air coolers
• Air conditioning
  – heaters
  – evaporators
  – condensers
Automotive Requirements

• Compact
  – Small face area and short flow depth for packaging

• Low pressure drop
  – Reduces pumping power for coolants
  – Increases temperature difference for refrigerants
  – Better charge air density for charge air coolers

• Low weight
  – Reduced material cost
  – Improved fuel economy and or payload

• Low cost and high volume

• Durable
## Quantitative Look at Automotive HX

<table>
<thead>
<tr>
<th>Heat Exchanger</th>
<th>Compactness m²/m³</th>
<th>Performance kW/m³K</th>
<th>Operating Pressure bar</th>
<th>Operating Temp, °C</th>
<th>Mass kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radiator</td>
<td>1000-1500</td>
<td>30-200</td>
<td>1.6-2.5</td>
<td>80-125</td>
<td>1.5-5.0</td>
</tr>
<tr>
<td>Condenser</td>
<td>950-1300</td>
<td>20-100</td>
<td>15-23</td>
<td>75-100</td>
<td>1.2-4.0</td>
</tr>
<tr>
<td>Heater</td>
<td>1800-2800</td>
<td>65-130</td>
<td>1.6-2.5</td>
<td>80-125</td>
<td>0.5-1.2</td>
</tr>
<tr>
<td>Evaporator</td>
<td>900-1000</td>
<td>40-80</td>
<td>3-3.8</td>
<td>3-7</td>
<td>1.2-3.5</td>
</tr>
<tr>
<td>Oil Cooler</td>
<td>500-1500</td>
<td>40-600</td>
<td>3-10</td>
<td>120-150</td>
<td>0.3-1.5</td>
</tr>
<tr>
<td>Charge Air Cooler</td>
<td>600-900</td>
<td>20-60</td>
<td>2-3.5</td>
<td>120-200</td>
<td>0.6-3.5</td>
</tr>
</tbody>
</table>

FROM COWELL REF #3
Compact Heat Exchanger Design, Characteristics and Trends

2. Design Process for Compact Heat Exchangers

NARSA Heavy Duty Heating and Cooling Conference
Sept 2012 Ann Arbor, MI

Instructor: Joe Borghese
Design Process Summary

- Preliminary Design:
  - Materials
  - Configuration
  - Surface Selection
  - Sizing

- Detailed Analysis:
  - Flow Distribution
  - Temperature Distribution
  - Interface Effects
  - Conduction Effects
  - Performance Mapping

- Requirements Review:
  - Basic Physics
  - Operating Conditions and Extremes
  - Envelope and Interfaces
  - Robustness
  - Fluids
  - Cost Targets
  - Delivery

- Manufacturing
- Design
- Structures
- Legacy Designs
- Fluid Properties
- Software Tools
- Surface Properties
- Literature
- Component Tests

Customer

Approved Design

Design Review
Requirements

• Establish design inputs
  – Fluids
  – Operating conditions
  – Available envelope and ducting interfaces
  – Environmental conditions
  – Manufacturing options

• Establish and rank design goals
  – Performance
  – Size and weight
  – Cost
  – Durability
Fluids

- Generally decided at system level
- Heat sink fluid is often ultimately air
  - Low density gas
  - Low specific heat
- Heat sources often liquid cooled
  - Ethylene-Glycol / Water mixtures
  - Propylene-Glycol / Water mixtures
  - Engine oil
  - Hydraulic oil
  - Refrigerants (R134a)
Design Operating Conditions

• Establish operating profile
  – Start, idle, accel, cruise, decel, climb, descend, idle, shutdown
  – Standard day, hot day, cold day and extremes
  – Humidity
  – Altitude (sea level to 10,000 ft ?)

• From operating profile choose design conditions, for example:
  – Extreme hot day (120°F) at 7000 ft
  – High heat load (climb)
  – Low flows (idle)
Envelope and Ducting

• Establish dimensions available for heat exchanger core and fluid manifolds
  – Envelope may determine heat exchanger surface selection

• Determine if fluid interfaces are fixed or can the application accommodate changes
  – Fluid interfaces may dictate heat exchanger flow arrangement

• Flexibility in envelope and ducting will allow optimization for performance, size, weight
Environmental Conditions

- Vibration
- Duct and mount loads
- Sand, dust, humidity, corrosive fluids
- Fouling
- Temperature and pressure extremes
Manufacturing Considerations

• What quantities are involved?
  – 10’s, 100’s, 1,000’s, >10,000

• What are the available manufacturing processes for:
  – Details (fins, tubes, plates, bars, mounts, ports)
  – Core brazing, joining
  – Manifold forming and joining

• Design can be *pulled* from what can be built
• Design can *push* new manufacturing technology
Design Goals and Optimization

- Rank design variables with customer
  - Envelope, size
  - Interfaces
  - Weight
  - Durability
  - Heat transfer rate
  - Hot side pressure drop
  - Cold side pressure drop
  - Cost
- Select what is to be optimized, for example:
  - Minimize size and cost while meeting heat transfer and pressure drops
  - Maximize durability while meeting heat transfer and pressure drops
Design Process Summary

**Preliminary Design**
- Materials
- Configuration
- Surface Selection
- Sizing

**Detailed Analysis**
- Flow Distribution
- Temperature Distribution
- Interface Effects
- Conduction Effects
- Performance Mapping

**Requirements Review**
- Basic Physics
- Operating Conditions and Extremes
- Envelope and Interfaces
- Robustness
- Fluids
- Cost Targets
- Delivery

**Design Review**

**Manufacturing**

**Structures**

**Materials**

**Legacy Designs**

**Fluid Properties**

**Software Tools**

**Surface Properties**

**Component Tests**

**Literature**
Compact Heat Exchanger Design, Characteristics and Trends

4. Heat Exchanger Performance Analysis

NARSA Heavy Duty Heating and Cooling Conference
Sept 2012 Ann Arbor, MI

Instructor: Joe Borghese
Performance Analysis Overview

• Modes of heat transfer
• Heat transfer within a heat exchanger
• Conductance
• Heat capacity rate
• Impact of flow arrangement
• Estimating heat rejection and exit temperatures
• Pressure losses
Heat Transfer

• The transfer of energy in the form of heat
• Energy (heat) is always conserved
  – 1ˢᵗ law of thermodynamics
  – Heat given up by hot fluid = heat gained by cold fluid
• Heat flows from hot to cold
  – 2ⁿᵈ law of thermodynamics
  – Heat transfer rate is proportional to the temperature difference
Modes of Heat Transfer: Conduction

• Conduction through a medium
  – Solid, like aluminum or steel
  – Gas, like still air or water

\[ Q_{\text{conduction}} = \frac{k \times A}{l} \times (T_{\text{hot}} - T_{\text{cold}}) \]

- \( k \) = thermal conductivity
- \( A \) = cross sectional area for conduction
- \( l \) = conduction length through media

• Occurs in fins and tubes of heat exchangers
Modes of Heat Transfer: Convection

• From flowing fluid to a surface
  – Flow may be due to pump, fan, motion of vehicle or buoyancy driven
  – Convection coefficients determined by analysis for simple geometries or by test for most applications
    \[
    Q_{\text{convection}} = h \times A \times (T_{\text{hot}} - T_{\text{cold}})
    \]
    \[
    h = \text{convection coefficient}
    \]
    \[
    A = \text{surface area exposed to flow}
    \]

• Occurs from the fluid to the fins and tubes of heat exchangers

Originally suggested by Issac Newton in 1701
Modes of Heat Transfer: Radiation

• From one surface to another
  – Radiation in infrared wavelengths
  – Highly dependent on surface properties

\[ Q_{\text{radiation}} = A_1 \times F_{1-2} \times \sigma \times \left( T_1^4 - T_2^4 \right) \]

- \( A_1 \) = surface area of body 1
- \( F_{1-2} \) = factor to account for body 1 and 2 surface emittance and geometrical view from 1 to 2
- \( \sigma \) = Stefan - Boltzmann constant

• Generally small (ignored) in most heat exchanger applications
Heat Transfer within a Heat Exchanger

$T_{\text{hot\_in}} \quad \text{CONVECTION} \quad T_{\text{hot\_out}}$

$T_{\text{cold\_out}} \quad \text{CONDUCTION} \quad T_{\text{cold\_in}}$

$T_{\text{hot\_in}} \quad \text{CONVECTION} \quad T_{\text{hot\_out}}$

$T_{\text{cold\_out}} \quad \text{CONVECTION} \quad T_{\text{cold\_in}}$
Conductance

- The hot and cold fluids are connected by the conductance
- Conductance is used to calculate the heat transfer

CONDUCTANCE FROM HOT FLUID TO WALL
\( (h_{\text{hot}} \times A_{\text{hot}}) \)

CONDUCTANCE THROUGH WALL
\( (k_{\text{wall}} \times A_{\text{wall}} / \text{thickness}_{\text{wall}}) \)

CONDUCTANCE FROM WALL TO COLD FLUID
\( (h_{\text{cold}} \times A_{\text{cold}}) \)
Overall Conductance

• The three conductances can be combined to determine an overall conductance

\[ UA = \frac{1}{h_{\text{cold}} A_{\text{cold}}} + \frac{t_{\text{wall}}}{k_{\text{wall}} A_{\text{wall}}} + \frac{1}{h_{\text{hot}} A_{\text{hot}}} \]

• Overall conductance \((UA)\) relates the heat transfer to the hot to cold temperature difference
  – Higher conductance allows more heat transfer at lower temperature difference

\[ Q = UA \times \Delta T \]
Fluid Heat Capacity Rate

• Capacity rate is the ability of a flowing fluid to absorb heat

\[ CR = w \times C_p \]

\( w = \) fluid flow rate

\( C_p = \) fluid heat capacity

• Capacity rate relates the heat transfer to the temperature change of fluid

\[ Q_{\text{hot}} = w_{\text{hot}} \times C_p \times (T_{\text{hot\_in}} - T_{\text{hot\_out}}) \]

\[ Q_{\text{cold}} = w_{\text{cold}} \times C_p \times (T_{\text{cold\_out}} - T_{\text{cold\_in}}) \]

\[ Q_{\text{hot}} = Q_{\text{cold}} \]
Fluid Heat Capacity Rate Ratio

- Relationship between hot and cold side capacity rates determines temperature profiles in heat exchanger

\[
CR_{hot} \approx CR_{cold} \\
\frac{CR_{min}}{CR_{max}} \approx 1.0
\]

\[
CR_{hot} < CR_{cold} \\
\frac{CR_{min}}{CR_{max}} \approx 0.25
\]

\[
CR_{hot} >> CR_{cold} \\
\frac{CR_{min}}{CR_{max}} \approx 0
\]
Heat exchanger performance can be calculated as an efficiency or effectiveness

\[ \varepsilon = \text{heat exchanger effectiveness} \]

\[ \varepsilon = \frac{Q_{\text{actual}}}{Q_{\text{ideal}}} = \frac{CR_{\text{hot}} \times (T_{\text{hot} \_\text{in}} - T_{\text{hot} \_\text{out}})}{CR_{\text{min}} \times (T_{\text{hot} \_\text{in}} - T_{\text{cold} \_\text{in}})} \]

\[ \varepsilon = f(UA, CR_{\text{hot}}, CR_{\text{cold}}, \text{flow arrangement}) \]
Flow Arrangements

- Flow arrangement determines the order in which the hot fluid and cold fluid interact

PURE COUNTER FLOW

SINGLE PASS CROSS FLOW

PURE PARALLEL FLOW

TWO PASS CROSS COUNTER FLOW
Effectiveness-NTU Charts

The number of transfer units, $NTU \equiv \frac{UA}{CR_{min}}$

$NTU = \frac{UA}{CR_{min}} = \frac{HX \text{ ability to transfer heat}}{\text{Fluid's ability to absorb heat}}$

$\epsilon = f(NTU, CR_{hot}, CR_{cold}, \text{flow arrangement})$

From Ref. #2

Counter Flow
Best Thermal Perf, Complex Design

Parallel Flow
Poorest Thermal Perf

Define
Effectiveness-NTU Charts

SINGLE PASS CROSS FLOW
GOOD PERF,
SIMPLE DESIGN

MULTI PASS CROSS FLOW
BETTER PERF,
SIMPLE DESIGN

FROM REF. #2
Heat Exchanger Performance Example

- Engine coolant (PGW) cooled by air
- Keep hot coolant inlet conditions constant
- Vary air flow
- Calculate performance using eff-NTU chart
Heat Exchanger Performance Example

- From fluid conditions and HX geometry, calculate UA, CR_hot and CR_cold
- Calculate NTU from above
- Look up effectiveness from single pass crossflow NTU chart
- From effectiveness, calculate Tcold out, then Q and Thot out

<table>
<thead>
<tr>
<th>HOT SIDE, PGW</th>
<th>COLD SIDE, AIR</th>
<th>HEAT EXCHANGER PERFORMANCE</th>
</tr>
</thead>
<tbody>
<tr>
<td>GPM</td>
<td>Tin, F</td>
<td>Tout, F</td>
</tr>
<tr>
<td>48.9</td>
<td>225.0</td>
<td>206.1</td>
</tr>
<tr>
<td>49.0</td>
<td>225.0</td>
<td>213.3</td>
</tr>
<tr>
<td>49.1</td>
<td>225.0</td>
<td>218.3</td>
</tr>
</tbody>
</table>
Steps in Calculating HX Performance

• For the hot and cold side:
  1. From the geometry calculate the flow area, prime surface area, fin area and passage hydraulic diameter
  2. Look up the fluid properties: specific heat, thermal conductivity, viscosity
  3. Calculate the fluid Reynolds number
  4. Look up the Colburn j factor for the given surface at the Reynolds number
  5. Calculate the convection heat transfer coefficient from the j factor
  6. Calculate the fin efficiency and overall surface efficiency if a fin is used
  7. Calculate conductance for that side

• Calculate overall conductance (UA) and NTU
• Look up effectiveness for the given flow arrangement
• Calculate the outlet temperatures from the effectiveness
Heat Exchanger Pressure Losses

• Pressure loss breakdown:
  – Inlet duct to manifold
  – Contraction from manifold into core
  – Friction within core
  – Acceleration loss due to density change
  – Expansion from core into manifold
  – Manifold to outlet duct

• Want to keep duct losses to minimum since they don’t aid the primary objective of heat transfer
Pressure Loss Through HX

\[ P_{\text{total}} = P_{\text{static}} + P_{\text{dynamic}} \]

- \( P_{\text{total}} \) decreases due to shock losses.
- \( P_{\text{static}} \) increases as flow slows down in manifold.
- \( DP_{\text{static}} \) is greater than \( DP_{\text{total}} \) because the exit duct is smaller.
- Static pressure changes with changes in flow area and total pressure.
- Total pressure changes due to irreversible losses.

Frictional loss in core.
Total and Static Pressures

- $P_{\text{total}} = P_{\text{static}} + P_{\text{dynamic}} = P_{\text{static}} + \frac{1}{2} \rho V^2$
- Generally for liquids the difference between total and static is not very large
  - Due to high density, flow velocities are relatively lower
- For gases, the difference between total and static is usually measurable
  - Low density yields high flow velocities
  - Dynamic pressure is function of the square of the velocity
  - More of a concern with charge air coolers
- Typical PGW and air flow example:

<table>
<thead>
<tr>
<th>Fluid</th>
<th>GPM/CFM</th>
<th>lb/min</th>
<th>Duct Dia, in</th>
<th>$P_{\text{total}}, \text{psia}$</th>
<th>$V, \text{ft/s}$</th>
<th>$P_{\text{dynamic}}, \psi$</th>
<th>$P_{\text{static}}, \text{psia}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>PGW</td>
<td>50</td>
<td>408</td>
<td>3</td>
<td>35</td>
<td>2.27</td>
<td>0.0339</td>
<td>34.97</td>
</tr>
<tr>
<td>Air</td>
<td>6349</td>
<td>400</td>
<td>10</td>
<td>14.54</td>
<td>182.7</td>
<td>0.2408</td>
<td>14.30</td>
</tr>
</tbody>
</table>
Core Pressure Drop Calculation

\[ \Delta P = \frac{(w/A_c)^2}{2 \times g \times \rho_{avg}} \times \left\{ \frac{f \times L}{r_h} + K_c + K_e + 2 \times \left( \frac{\rho_{in}}{\rho_{out}} - 1 \right) \right\} \]

Where

- \( w \) = fluid mass flow
- \( A_c \) = core flow area
- \( g \) = gravitational constant
- \( \rho \) = fluid density
- \( f \) = friction factor (Fanning)
- \( L \) = flow length through core
- \( r_h \) = passage hydraulic radius \((D_h/4)\)
- \( K_c, K_e \) = Contraction and Expansion total pressure loss coefficients

- Core total pressure drop is based on the fluid dynamic pressure in the core
- Components are: Core friction, Inlet contraction and expansion losses, Flow acceleration
Fluid Pumping Power

• Energy required to move fluid through heat exchanger is proportional to the pressure drop

\[ P = \frac{w \times \Delta P}{\rho} \]

• Pumping power for air will be greater than for liquid (due to density differences)

• Want to minimize air side pressure losses
  – Large face area
  – Short flow length
  – Surface selection
Compact Heat Exchanger Design, Characteristics and Trends

5. Heat Exchanger Surfaces

NARSA Heavy Duty Heating and Cooling Conference
Sept 2012 Ann Arbor, MI

Instructor: Joe Borghese
Surface Classification and Selection

• Surface classification:
  – Prime or extended surface
  – Plain or enhanced surface

• Surface selected according to
  – HX type (tubular, bar plate, plate, etc.)
  – Pressure containment
  – Contamination
  – Performance and design optimization
Prime Surface Examples

- Plain tubes
- Turbulated tubes (using dimples or inserts)
- Flattened tubes
- Plates
- Corrugated plates

*Temperature difference from hot to cold is only in the separating surface*
Extended Surface Examples

- Finned tubes
- Plain strip fins
- Offset strip fins
- Louvered strip fins
- Wavy strip fins

Temperature difference from hot to cold is within the fins and the separating surface
Extended Surface (Fin) Efficiency

- Fins will have a temperature gradient from root to tip
- Fin area must be corrected for this gradient using a fin efficiency term

\[ \eta_{\text{fin}} = \frac{\text{actual fin heat transfer}}{\text{heat transfer if entire fin was at root temp}} \]

\[ \eta_{\text{fin}} = \tan\left( L_e \cdot \sqrt{\frac{2 \cdot h}{k \cdot t}} \right) \]

Where

\[ L_e = \text{effective length of fin} \]

- Fin efficiency increases with increasing \( k, t \); decreases with increasing \( h \)

For rectangular fin with adiabatic tip
Surface Performance

- All surface performance is characterized by two dimensionless groups:
  - Friction factor for pressure drop
  - Colburn j factor for heat transfer

\[
f = \Delta p \times \frac{r_h}{L}
\]

\[
j = St \times Pr^{2/3} = \frac{h}{\left(\frac{w}{A}\right) \times C_p} \times \left(\frac{C_p \times \mu}{k}\right)^{2/3}
\]

- Data is correlated using the flow Reynolds number

\[
Re = \left(\frac{w}{A_c}\right) \times D_h
\]

\[
\frac{\mu}{\text{viscous forces}} = \frac{\text{inertial forces}}{\text{viscous forces}}
\]

Where

\[
\mu = \text{fluid viscosity}
\]
Flow Regimes for Uninterrupted Channels

- **Laminar**
  - $Re < 2300$

- **Transition**
  - $2300 < Re < 10,000$

- **Turbulent**
  - $Re > 10,000$

FROM REF. #1
Circular Tube Heat Transfer and Flow Friction

- Uninterrupted channel shows definite transition region

FROM REF. #1
Surface Enhancement

- Fully developed flow is characterized by thicker boundary layers.
- There is more wall to bulk mixing as the boundary layer develops.
- Heat transfer is improved if boundary layer is continually re-developing.
- Many geometries are used to disturb boundary layer and improve heat transfer (dimples, louvers, offsets, waves, ...).
- Boundary layer disturbance increases pressure drop.

[Diagram of fluid flow in offset fin passage from Ref. #1]
Enhancement Effect on Finned Surfaces

- Compare plain, louvered and offset fins
- Plain has low f, low j
- Louver has higher j, higher f
- Offset has higher j, slightly higher f

DATA FROM REF. #2
Fin Selection Example

• Size a tube and center heat exchanger for the following conditions:
  – 50 GPM of PGW enters at 225 F
  – Cooled by 400 lb/min of air entering at 120 F
  – Must cool PGW to 213.5 F (80 kW)
  – Allow 0.5 psid on liquid side, 1.5 in H₂O on air side

• Size using the following surfaces:
  – Liquid side: plain flattened tube, finned flat tube
  – Air side: Plain fins (11 and 20 fins/in), Louvered fins (11 fins/in x ¼” spacing), offset fin (16 fins/in x 1/8” offset)
Fin Selection Sizing Results

- 6 designs generated, ALL have same performance
- Choose fins surfaces to optimize design
Compact Heat Exchanger Design, Characteristics and Trends

6. Engine Cooling Systems

NARSA Heavy Duty Heating and Cooling Conference
Sept 2012 Ann Arbor, MI

Instructor: Joe Borghese
Objectives of Engine Cooling System

• Maintain the highest and most efficient operating temperature within the engine.

• Bring the engine up to the operating temperature as quickly as possible in order to reduce the wear on the engine components and increase the fuel economy.
Engine Energy Balance

INPUT, GASOLINE 100%

RADIANT LOSS = 9%

COOLING SYSTEM RADIATOR LOSS 33% OF INPUT

OUTPUT = 25% OF INPUT

EXHAUST LOSS = 33% OF INPUT
Engine Coolant Flow Paths
Engine Operating Temperature

If the engine temperature is too high, various problems will occur:

• Overheating of lubricating oil causing it to breakdown
• Overheating of parts causing loss of strength
• Reduced clearance between engine parts causing increase in friction and resultant excessive wear.

If the engine temperature is too low, various problems will occur:

• Poor fuel mileage and power loss due to less efficient combustion process.
• Increased carbon buildup due to condensation of the fuel and excessive buildup on the intake valves.
• Increased varnish and sludge buildup within the lubrication system due to the cooler engine.
Sizing of Engine Cooling Components

• In order to design the engine cooling system, the following inputs are required:
  – Engine full load heat rejection to the coolant
  – Automatic transmission heat rejection to coolant
  – Engine oil cooler heat rejection to the coolant (if used)
  – Any other heat exchanger (e.g., condenser, intercooler, fuel cooler, etc.) heat transfer performance and pressure drop characteristics
  – Coolant pump performance, coolant loop pressure drop and pump power target
  – Fan performance and fan input power target
  – Ram airflow target and pressure drop from the air dam through the underhood airflow system.
Engine Coolant

- 50/50 mixture of ethylene glycol and water (EGW)
- The coolant provides protection against freezing (−34°F freezing point) and boiling (226°F boiling point at ambient pressure).
- Additives provide corrosion protection in the cooling system.
- Different specification coolants are used for aluminum versus cast iron engine and Cu-Br versus Al radiators.
The abscissa shows the water-glycol mixture with glycol concentration varying from 0 to 100% from left to right.
Propylene Glycol Water (PGW) Mixtures

- 50/50 mixture of propylene glycol and water provides freeze protection to -28 °F, boiling to 222 °F
  - Requires 60/40 mixture to achieve same freeze protection
- PGW viscosity is higher than EGW resulting in higher pumping power required
- Thermal conductivity is slightly lower but specific heat is about 5% higher
- Non-toxic

Using PGW may result in slightly higher pumping power and lower freeze/boil protection
But is Non-toxic
Air Flow Determination

- **Driving forces**
  - Ram air effect due to vehicle speed
  - Low pressure discharge areas (under vehicle)
  - Fans

- **Flow resistances**
  - Bumper, grille
  - Condenser, Radiator, Charge air cooler, oil coolers
  - Exit flow path(s) to ambient through engine compartment, upper and lower exits

- Air flow is set where pressure drop through the resistances equals the pressure rise through the drivers
Fan Drive Systems

Fan drive systems can be segmented into three types of fan drives for providing shaft power to the fan assembly.

- Engine Driven Fan Drives (up to 20+ kW)
- Electric Motor Fan Drives (up to 3 kW)
- Hydraulic Fluid Fan Drives (1.5 kW to 5 kW)
Fan Drive Systems

• The engine driven fan drive is the traditional means of providing power to the fan. Some innovations over the years have occurred including viscous coupling of the fan to the drive belt, molded plastic fan versus the stamped-metal fan, and more recently a move toward controlling the fan clutch electronically.

• Electric fan drives are the most common due to the ease of application, flexibility in mounting configuration, and ease of control. Various configurations have been applied with each having their particular benefits.

• Hydraulic fluid fan drive system consists of a hydraulic pump running off the engine that provides fluid power to a hydraulic motor that drives the fan(s). The advantage of this fan drive is the amount of power that can be delivered to a remotely mounted fan, 2.5 kW or more. This type of fan drive has been applied to some off highway vehicles.
Radiator Fan Systems

- **Puller Fan System**
  - Condenser
  - Radiator
  - Shroud
  - Airflow

- **Pusher Fan System**
  - Condenser
  - Radiator
  - Airflow

- **Center-Mounted Fan Drive System**
  - Condenser
  - Radiator
  - Shroud
  - Airflow
## Puller Fan Systems

### ADVANTAGES

- Heat exchangers act as flow straighteners to the puller fan providing more uniform inlet conditions to the fan blade set, thus permitting the fan to operate at a higher efficiency.
- Using additional ducting, puller fans can also be used to draw air from engine compartment components or to direct the warm air off from the fan to provide some cooling of underhood components. Toyota and Volvo have used puller fans to draw air through battery and electronics cool boxes.
- Puller fans are generally well protected for debris fouling the fan and preventing the fan from operating.

### DISADVANTAGES

- The puller fan operates at the highest air temperature in the cooling system. The higher temperature reduces the mass flow rate that the fan can move since a fan is a volumetric flow device. Also these high temperatures reduce the durability of the fan motor and/or increase the cost of the motor and motor controllers.
- The high ambient temperatures also increase the cost of materials for the fan, the shroud, and the motor.
- Shroud and motor durability may be affected by exhaust manifold heat radiation or may require additional heat shielding on the motor and shroud. This issue is even becoming more severe due to the trend toward close-coupled catalysts to the exhaust manifold in the underhood compartment.
## Pusher Fan Systems

### ADVANTAGES

- Fan operates in near-ambient conditions, which improves the fan durability, and increase the mass flow rate moving capability of the pusher fan.
- Fans are generally easy to service in this location.
- Pusher fan can be designed and can operate at nearly the same total system efficiencies as puller fans. When designed with a full-coverage shroud, reasonable flow distribution can be realized over the heat exchangers.

### DISADVANTAGES

- The major disadvantage of pusher fans is the ease of fouling/damage caused by debris and snow and ice.
- Airflow distribution on the heat exchanger cores is also an issue. The lack of ideal diffusion to the condenser results in reduced airflow and nonuniform airflow to heat exchangers, thus limiting heat transfer performance and resulting in higher airside pressure drop.
- A pusher fan results in part of the flow from the condenser bypassing the radiator or requires a higher level of air path sealing (ducting) between the fan, condenser and radiator.
- A pusher fan tends to recirculate more cooling air at idle since the exiting airflow from the cooling module lacks momentum (both speed and direction).
Center Mounted Fan Systems

**ADVANTAGES**

- A CMF produces less noise because its center-mounted location permits the heat exchangers to act as sound dampers.
- The condenser acts as a flow straightener to the center-mounted fan permitting the fan to operate at a higher efficiency.
- Center-mounted fans are generally well protected from fouling or damage by debris.
- Due to the radiator being behind, the CMF is also well shielded from exhaust manifold and any close-coupled catalyst heat radiation.
- The CMF can provide thermal management functions to other underhood components.
- The center-mounted fan may be able to be designed more efficiently than any other system since both the inlet-flow and the outlet-flow conditions to the fan are controlled.

**DISADVANTAGES**

- The CMF takes a longer axial, fore-aft, dimension than either the puller or pusher fan systems due to the additional clearance required between the motor(s) and the heat exchangers.
- The radiator airflow distribution may be an issue without the proper fan and shroud design. Since the fans act as a pusher fan onto the radiator, the same airflow distribution issues are present as with pusher fans.
- A CMF, as do pushers, tends to recirculate more cooling airflow at idle since the exiting airflow from the cooling module lacks momentum (both speed and direction).
Electric Assist Pusher Fans

Electric Assist Pusher Fan

- A single or dual electric pusher fan(s) can be added to assist the engine driven fan system at low vehicle speeds and severe ambient conditions.
- These fans have generally lower power levels than an all-electric cooling system.
- The amount of idle airflow recirculation can be increased (or at least not improved) when this fan type is applied to a vehicle.

Applications

- Current applications include both cars and trucks where additional cooling is required. Motor applications include both the standard brush type and a brushless DC motors

Assist Pusher Fan System
Crossflow vs. Downflow Radiators

Crossflow Radiators

Downflow Radiators
Crossflow Radiators

ADVANTAGES

• Fewer parts, manufacturing advantage, minimum tooling investment.
• Fewer joints, inherently fewer leak paths.
• Less wet weight, shorter tanks, less coolant volume.
• More flexibility to change face area by width change.
• Typically 10-15% more face area for a given size.
• Can have oil coolers in both tanks.
• Will have slightly higher performance if the center height, core constant and core depth are kept the same.

DISADVANTAGES

• Due to longer tubes, the brazing process is not as forgiving as for the downflow radiator and need to cut the core reinforcement for thermal stress relief.
• Higher coolant pressure drop.
• Wide cores (>700 mm) with dual fans may need stabilization to the core reinforcement.
• Less plumbing flexibility than that for a typical downflow radiator.
• Less drawdown deaeration protection than a typical downflow radiator.
Downflow Radiators

**ADVANTAGES**
- Design flexibility in inlet and outlet fitting locations and shroud/fan mounting features.
- Possibly better deaeration.
- Saw cuts are typically not required for shallow cores with reinforcement lengths less than 425 mm.
- Attachment of the fan to the tank easier in downflow because of short moment arms; legs needed in crossflow.
- Reduced coolant pressure drop.

**DISADVANTAGES**
- Higher material cost due to increased parts count.
- About 30% higher assembly time needed due to increased parts.
- Must retool header, gasket and tanks to change the core width.
- Cannot install an oil cooler in the upper tank because it is not always submerged in the coolant.
- Long tanks result in poor coolant distribution at low flow rates.
Oil Coolers

• Oil coolers used to maintain desired oil temperatures
  – Gasoline engine oil sump ~285F
  – Diesel engine oil sump ~265F
  – Transmission oil ~285F

• Common to have transmission oil coolers in radiator tank

• Air-oil coolers may be added for transmission, power steering and engine oil

• Low duty coolers may be plain or finned tubes

• Higher performance coolers will use louvered fins in bar-plate or tube-center configuration
• Engine output is increased (relative to its size) using compressed (boosted) air charge
• Compressing air raises its temperature and lowers its density
• Charge air cooler increases the charge air density thus improving output
• Also aids in reducing NOX with lower combustion temperature
• Pressure drop of CAC causes slight reduction of boost
Types of Charge Air Coolers

- Use engine coolant
- Air outlet temperature limited by engine coolant temperature (~200°F)

- Use ambient air
- Air outlet temperature limited by air temperature (120-140°F)
Manifold Design in Air to Air Charge Air Coolers

- Manifold (tank) design is not as critical in coolant or oil manifolds because velocity pressure is low in most liquid applications.
- If liquid tanks are large enough the static pressure change in the tank is minimal and the flow will distribute evenly.
- For air flowing in compact manifolds, the static pressure change in the manifolds may give rise to non-uniform flow distribution and negatively impact performance.
Manifold Performance

U-Flow
- Static pressure rises in inlet tank as flow decreases
- Pressure difference across core more uniform
- Static pressure decreases in outlet tank as flow increases

Z-Flow
- Pressure difference across core very non-uniform

U-FLOW CONFIGURATION PREFERRED FOR BETTER FLOW DISTRIBUTION

FROM REF. #1
Manifold Design

• **U-Flow**
  – use convex tapered inlet manifold
  – constant area outlet manifold
  – The larger the outlet the better

• **Z-Flow**
  – use concave tapered inlet manifold
  – constant area outlet manifold
  – More difficult to get Z-flow manifolds working properly since flow area wants to go to zero at “dead-end”
Compact Heat Exchanger Design, Characteristics and Trends

7. Auto Air Conditioning Systems

NARSA Heavy Duty Heating and Cooling Conference
Sept 2012 Ann Arbor, MI

Instructor: Joe Borghese
**Systems Overview**

- **Orifice Tube**
  - Less expensive fixed orifice
  - Accumulator/drier protects compressor and stores refrigerant

- **Expansion valve (TXV)**
  - More complex TXV senses and controls evaporator superheat (5F) to protect compressor
  - Receiver stores refrigerant and ensures no vapor to TXV
Vapor Compression Cycle with R134a

Diagram showing the components of a vapor compression cycle with R134a:
- Compressor
- Condenser
- Expansion Valve
- Evaporator
- dP of AD (if used)
Condenser Heat Exchanger

• Rejects refrigeration system heat to air
• Total heat rejection = evaporator heat (~60%) plus compressor work (~40%)
• Condenser adds heat load and air pressure loss to engine cooling heat exchanger
• Due to R134a system pressure levels want the condenser at the coolest location in the air heat sink
• Reduce size by optimizing air side surface
  – Condensing heat transfer coefficients can be 25 times air side
• Low refrigerant pressure drop maintains air-refrigerant temperature difference
Condenser Development

Technology Trend Over Time
Modern Folded Flow Condenser Design

- Refrigerant flow is folded across the air flow.
- Last "fold" or "pass" is mostly/all liquid so has fewer passages.
- Extruded tubes contain high pressure new shorter passage heights and more fins, increasing heat transfer.
Compact Heat Exchanger Design, Characteristics and Trends

8. Recent Developments and Concluding Remarks

NARSA Heavy Duty Heating and Cooling Conference
Sept 2012 Ann Arbor, MI

Instructor: Joe Borghese
Multi Louvered Fin

- Louvered fins are preferred for balance of heat transfer enhancement, pressure drop and cost
- Louvers are being refined
  - More louvers in the flow direction
  - Longer louver cut in the fin height direction
  - Both triangular and rectangular forms being used
Tubeside Enhancement

- In order to enhance the tube side performance in the Reynolds number range of 1000-4000, the tube side augmentation is being used in some applications. This enhancement on the tube wall is in the form of bumps, interrupted or continuous transverse ribs to the flow direction, or a turbulator inside the tubes.

Bumped Tube
Unified Condenser and Radiator

- **Description:**
  - Combine radiator and condenser
  - Process/manufacturing of single heat exchanger

- **Benefits:**
  - Reduced assembly & brazing cost (10%)
  - Eliminate mounting brackets
  - Reduced Weight (10%)
  - Improved Airflow Management
  - Improved Packaging
  - Improved Recyclability

**Current Type**

Radiator: 414x480x29, fp1.0 mm
Condenser: 373x508x16 fp1.3 mm

**UCR**

UCR: 393.3x480x36, fp1.3 mm
Advanced Systems

• Hybrid gas/electric systems require power electronics cooling
  – Inverter coolant loops added
  – Offset by smaller gas engine radiator

• Fuel cell systems require fuel cell stack and power electronics cooling

• Advanced gas engine systems will put thermostat under control of engine control unit

• Thermal storage heat exchangers being considered for reduced start up emissions
Systems Consideration in Design

• Combination of engine, A/C, electronics, charge air, transmission, oil cooling along with vehicle aerodynamics and air fans require a *SYSTEM* approach to component design

• Accurate component models within high level system model are required in order to trade heat exchanger packaging, NTU, and pressure drop with air flow system
Concluding Remarks

• Current and advanced automotive systems will continue to require cooling

• High performance, compact heat exchangers can be optimized given a range of well designed heat transfer surfaces

• The greatest gains in weight or size savings can be made when considering all cooling requirements in a thermal management system