Comparison of Experimental and Analytical Substation Bus Temperatures

Presented by: Joe Goldenburg, Tony Pribble
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Who we are

• Joe Goldenburg
• Leads GA Tech NEETRAC’s Mechanical Section
• BME GA Tech
• MSOR GA Tech
• Licensed Engineer, Georgia
• 26 years experience
• Chair ANSI C119.0 OH Connectors
• Member ASTM B01 OH Conductors
• Member Overhead Lines Committee
Who we are

- Tony Pribble
- Research Engineer – NEETRAC Lab
- BSME Iowa State University
- MSE-MS Georgia Tech – In progress
- EIT
NEETRAC Overview

• Non-profit, 37 member industry consortium at Georgia Tech
  – 37 Members – Utilities and Manufacturers
  – Represent ~65% of US + Canadian electric customers
• Provide third-party testing and application research services.
NEETRAC Overview
Agenda

1. History of the project
2. Heat transfer refresher
   – Convection
   – Radiation
3. Three methods
   – 605
   – CFD
   – Experiments
Agenda

4. How 605 works
5. How CFD works
6. How the experiments work
7. Results
8. Discussion / question
9. Lessons learned
History of the Project

• In 2012, NEETRAC members funded a project to develop a bus ampacity software package.
  – Automates IEEE 605 with some slight modifications.
• CFD simulations were run to determine the steady state bus temperature.
  – Predicted values from 605 were used to verify the results.
• CFD and 605 didn’t match in some cases.
  – 605 made several simplistic assumptions not made in CFD.
• A wind tunnel was built and experimental data collected.
History of the Project

• The temperature differences were confirmed in experiments.
• NEETRAC members requested we share the data with the 605 committee.
• This presentation is meant to inform the committee of available data and resources.
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Steady-State Heat Transfer

\[ Q_{\text{resistance}} + Q_{\text{sun}} = Q_{\text{convection}} + Q_{\text{radiation}} \]
Steady-State Heat Transfer

\[ I^2R + Q_{\text{sun}} = hA\Delta T + \varepsilon\sigma A(T^4 - T_{\text{sur}}^4) \]

Ignore

\[ Q_{\text{resistance}} \]

\[ Q_{\text{convection}} \]

\[ Q_{\text{radiation}} \]
Forced Convection

\[ Q_c = hA(T_{air} - T_w) \]

Convection Coefficient
[w/m²K]
Forced Convection

\[ Q_{c1} = hA(T_{air} - T_w) \]

\[ Q_{c2} = hA(T_{air} - T_w) \]

\( h \) is different
Forced Convection

The convection coefficient, $h$, is affected by:

- Geometry
- Fluid properties
- Fluid velocity
Forced Convection

• How is $h$ determined?
• The Nusselt number is a dimensionless quantity used to find $h$.

$$ Nu = \frac{hL}{k} $$

- Convection Coefficient
- Characteristic Length
- Thermal Conductivity of the Fluid
Forced Convection

Experiments have shown a relationship between the Nusselt, Reynolds, and Prandtl numbers to find \( h \).

\[
Nu = \frac{hL}{k} = C(Re^m)Pr^{0.33}
\]

Fluid Velocity

\( Re = \frac{VL}{\nu} \)

Characteristic Length

Kinematic Viscosity

Prandtl – Typically taken as 0.7 for air
Forced Convection

\[ T_{\text{air}}, V_{\text{air}} \quad Q_c \]

If \( T_{\text{air}} < T_w \)

\[ Nu = \frac{hL}{k} = 0.664(Re^{0.5})Pr^{0.33} \]

*Flat Plate Nusselt Correlation*
Forced Convection

\[
Nu = \frac{hL}{k} = 0.3 + \frac{0.62 Re^{1/2} Pr^{1/3}}{2} \left( 1 + \left( \frac{Re}{282000} \right)^{8/5} \right)^{4/5} \left( 1 + \left( \frac{0.4}{Pr} \right)^{3/2} \right)^{1/4}
\]

Round Body Nusselt Correlation
Steady-State Heat Transfer

\[ I^2R = hA\Delta T + \varepsilon\sigma A(T^4 - T_{sur}^4) \]
Radiation

Radiation is the emission of energy through EM waves; In substations, this happens at infrared wavelengths.

\[ Q_{rad} = \varepsilon \sigma A (T^4 - T_{sur}^4) \]

- Emissivity
- Stefan-Boltzmann Constant
Steady-State Heat Transfer

\[ Q_{\text{resistance}} = Q_{\text{convection}} + Q_{\text{radiation}} \]
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Heat Transfer Solution Methods

Three methods to solve the steady-state heat transfer problem.

• IEEE 605
• Experimental
• CFD/FEA

\[ Nu = \frac{hL}{k} = 0.664(Re^{0.5})Pr^{0.33} \]

IEEE 605

CFD/FEA

Experimental
IEEE 605 Calculation

IEEE 605 uses the Nusselt correlation for a flat plate to find $h$.

– All bus geometries (except round) are broken into a series of horizontal flat plates.

– Disregards orientation

$$Nu = \frac{hL}{k} = 0.664(Re^{0.5})Pr^{0.33}$$

*Nusselt correlation for a flat plate*
IEEE 605 Calculation

Flow Direction

A  B  C

D

=+

Flow Direction

A  B  C

D

+=
IEEE 605 Calculation

Flow Direction

A

B

C

D

= B

+ D

A

+ C
IEEE 605 Calculation

A

B

C

D

A

B

D

C

No symbol
IEEE 605 Calculation

\[ Nu = \frac{hL}{k} = 0.3 + \frac{0.62 \text{Re}^{1/2} \text{Pr}^{1/3}}{(1 + \left(\frac{0.4}{\text{Pr}}\right)^{2/3} \left(1 + \left(\frac{\text{Re}}{282000}\right)^{5/8}\right)^{4/5}} \]

**Round Body Nusselt Correlation**
IEEE 605 Calculation

Flow Direction
IEEE 605 Calculation

These regions are approximated by free (natural) convection.
IEEE 605 Calculation

Force Convection

Natural Convection – Horizontal Surfaces

Notice these are all downward facing surfaces.
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Steady-State Heat Transfer

\[ Q_{\text{resistance}} = Q_{\text{convection}} + Q_{\text{radiation}} \]
CFD Simulation

Flow Direction
CFD Simulation

Fluid Velocity

Bus Temperature
CFD Simulation

Swirling & Recirculation
CFD Simulation

Un-even heat flux
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Experimental
Wind Tunnel Anatomy

Flow Straightener

Compression Zone

Sample Chamber

Expansion Zone

Bus
Wind Tunnel Anatomy

Sample Chamber

Expansion Zone

Fan
Experimental

• We assume:
  – Heat conduction from the bus is negligible.
  – Surface-to-surface radiation model.
  – Input air flow is uniformly laminar.

• Wind tunnel data and CFD results are compared for agreement.
  – This helps us check our assumptions and verify the results.
# Experimental

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<thead>
<tr>
<th>Geometry</th>
<th>Wind Speed Targets</th>
<th>Temperature Targets</th>
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<tbody>
<tr>
<td>Rectangular</td>
<td>0.61 m/s</td>
<td>70 C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>110 C</td>
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<td>190 C</td>
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<td>Rectangular Vertical</td>
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<th>Geometry</th>
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In Progress
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Results

• On the proceeding slides, the steady state bus temperature is displayed by method (605, CFD, Experimental)

• Experiments were run first. From these runs, input current was recorded. This current was then used in CFD & 605 to determine the bus temperature.

• Nussult correlations were developed from this data.
Horizontal Rectangular Results

*Each point represents the steady state temperature determined by the respective method.

22°C difference between 605 & exp.
Vertical Rectangular Results

17 C difference between 605 & exp.
UAB Closed Results

31°C difference between 605 & exp.

0.6 m/s

1 m/s

2 m/s

Method
- CFD
- Experimental
- IEEE 605
UAB Open Results

33 °C difference between 605 & exp.
IWBC Results

In progress
Experimental Results

• For vertical rectangular, IEEE 605 is conservative.
  – Translates to lost ampacity

• For horizontal rectangular, closed UAB, open UAB, and IWBC IEEE 605 is aggressive.
  – Predicts lower steady-state temperature than found in experiments
Survey

• NEETRAC sent out a survey to determine the popularity of each style bus.
Total Number of Substations

(3 Responses)
Average Percentage of Bus Used

(3 Responses)

- Round: 64%
- Rectangular: 14%
- Universal Angle: 17%
- Integral Web: 5%
Flat Plate Correlation

• Advantages of flat plate correlations:
  – Can be built up to approximate complex geometries
  – Inexpensive
  – Computationally simplistic

• Disadvantages of flat plate correlations:
  – Inaccurate, upwards of 20% error
Whole Body Correlation

• Advantages of whole body correlations
  – More accurate
  – Computationally less complex (one formula)

• Disadvantages of whole body correlations
  – Only good for specific geometry, orientation, and Reynolds number range.

• Given the limited number of geometries, NEETRAC recommends using whole body correlations, as is currently done with round.
Correlations

Rectangular - Horizontal
\[ \ln(\frac{Nu}{Pr^{0.33}}) = -1.834 + 0.6553 \ln(Re) \]

\[ Nu = \frac{hL}{k} = c(Re^m)Pr^{0.33} \]

\[ Nu = \frac{hL}{k} = 0.160(Re^{0.655})Pr^{0.33} \]

\[ S = 0.0572904 \]
\[ R-Sq = 97.7\% \]
\[ R-Sq(adj) = 97.6\% \]
Correlations

Rectangular - Vertical

\[ \ln(\frac{Nu}{Pr^{0.33}}) = -0.5852 + 0.5504 \ln(Re) \]

\[ Nu = \frac{hL}{k} = 0.557(Re^{0.55})Pr^{0.33} \]

- **Regression**
- **95% CI**

- **S** 0.0982542
- **R-Sq** 90.8%
- **R-Sq(adj)** 90.4%
Correlations

UAB Closed

\[ \ln\left(\frac{Nu}{Pr^{0.33}}\right) = -2.354 + 0.7121 \ln(Re) \]

\[ Nu = \frac{hL}{k} = 0.079(Re^{0.71})Pr^{0.33} \]

\[ S = 0.0641092 \]

\[ R-Sq = 97.4\% \]

\[ R-Sq(adj) = 97.3\% \]
Correlations

UAB - Open

$$\ln\left(\frac{Nu}{Pr^{0.33}}\right) = -2.523 + 0.7154 \ln(Re)$$

$$Nu = \frac{hL}{k} = 0.080(Re^{0.72})Pr^{0.33}$$

- Regression
- 95% CI

S 0.0729692
R-Sq 96.9%
R-Sq(adj) 96.7%
Correlations

In progress
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Recommendations

• IEEE 605 move to whole body correlations.
• NEETRAC member have agreed to provide our data to the substation committee.
• NEETRAC members have funded our participation in the next revision of 605.
Questions?
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Radiation

• The below equation assumes radiation from the body is lost to the surroundings.
• This is not always the case.

\[ Q_{rad} = \varepsilon \sigma A (T^4 - T_{sur}^4) \]
Radiation

$$Q_{r_2} = \varepsilon_2 \sigma A_2 (T_2^4)$$

$$Q_{r_1} = \varepsilon_1 \sigma A_1 (T_1^4)$$

How much radiation is “seen” by the other surface is determined by it’s view factor.
Radiation

• The view-factor is the percentage of energy emitted by one surface that irradiates another.

• This is known as a *Surface-to-Surface* model.

• Modern day simulation software can calculate view factors for complex geometries.
  
  – Not available during the creation of 605

\[ Q_{r,1-2} = \varepsilon_1 \sigma T_1^4 F_{1-2} - \varepsilon_2 \sigma T_2^4 F_{2-1} \]
Changing $\varepsilon$ from 0.3 to 0.5 results in about $10^\circ$C change
(As determined in CFD simulations)