# Improving Motor Efficiency and Motor Miniaturisation
# The Role of Thermal Simulation


**Dr David Staton**

**Motor Design Ltd**

**dave.staton@motor-design.com**


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# Todays Topics

- Motor Design Ltd (MDL)
- Need for improved tools for thermal analysis of electric machines
- Important Issues in Thermal Analysis of Electric Motors
- Examples of use of thermal analysis to optimise electric machine designs

# Motor Design Ltd

- Based in Ellesmere, Shropshire, UK
  - On England/Wales border
  - South of Chester and Liverpool
- MDL Team:
  - Dave Staton (Software Development & Consultancy)
  - Mircea Popescu (Consultancy)
  - Douglas Hawkins (Software Development & Consultancy)
  - Gyula Vainel (Motor Design Engineer)
  - Lyndon Evans (Software Development)
  - James Goss (EngD – Motor-LAB)
  - Lilo Bluhm (Office Manager)
- Many University Links:
  - Sponsor 3 Students in UK at present
  - Many links with universities throughout world
    - Bristol, City, Edinburgh, Mondragon Sheffield, Torino, ...


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# Dave Staton

- Apprentice/Electrician - Coal Mining Industry (1977 - 1984)
  - BSc in Electrical Engineering at Trent Polytechnic (Nottingham)
- PhD at University of Sheffield (1984 - 1988)
  - CAD of Permanent Magnet DC Motors (with GEC small machines)
  - 95% electromagnetic aspects and less than 5% on thermal aspects
- Design Engineer - Thorn-EMI CRL (1988-1989)
  - design of electric motors for Kenwood range of food processors
- Research Fellow - SPEED Laboratories (1989 - 1995)
  - help develop SPEED electric motor design software
  - predominantly on electromagnetic aspects
- Control Techniques (part of Emerson Electric) (1995 - 1998)
  - design of servo motors
  - More involved in thermal analysis as we were developing radically new motor constructions (segmented laminations) that we had no previous experience.
- Set up Motor Design Ltd in 1998 to develop heat transfer software for electric machine simulation
  - there was no commercial software for thermal analysis of motors
  - such analysis was becoming more important in the design process (volume/weight minimisation, energy efficiency, etc.)
- Given many courses on motor design and thermal analysis of electric machines worldwide

# Motor Design Ltd (MDL)

- set up in 1998 to develop software for design of electric motors and provide motor design consulting and training
- distribute **SPEED**, **Motor-CAD**, **FLUX** and **PORTUNUS** software
  - complete package for electric motor and drive simulation
  - software package also used in our consulting work which helps with development

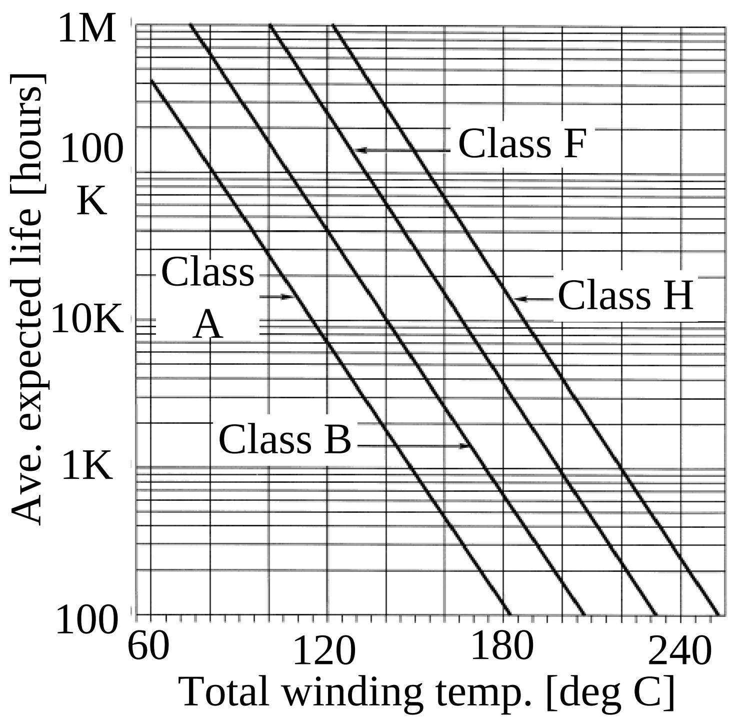
# Todays Topics

- Motor Design Ltd (MDL)
- **Need for improved tools for thermal analysis of electric machines**
- Important Issues in Thermal Analysis of Electric Motors
- Examples of use of thermal analysis to optimise electric machine designs

# Thermal and Electromagnetic Design

- Traditionally in electric motors design the thermal design has received much less attention than the electromagnetic design
  - electric motor designers tend to have an electrical engineering background rather than mechanical?
  - CAD tools for thermal design have tended to be very specialized and require extensive knowledge of heat transfer
    - MDL have developed Motor-CAD + Portunus thermal/flow library to make it easier to carry out thermal analysis without being a thermal expert
    - FEA and CFD software are starting to have features included to make it easier to set up electric motor thermal models
- There is a similar situation in electronics cooling
  - Electronics designers tend to have an electronics background with little knowledge of heat transfer

# Need for thermal analysis of electric motors




- Temperature of motor is ultimate limit on performance and should be given equal importance to the electromagnetic design
  - Life of motor is dependent upon the winding temperature

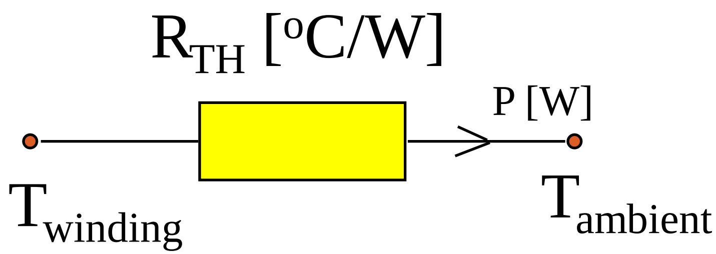
# Need for thermal analysis of electric motors

- There is a requirement for smaller, cheaper and more efficient motors so an optimised design is required
  - Losses depend on temperature and temperature on losses
- Simple sizing method based on such inputs as limiting winding current density are no good for optimisation (see next few slides)
  - depend on users experience for the manufacturing process and material used and so tend to become very inaccurate when trying something new in the design
  - Give the designer no indication of where to concentrate design effort to reduce temperature
- Mix of analytical and numerical methods required
  - Many applications do not have a steady-state operation so in order to obtain a reasonable transient calculation time lumped circuit techniques are required
  - CFD/FEA are useful to help set up accurate models
    - Part models best. e.g. heat transfer through winding, fan model, etc.

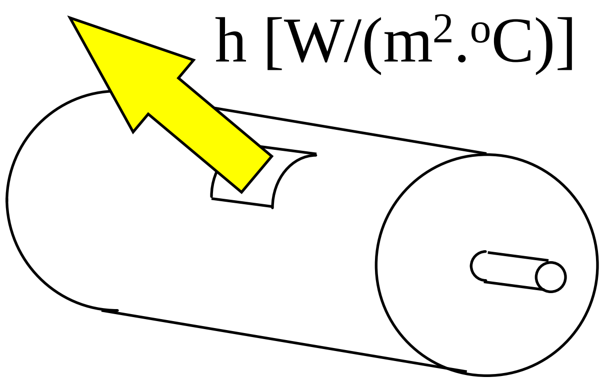
# Traditional Motor Sizing Methods

- sizing based on single parameter
  - thermal resistance
  - housing heat transfer coefficient
  - winding current density
  - specific electric loading
- thermal data from
  - simple rules of thumb
    - 5 A/mm2, 12 W/m2/C etc.
  - tests on existing motors
  - competitor catalogue data
- can be inaccurate
  - single parameter fails to describe complex nature of motor cooling
- poor insight of where to concentrate design effort
- Alternatives are analytical lumped circuit thermal analysis and numerical thermal analysis

Thermal Resistance




Heat Transfer Coefficient



# Typical Rules of Thumb

| Condition | A//mm$^2$ | a/in$^2$ |
|---|---|---|
| Totally enclosed | 1.5 – 5 | 1000 – 3000 |
| Air-over; fan-cooled | 5 – 10 | 3000 – 6000 |
| Liquid cooled | 10 – 30 | 6000 - 20000 |

- Air Natural Convection
  - $h = 5\text{-}10\ W/(m^2.C)$
- Air Forced Convection
  - $h = 10\text{-}300\ W/(m^2.C)$
- Liquid Forced Convection
  - $h = 50\text{-}20000\ W/(m^2.C)$

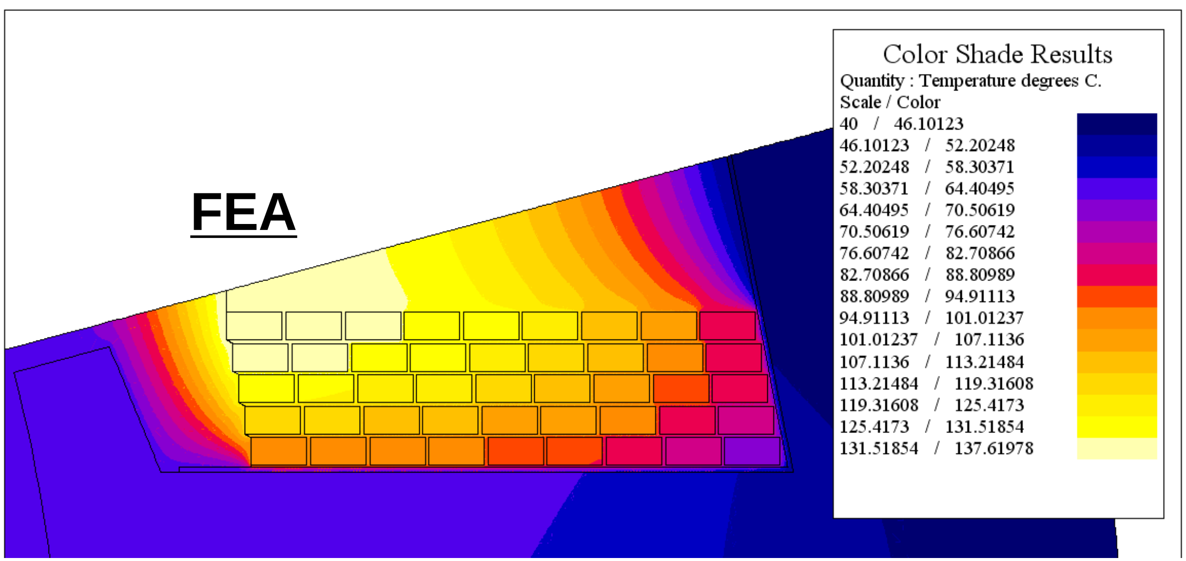
| Class of machine | TRV [kNm/m$^3$] | σ [lbf/in$^2$] |
|---|---|---|
| Small totally-enclosed motors (Ferrite magnets) | 7 – 14 | 0.5 – 1 |
| Totally-enclosed motors (sintered Rare Earth or NdFeB) | 14 – 42 | 1 – 3 |
| Totally-enclosed motors (bonded NdFeB magnets) | 20 | 1.5 |
| Integral-hp industrial motors | 7 – 30 | 0.5 – 2 |
| High-performance servomotors | 15 – 50 | 2 – 4 |
| Aerospace machines | 30 – 75 | 2 – 5 |
| Large liquid-cooled machines (e.g. turbine-generators) | 100 – 250 | 7 – 18 |

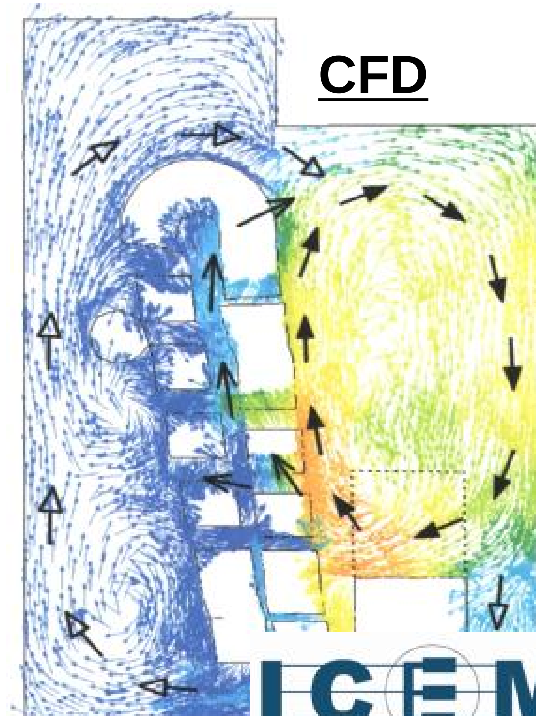
- Wide range of possible values makes past experience very important
  - Otherwise the design will not be correctly sized
  - Values may not be valid if change manufacturing process, material, etc.
  - Tables taken from: “SPEED Electric Motors”, TJE Miller

# Numerical Thermal Analysis

- Two basic types available that subdivide problem into small element or volumes and temp/flow solved:
  - Finite element analysis (FEA)
    - Useful to accurately calculate conduction  heat transfer
  - Computational fluid dynamics (CFD)
    - Simulation of fluid flow involves the solution of a set of coupled, non-linear, second order, partial differential equations – conservation equations (velocity, pressure & temperature)
    - Many of the Fluent CFD slides are from the

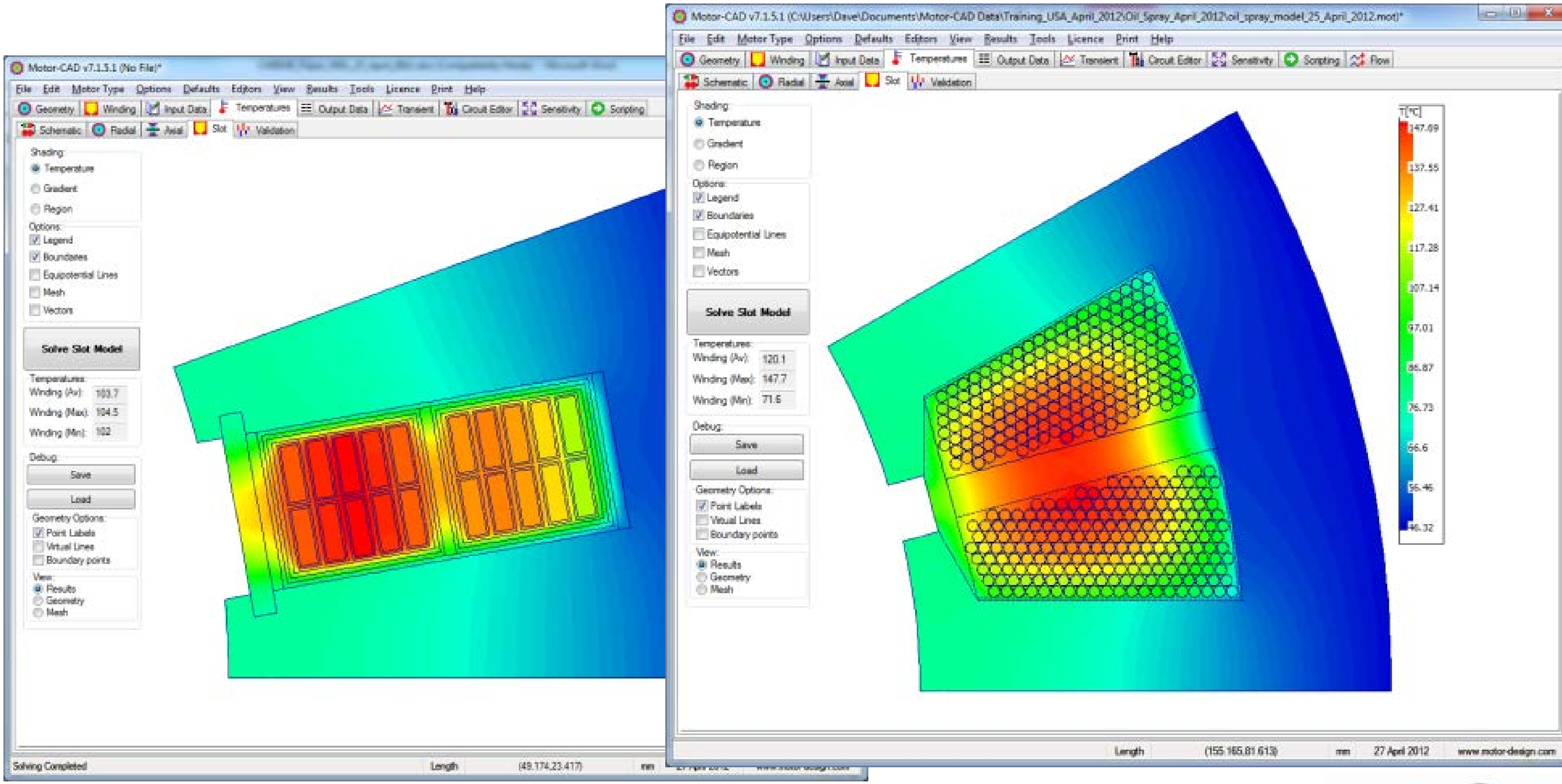
      Example from University of Nottingham



# Motor-CAD Integrated Thermal FEA Solver

- Just a few seconds to generate a mesh and calculate
- Help improve accuracy through calibration of analytical model

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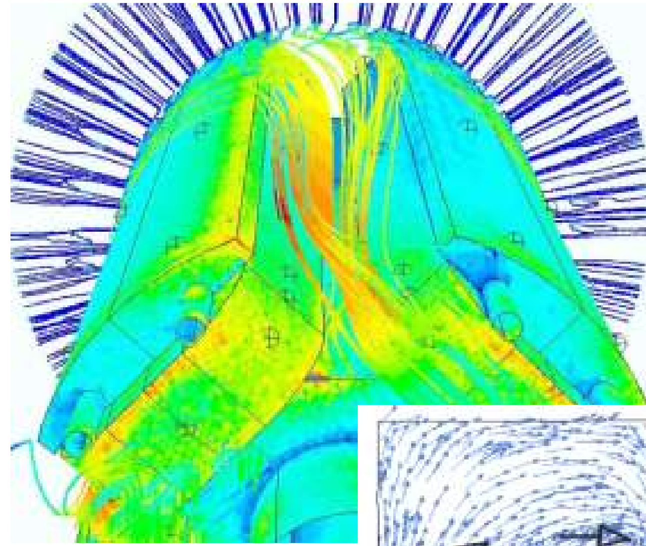
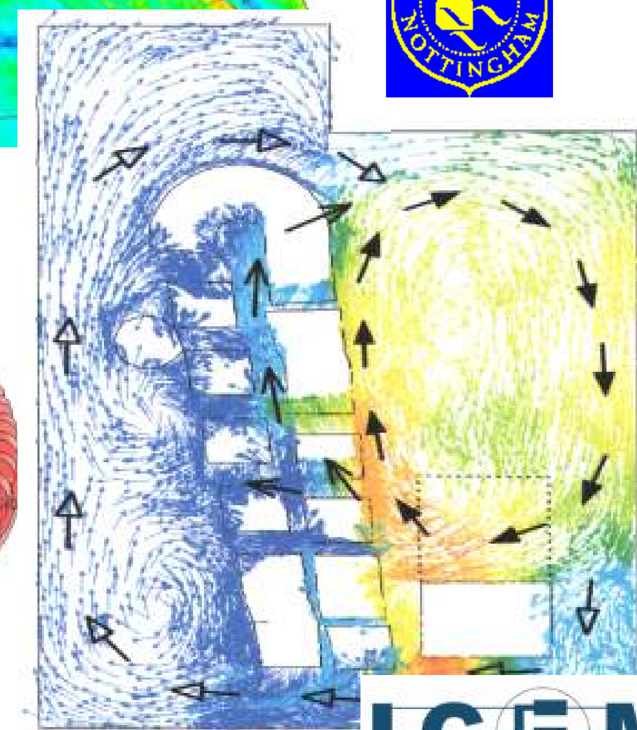
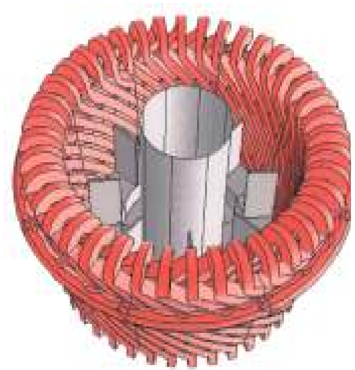
# Computational Fluid Dynamics (CFD)

- The expected accuracy is not as great as with electromagnetic FEA
  - Due to complexities of geometry and turbulent fluid flow
- Impossible to model actual geometry perfectly
  - But trend predictions and visualisation of flow are useful
- Can be very time consuming to construct a model and then calculate
  - Can be several weeks/months
- Best use of results to calibrate analytical formulations







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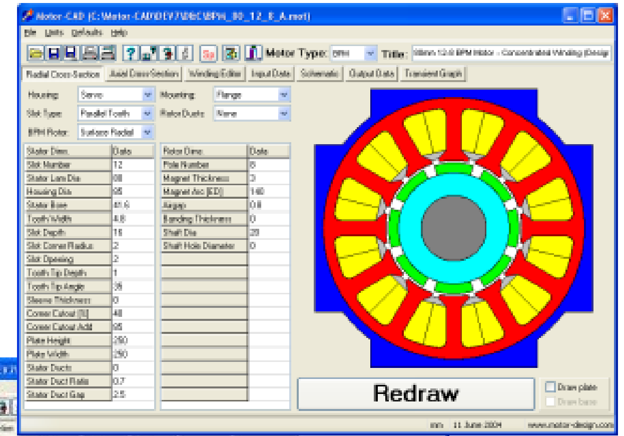
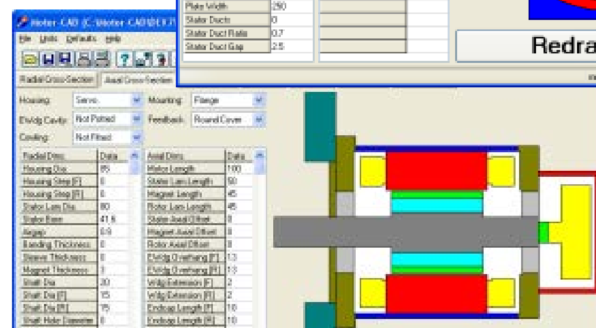
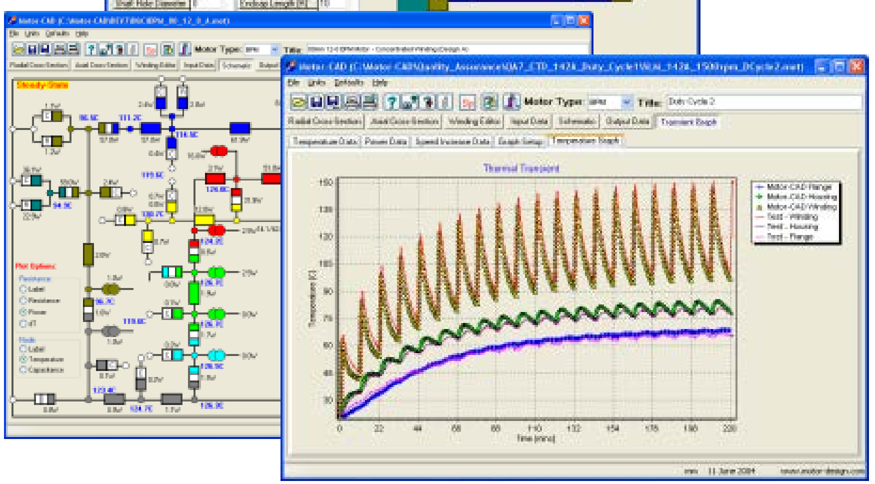
# Motor-CAD Software

- Analytical network analysis package dedicated to thermal analysis of electric motors and generators
  - input geometry using dedicated editors
  - select cooling type, materials, etc and calculate steady state or transient temperatures
  - all difficult heat transfer data calculated automatically
    - easy to use by non heat transfer specialists
  - provides a detailed understanding of cooling and facilitates optimisation
    - what-if & sensitivity analysis

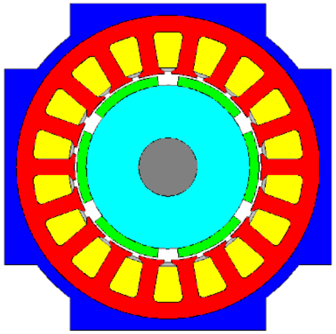
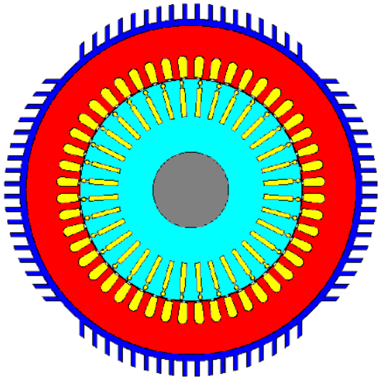
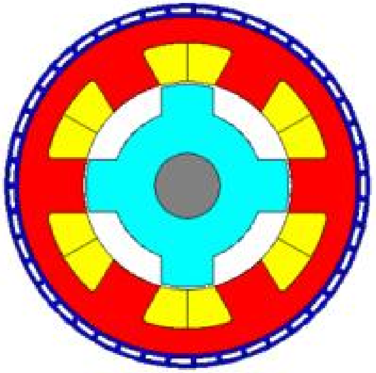
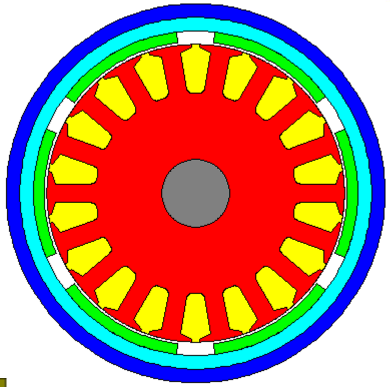
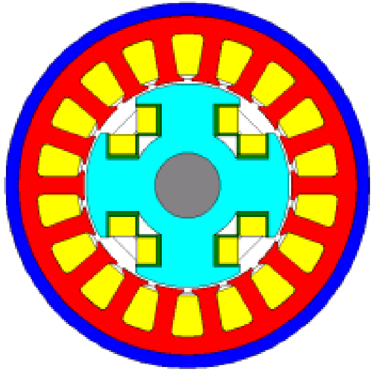
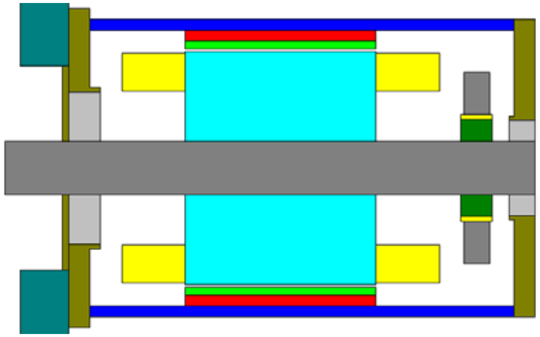
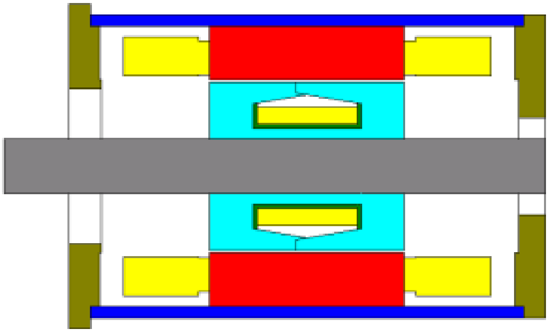




# Motor-CAD Motor Types

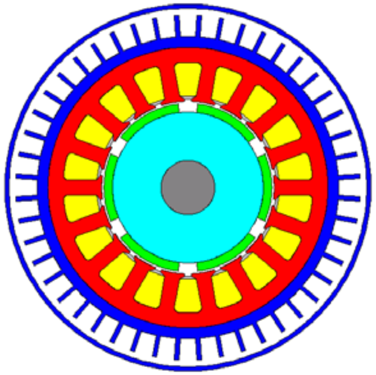
- Brushless Permanent Magnet
  - Inner and outer rotor
- Induction/Asynchronous
  - 1 and 3 phase
- Switched reluctance
- Permanent magnet DC
- Wound Field Synchronous
- Claw pole

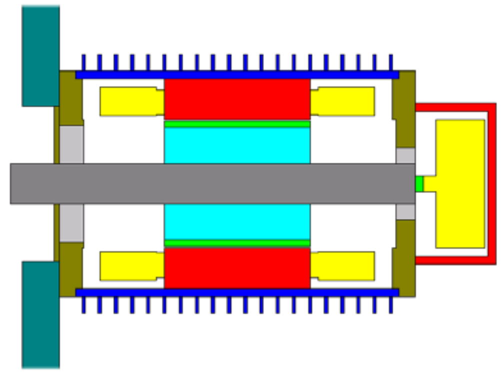








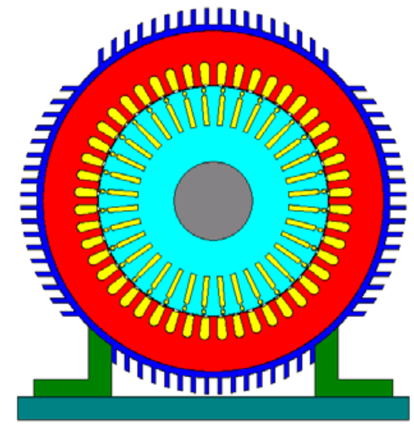
# Cooling Types

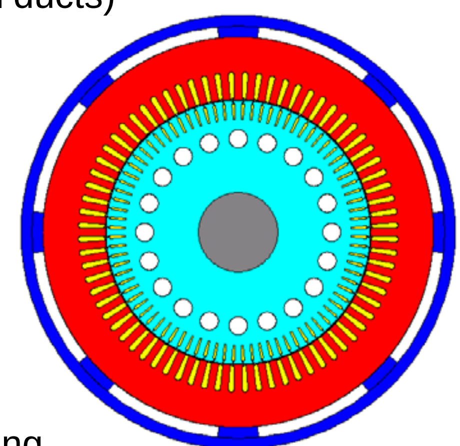
Motor-CAD includes proven models for an extensive range of cooling types

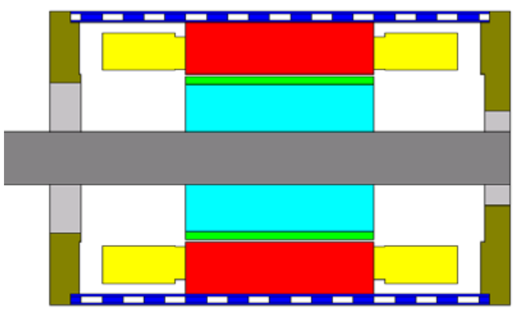
- Natural Convection (TENV)
  - many housing design types
- Forced Convection – (TEFC)
  - many fin channel design types
- Through Ventilation
  - rotor and stator cooling ducts
- Open end-shield cooling
- Water Jackets
  - many design types (axial and circumferential ducts)
  - stator and rotor water jackets
- Submersible cooling
- Wet Rotor & Wet Stator cooling
- Spray Cooling
- Direct conductor cooling
  - Slot water jacket
- Conduction
  - Internal conduction and the effects of mounting
- Radiation
  - Internal and external



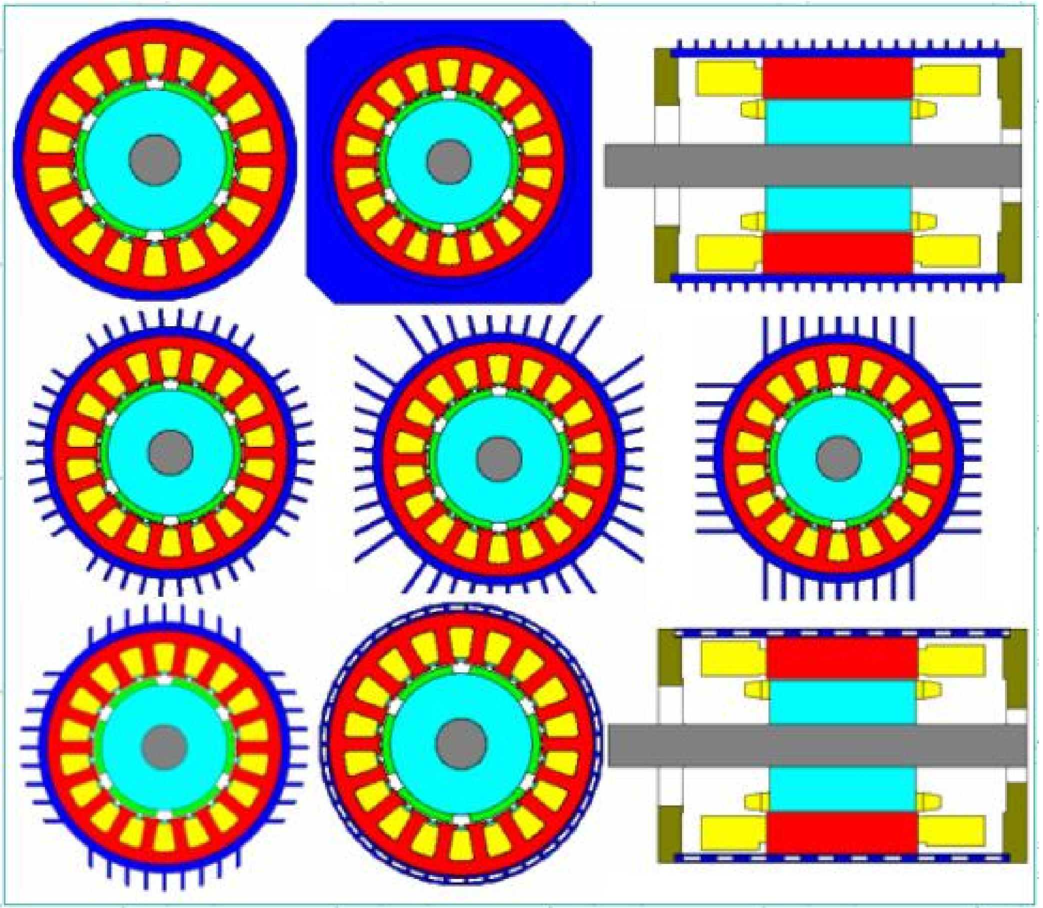


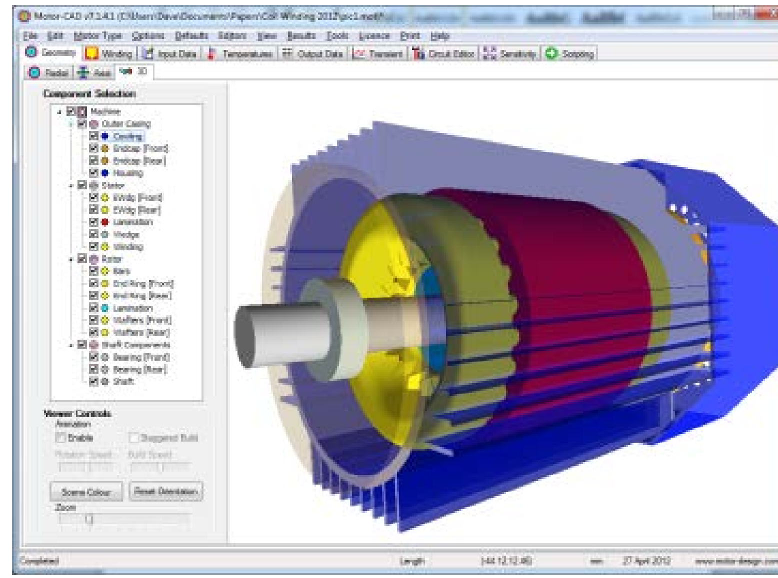






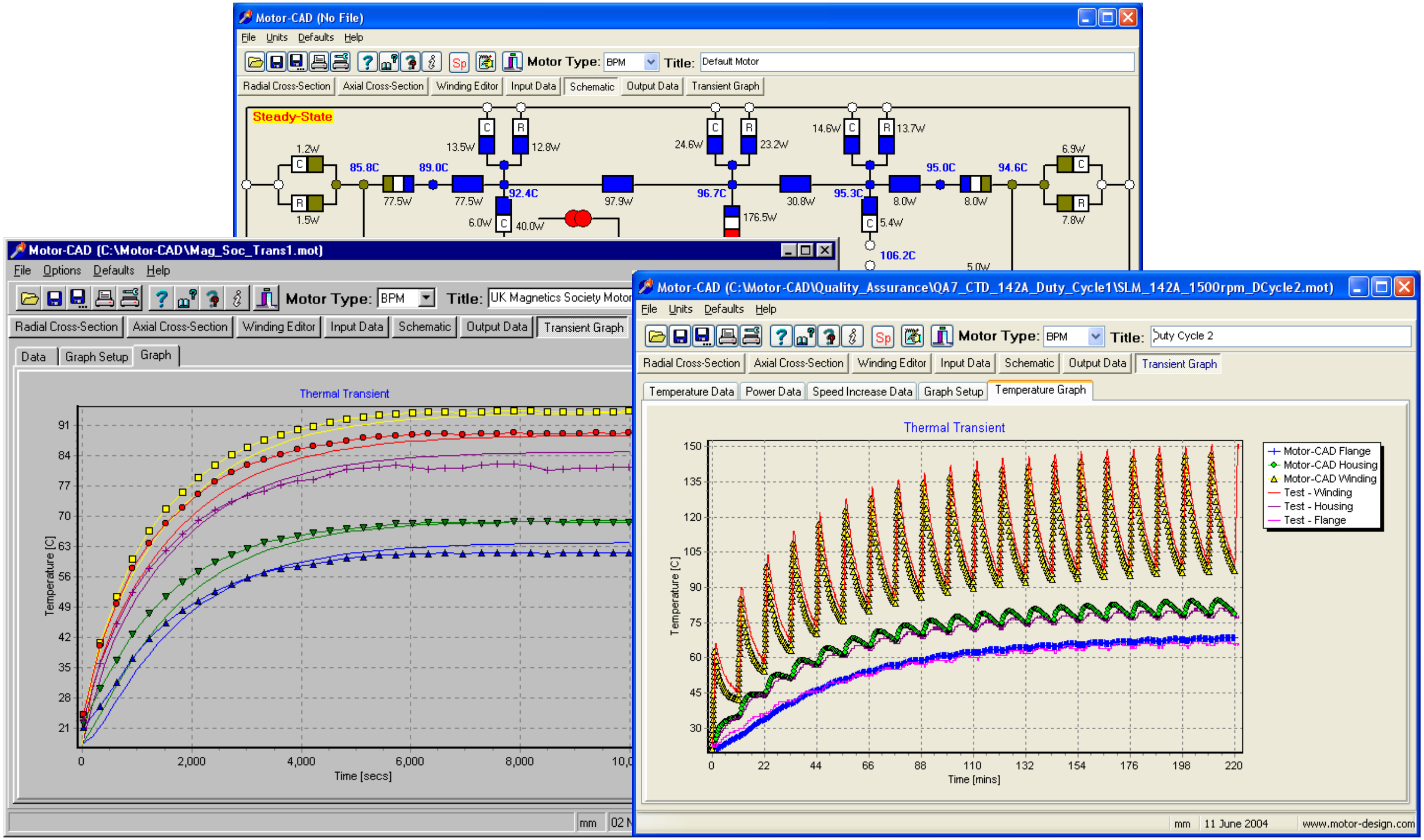
# Housing Types





- Many housing designs can be modeled and optimized
  - the designer selected a housing type that is appropriate for the cooling type to be used and then optimizes the dimensions, e.g. axial fin dimensions and spacing for a TEFC machine

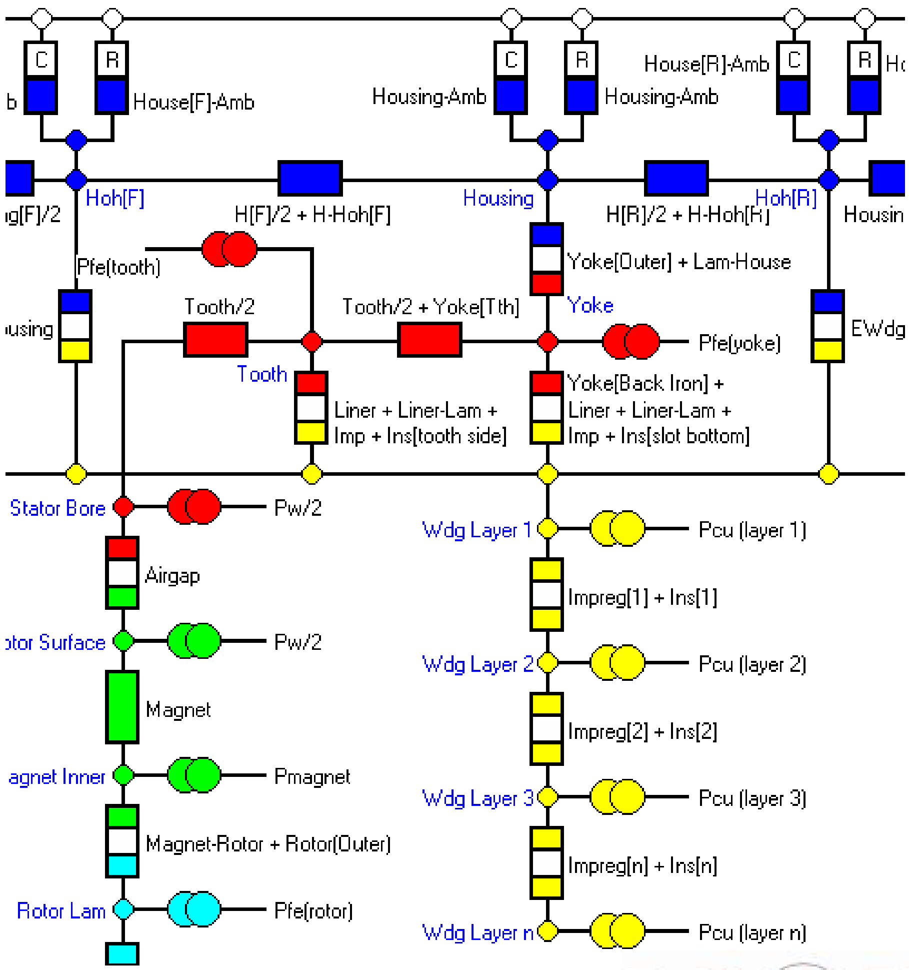
# Steady-State & Transient Analysis




- Some application are steady state and some operate with complex transient duty cycle loads

# Thermal Lumped Circuit Models

- similar to electrical network so easy to understand by electrical engineers
  - thermal resistances rather than electrical resistances
  - power sources rather than current sources
  - thermal capacitances rather than electrical capacitors (not shown here)
  - nodal temperatures rather than voltages
  - power flow through resistances rather than current
  - Place nodes at important places in motor cross-section

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# Thermal Lumped Circuit Models

- thermal resistances placed in the circuit to model heat transfer paths in the machine
  - conduction ($R = L/kA$)
    - path area (A) and length (L) from geometry
    - thermal conductivity (k) of material
  - convection ($R = 1/hA$)
    - heat transfer coefficient ($h$ – $W/m^2.C$) from empirical dimensionless analysis formulations (correlations)
      - many well proven correlations for all kinds of geometry in heat transfer technical literature
  - radiation ($R = 1/hA$)
    - $h = \sigma\, \varepsilon_1\, F_{1\text{-}2}\, (T_1^4 - T_2^4) / (T_1 - T_2)$
    - emissivity ($\varepsilon_1$) & view factor ($F_{1\text{-}2}$) from surface finish & geometry
- power input at nodes where losses occur
- thermal capacitances for transient analysis
  - Capacitance = Weight $\times$ Specific Heat Capacity

# Heat Transfer Ohms Law:

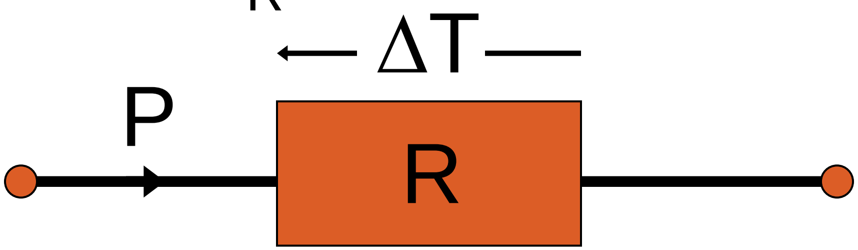
- In a thermal network the heat flow is given by:

$$P = \frac{\Delta T}{R} \qquad \text{electrical circuit} \quad I = \frac{V}{R}$$

$P$ = power [Watts]

$\Delta T$ = temperature difference [C]

$R$ = thermal resistance [C/W]




# Fluid Temperature Rise:

$$\Delta T = \frac{\text{Power Dissipated}}{\text{Volume Flow Rate x Density x } C_p}$$

$\Delta T$ = temperature difference [C]

$C_p$ = Specific Heat Capacity [J/kg C]

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- **Important Issues in Thermal Analysis of Electric Motors**
- Examples of use of thermal analysis to optimise electric machine designs

# Important Issues in Thermal Analysis of Electric Motors

- Conduction, Convection and Radiation
- Losses
- Winding Heat Transfer
- Interface Thermal Resistance
- Accuracy and Calibration

# Conduction Heat Transfer

- Heat transfer mode in a solid due to molecule vibration
- Good electrical conductors are also good thermal conductors
  - Would like good electrical insulators that are good thermal conductors
    - material research to try to achieve this
  - Metals have large thermal conductivities due to their well ordered crystalline structure
    - k is usually in the range of 15 – 400 W/m/C
  - Solid insulators not well ordered crystalline structure and are often porous
    - k is typically in the range of 0.1 – 1W/m/C (better than air with k ≈ 0.026W/m/C)
- Conduction thermal resistance calculated using R = L/kA
  - Path length (L) and area (A) from geometry, e.g. tooth width and area
  - Thermal conductivity (k) of material, e.g. that of electrical steel for tooth
- Only complexity is in the calculation of effective L, A and k for composite components such as winding, bearings, etc.
  - Motor-CAD benefits from research using numerical analysis and testing to develop reliable models for such complicated components

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# Convection Heat Transfer

- Heat transfer mode between a surface and a fluid due to intermingling of the fluid immediately adjacent to the surface (conduction here) with the remainder of the fluid due to fluid motion
  - Natural Convection – fluid motion due to buoyancy forces arising from change in density of fluid in vicinity of the surface
  - Forced Convection – fluid motion due to external force (fan and pump)
- Two types of flow
  - Laminar Flow, streamlined flow at lower velocities
  - Turbulent Flow, eddies at higher velocities
    - Enhanced heat transfer compared to laminar flow but a larger pressure drop
- Convection thermal resistance is calculated using:

$$R_C = 1/\ (A\ h) \qquad [C/W]$$

A = surface area [$m^2$]  h = convection heat transfer coefficient [$W/m^2/C$]

- Need to predict h – rule thumb, dimensionless analysis, CFD?

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# Convection Heat Transfer Coefficient

- $h_C$ can be calculated using empirical correlations based on dimensionless numbers (Re, Gr, Pr, Nu)
  - Just find a correlation with a similar geometry and cooling type to that being studied
    - Cylinder, flat plate, open/enclosed channel, etc.
  - Dimensionless numbers allow the same formulation to be used with different fluids and dimensions to those used in the original experiments
  - Hundreds of correlations available in the technical literature allowing engineers to carry out the thermal analysis of almost any shape of apparatus
    - Motor-CAD automatically selects the most appropriate correlation that matches the cooling type and surface shape

# Convection Analysis Dimensionless Numbers

- Reynolds number ($R_e$)
- Grashof number ($G_r$)
- Prandtl number ($P_r$)
- Nusselt number ($N_u$)

$$R_e = \rho\, v\, L / \mu$$

$$P_r = c_p\, \mu / k$$

$$G_r = \beta\, g\, \theta\, \rho^2\, L^3 / \mu^2$$

$$N_u = h\, L / k$$

$h$ = heat transfer coefficient [W/m$^2$C]

$\mu$ = fluid dynamic viscosity [kg/s m]

$\rho$ = fluid density [kg/m$^3$]

$k$ = thermal conductivity of the fluid [W/mC]

$c_p$ = specific heat capacity of the fluid [kJ/kg.C]

$v$ = fluid velocity [m/s]

$\theta$ = surface to fluid temperature [C]

$L$ = characteristic length of the surface [m]

$\beta$ = coefficient of cubical expansion of fluid [1/C]

$g$ = acceleration due to gravity [m/s2]

- The dimensionless numbers are functions of fluid properties, size (characteristic length), fluid velocity (forced convection), temperature (natural convection) and gravity (natural convection)

# Natural and Forced Convection Correlations

- Natural convection is present over most surfaces
  - Present even over surfaces that are designed for forced convection
    - e.g. TEFC machine with axial fins (at low speed the fan will not provide much forced air and natural convection will dominate)
  - Large set of correlations typically required for complex shapes
    - For very complex shapes area averaged composite correlations are used where the complex geometry (e.g. finned housing) is divided into a set of simpler shapes for which convection correlations are known
      - For instance a cylinder with axial fins is divided to various cylindrical and open fin channel sections of different orientation
      - A positive aspect is that if extremely small fins are attached to the cylinder the same results are given as for a cylindrical correlation
- Forced convection is present over surfaces with fluid movement due to a fan or pump (or movement of the device or wind)
  - A more limited set of correlations are required to calculate forced convection in electrical machines (Flat Plate, Open Fin Channel, Enclosed Channel, Rotating Airgap, End Space Cooling)

# General Form for Convection Correlations

## Natural Convection

General form of natural convection correlation :

$$N_u = a\,(G_r P_r)^b$$

– a & b are curve fitting constants

– Transition from laminar to turbulent flow:

107 < GrPr < 109 (GrPr = Ra – Rayleigh number)

## Forced Convection

General form of convection correlation for forced convection:

$$Nu = a\,(Re)^b\,(Pr)^c$$

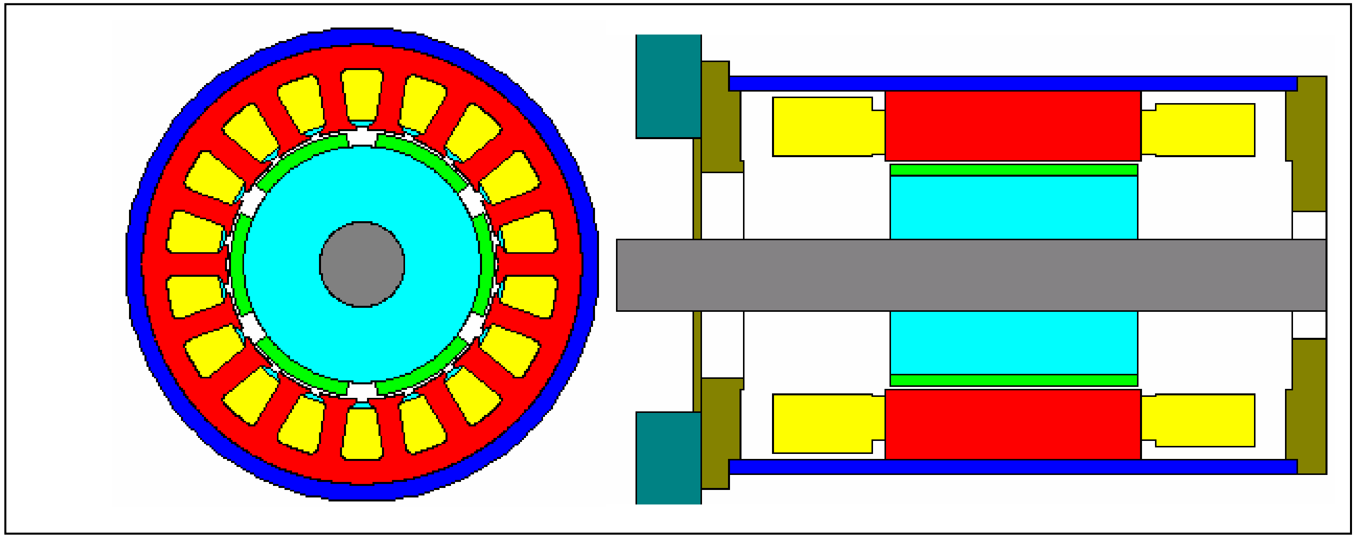
– a, b & c are curve fitting constants

– Internal flow laminar/turbulent transition $R_e \approx 2300$

(fully turbulent $R_e > 5 \times 10^4$)

– External flow laminar/turbulent transition $R_e \approx 5 \times 10^5$

# Horizontal Cylinder (Natural Convection)



- A formulation for average Nusselt number of a horizontal cylinder of diameter d:

  $N_u = 0.525\ (G_r\ P_r)^{0.25} \quad (10^4 < G_rP_r < 10^9)$ Laminar

  $N_u = 0.129\ (G_r\ P_r)^{0.33} \quad (10^9 < G_rP_r < 10^{12})$ Turbulent

  $h = N_u \times k / d$

  Fluid properties at mean film temperature (average of surface and bulk fluid temperatures)

# Vertical Cylinder (Natural Convection)

- A formulation for average Nusselt number of a vertical cylinder of height L:

$$N_u = 0.59\,(G_r\,P_r)^{0.25}$$

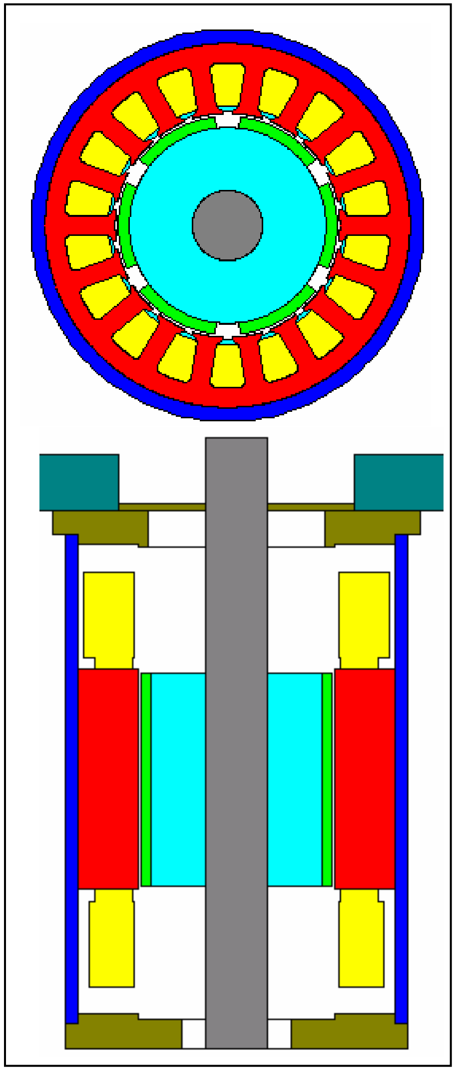
$$(10^4 < G_rP_r < 10^9) \quad \text{Laminar}$$

$$N_u = 0.129\,(G_r\,P_r)^{0.33}$$

$$(10^9 < G_rP_r < 10^{12}) \quad \text{Turbulent}$$

$$h = N_u \times k / L$$

Fluid properties at mean film temperature (average of surface and bulk fluid temperatures)



# Vertical Flat Plate (Nat Convection)

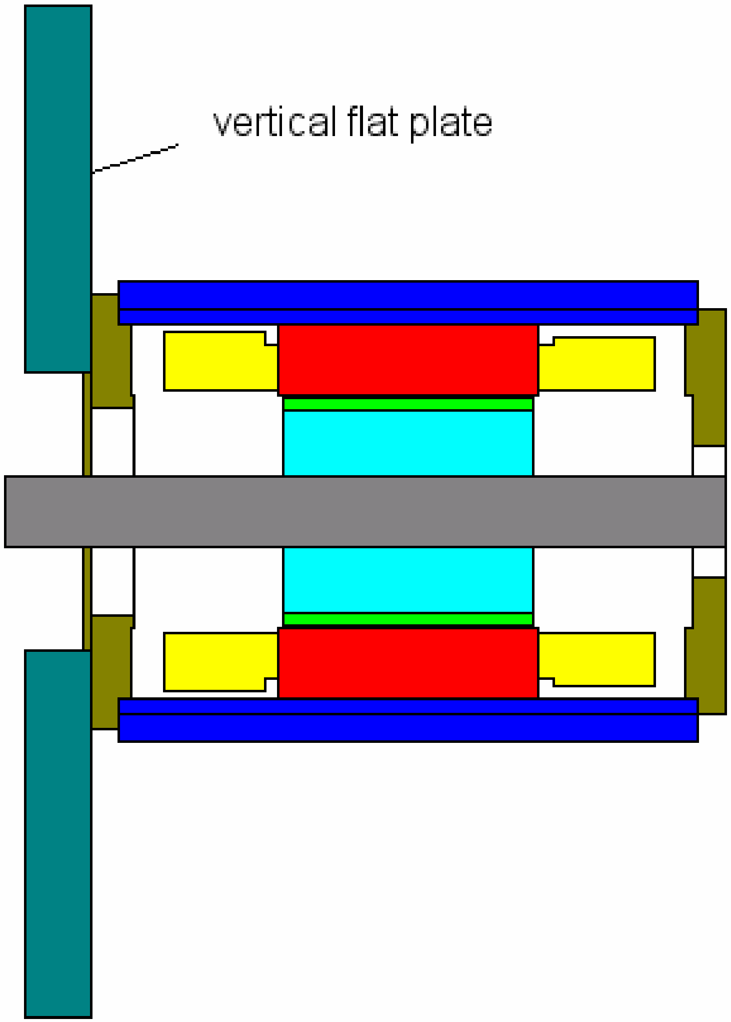
- A formulation for average Nusselt number of a vertical flat plate of height L:

$$N_u = 0.59\ (G_r\ P_r)^{0.25}$$
$$(10^4 < G_rP_r < 10^9) \quad \text{Laminar}$$

$$N_u = 0.129\ (G_r\ P_r)^{0.33}$$
$$(10^9 < G_rP_r < 10^{12}) \quad \text{Turbulent}$$

$$h = N_u \times k / L$$

fluid properties at mean film temperature (average of surface and bulk fluid temperatures)



# Horizontal Flat Plate (Nat Conv)

Upper Face:

$$N_u = 0.54\ (G_r\ P_r)^{0.25}$$

$$(10^5 < G_rP_r < 10^8) \quad \text{Laminar}$$

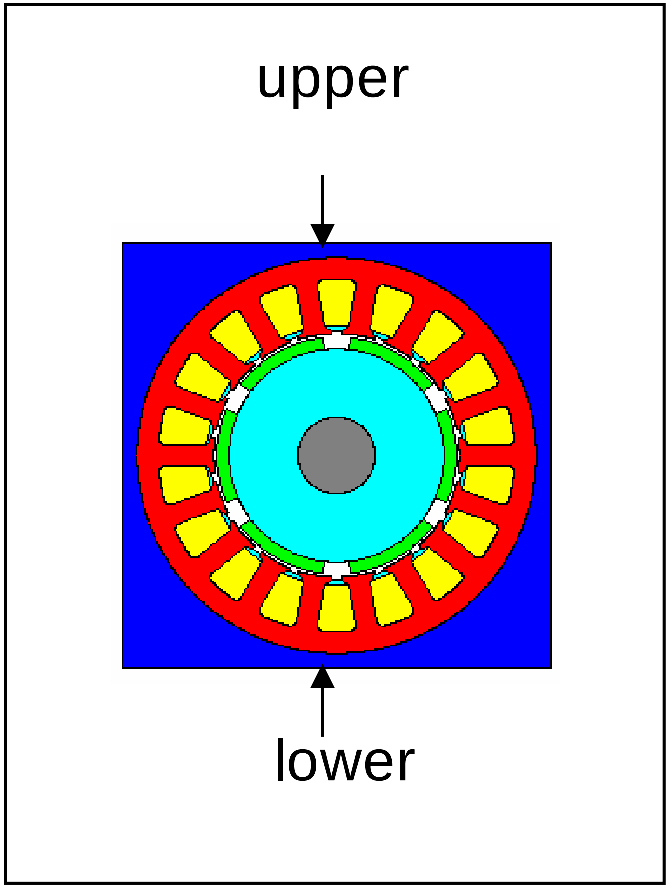
$$N_u = 0.14\ (G_r\ P_r)^{0.33}$$

$$(10^8 < G_rP_r) \quad \text{Turbulent}$$

Lower Face:

$$N_u = 0.25\ (G_r\ P_r)^{0.25}$$

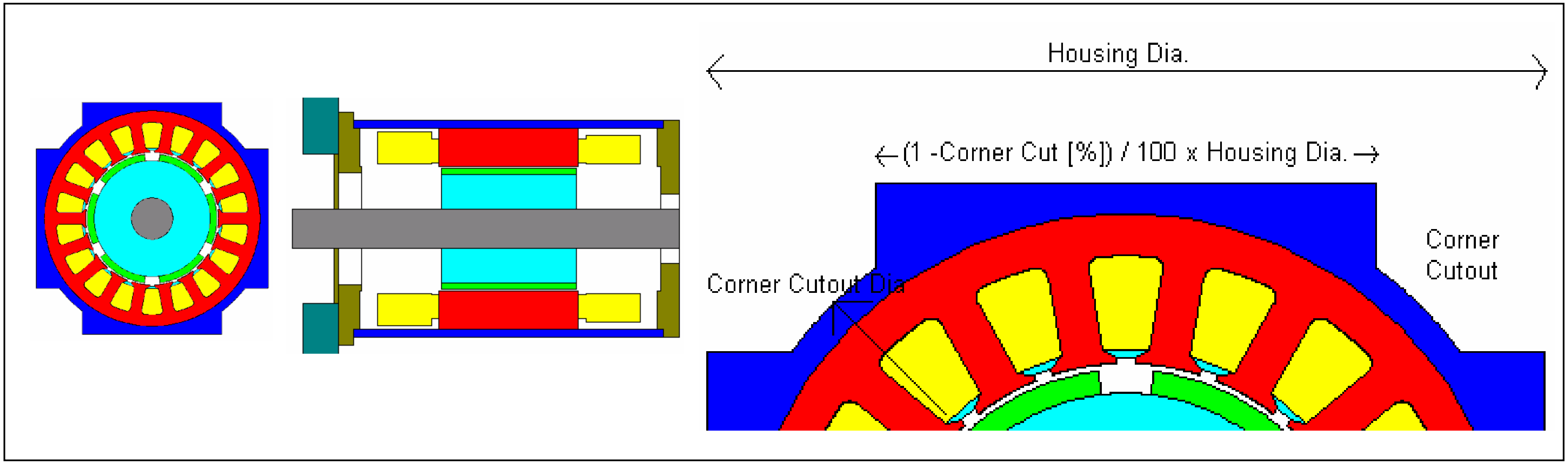
$$(10^5 < G_rP_r < 10^8) \quad \text{Laminar}$$




$$h = N_u \times k / L$$

Fluid properties at mean film temperature (average of surface and bulk fluid temperatures)

# Horizontal Servo Housing (Nat Conv)




- Average of the following:
  - Horizontal Cylinder × Corner Cutout [%]/100
  - Horizontal Square Tube × {1 - Corner Cutout [%]/100}
- As corner cut-out approaches 100% then the cylinder correlation predominates and as it approaches 0% the tube correlation predominates

# Vertical Fin Channel (N Conv)

- Ref [1] gives a formulation for Nusselt number of u-shaped vertical channels (laminar flow):

$$N_u = \frac{r/L \times G_r P_r}{Z} \times \left(1 - \exp\left[-Z\left(\frac{L \times 0.5}{r \times G_r P_r}\right)^{0.75}\right]\right)$$

$$Z = 24 \times \frac{\left[1 - 0.483 \times \exp\left(-0.17/a\right)\right]}{\left\{\left[1 + a/2\right]^3 \times \left[\begin{array}{l} 1 + \left(1 - \exp\left(-0.83a\right)\right) \times 9.14\sqrt[3]{a} \\ \times \exp\left(-465 \times fin\_spacing\right) - 0.61 \end{array}\right]^3\right\}}$$

h = $N_u$ × k / r

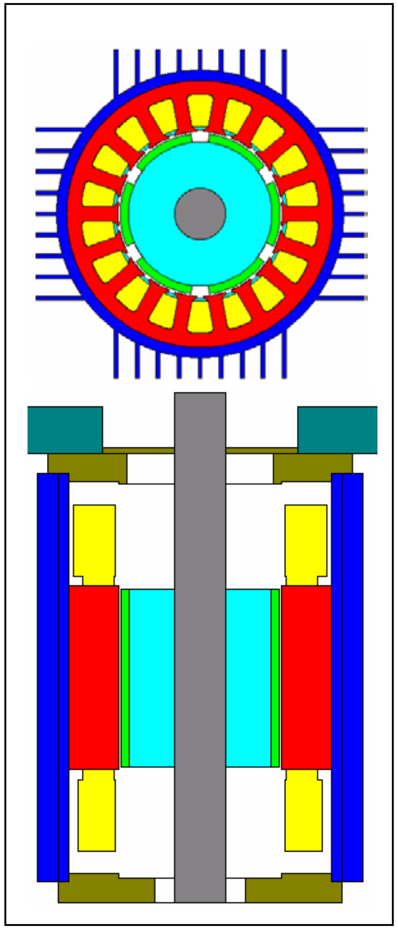
a = channel aspect ratio fin_spacing/fin_depth

r = Characteristic Length (fin hydraulic radius)

= 2 × fin_depth × fin_spacing /(2 × fin_depth + fin_spacing)

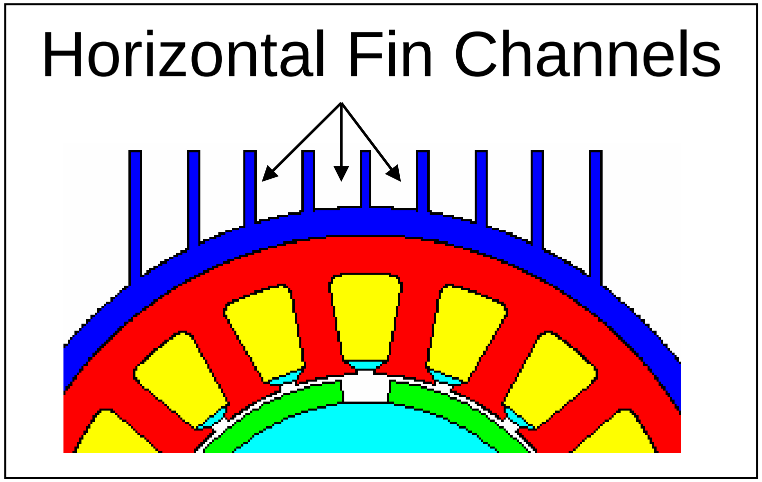
L = fin height

The fluid properties are evaluated at the wall temperature (except volumetric coefficient of expansion which is evaluated at the mean fluid temperature).



[1] Van De Pol, D.W. & Tierney, J.K. : Free Convection Nusselt Number for Vertical U-Shaped Channels, Trans. ASME, Nov. 1973

# Horizontal Fin Channel (Nat Conv)




- Ref [1] gives a formulation for Nusselt number of u-shaped horizontal channels (laminar flow):

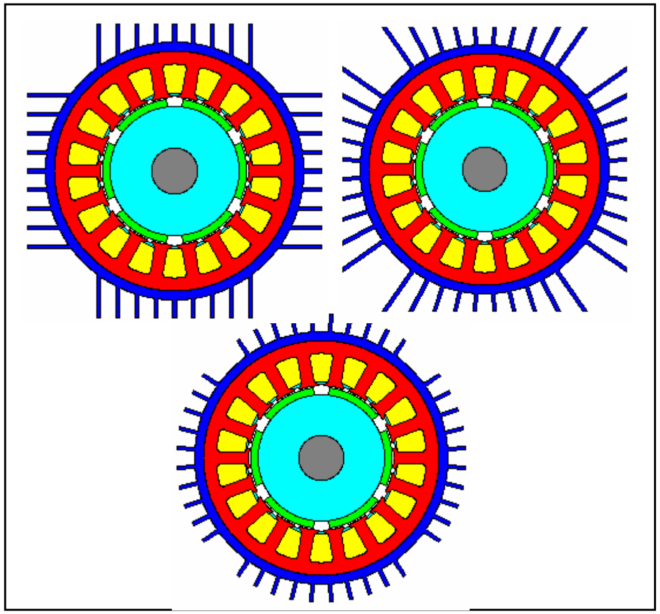
$$N_u(s) = 0.00067 \times G_r(s) P_r(s) \times \left[ 1 - \exp\left\{ \left( \frac{-7640}{G_r(s)} P_r \right)^{0.44} \right\}^{1.7} \right]$$

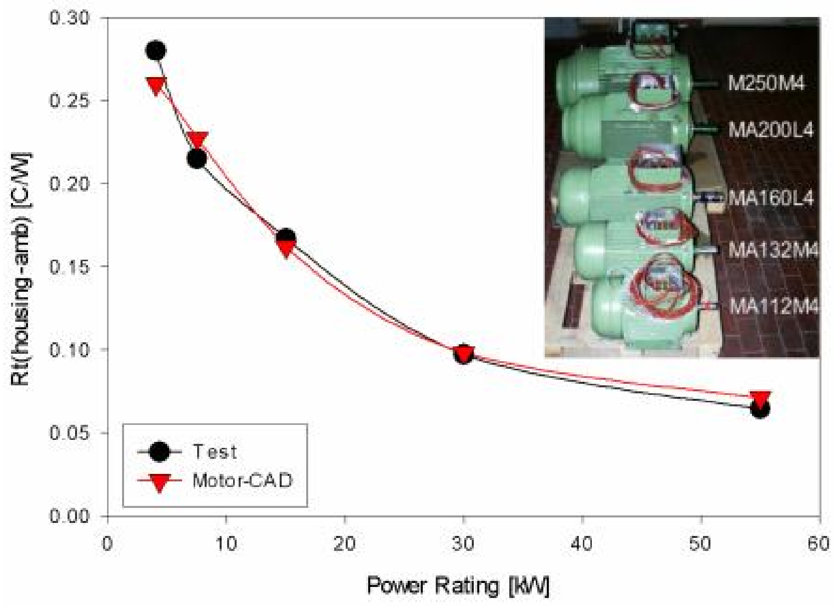
h = $N_u$(s) × k / s

s = fin spacing used as characteristic dimension

[1] Jones, C.D., Smith, L.F. : Optimum Arrangement of Rectangular Fins on Horizontal Surfaces for Free-Convection Heat Transfer, Trans. ASME, Feb 1970.

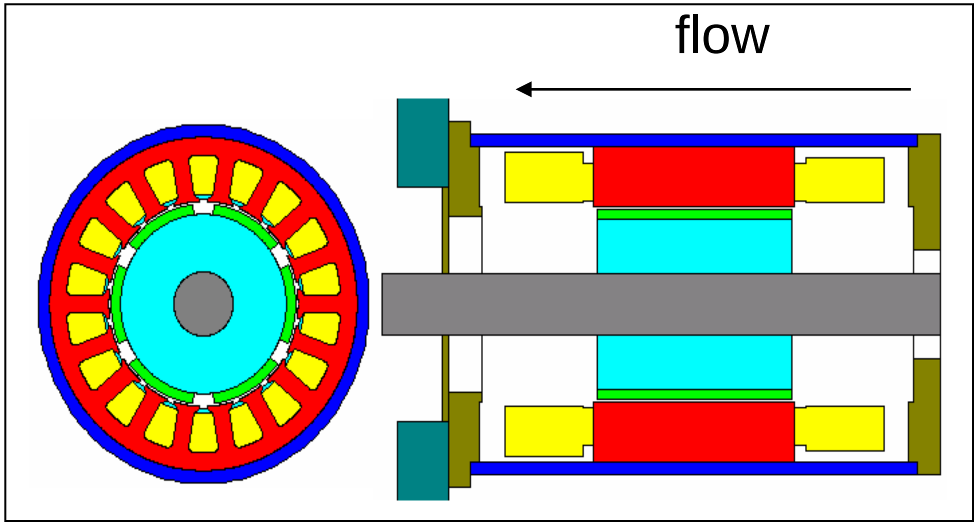
# Axial Finned Housing with Natural Convection






- The axial finned motors above are designed for a shaft mounted fan (TEFC)
- When used as a variable speed drive at low speed there is little forced convection as natural convection dominates
  - Therefore we must be able to calculate such surfaces with natural convection
- Use of composite calculations based on averages of all the different simple shapes found in the more complex shape can give accurate results
  - Good agreement between Motor-CAD with default data and measured data for a range of TEFC motors operating at zero speed (work carried out by Boglietti at Politecnico di Torino) – see graph above

# Flat Plate Forced Conv (External Flow)




- Ref [1] gives a formulation for average Nusselt number of flat plate (or horizontal cylinder) length L:

  $N_u = 0.664\ (R_e)^{0.5}\ (P_r)^{0.33}$ $\quad (R_e < 5 \times 10^5)$

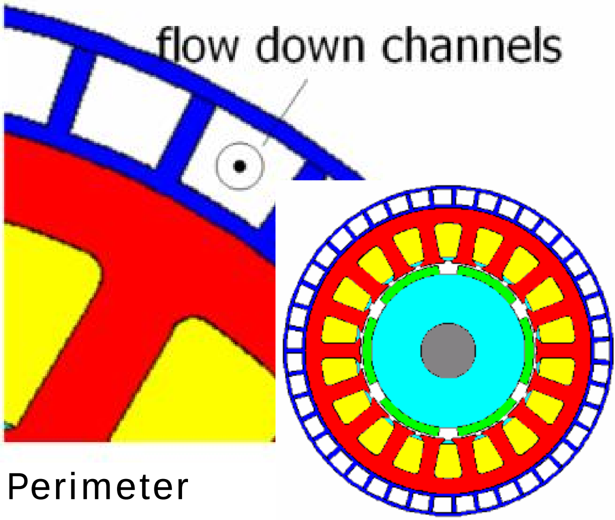
  $N_u = [0.037\ (R_e)^{0.8} - 871]\ (P_r)^{0.33}$ $\quad (R_e > 5 \times 10^5)$

  $h = N_u \times k / L$

  Fluid properties at mean film temperature (average of surface and bulk fluid temperatures)

[1] Incropera, F.P & DeWitt, D.P.: Introduction to Heat Transfer, Wiley, 1990.

# Enclosed Channel Forced Convection




- The following formulations are used to calculate h from enclosed channels

  Re = Dh x Fluid Velocity / Kinematic Viscosity

  Dh = Channel Hydraulic Diameter

  - Dh = 2 × Gap [Concentric Cylinders]
  - Dh = 4 × Channel Cross Sectional Area / Channel Perimeter [Round/Rectangular Channels]

  h = Nu × (Fluid Thermal Conductivity) / (Dh)

- The flow is assumed to be fully laminar when Re < 2300 in Round/Rectangular Channels and when Re < 2800 in Concentric Cylinders
- The flow is assumed to be fully turbulent when Re > 3000 (in practice the flow may not be fully turbulent until Re > 10000)
- A transition between laminar and turbulent flow is assumed for Re values between those given above

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# Enclosed Channel (Forced Conv)

## Laminar Flow

- Concentric Cylinders (adaptation of formulation for parallel plates which includes entrance length effects):

$$N_u = 7.54 + \left[0.03 \times {D_h}/{L} \times R_e \times P_r\right] \div \left[1 + 0.016 \times \left({D_h}/{L} \times R_e \times P_r\right)^{2/3}\right]$$

  – The 2nd term in the above equation is the entrance length correction which accounts for entrance lengths where the velocity and temperature profiles are not fully developed.

- Round Channels (which includes entrance length effects):

$$N_u = 3.66 + \left[0.065 \times {D_h}/{L} \times R_e \times P_r\right] \div \left[1 + 0.04 \times \left({D_h}/{L} \times R_e \times P_r\right)^{2/3}\right]$$

- Rectangular Channels (adaptation of formulation for round channels):

$$N_u = 7.49 - 17.02 \times {H}/{W} + 22.43 \times \left({H}/{W}\right)^2 - 9.94 \times \left({H}/{W}\right)^3$$

$$+ \left[0.065 \times {D_h}/{L} \times R_e \times P_r\right] \div \left[1 + 0.04 \times \left({D_h}/{L} \times R_e \times P_r\right)^{2/3}\right]$$

where H/W = Channel Height/Width Ratio

# Enclosed Channel (Forced Conv)

## Turbulent Flow

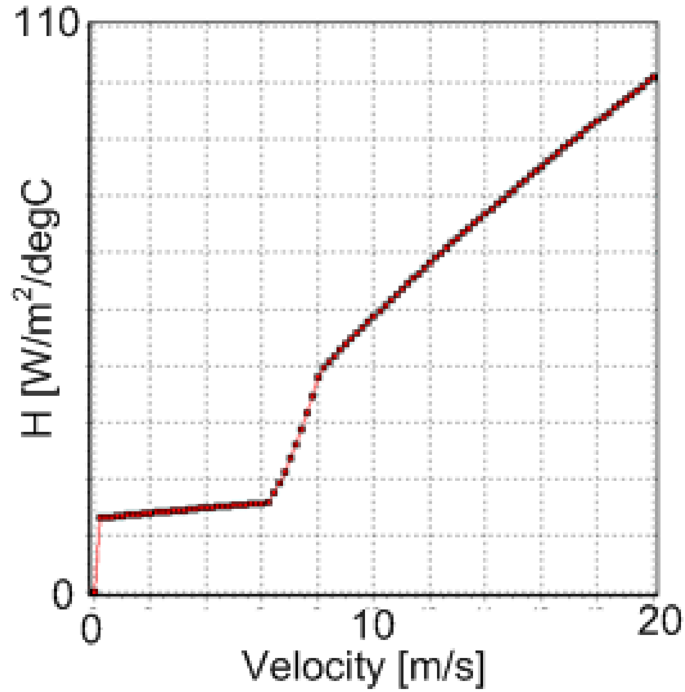
- Calculated using Gnielinski's formula for fully developed turbulent flow,

  i.e., $3000 < R_e < 1 \times 10^6$:

$$N_u = \left[ \left( f/8 \right) \times \left( R_e - 1000 \right) \times P_r \right] \div \left[ 1 + 12.7 \times \left( f/8 \right)^{0.5} \times \left( P_r^{2/3} - 1 \right) \right]$$

Friction Factor and for a smooth wall is:

$$f = [0.790 \times \mathrm{Ln}(R_e) - 1.64]^{-2}$$




## Transition from Laminar to Turbulent Flow

- $N_u$ is calculated for both laminar and turbulent flow using the above formulations. A weighted average (based on $R_e$) is then used to calculate $N_u$.
- It is noted that h increases dramatically as the flow regime changes from being laminar to turbulent flow.

# Fin Channel (Open) – Forced Conv

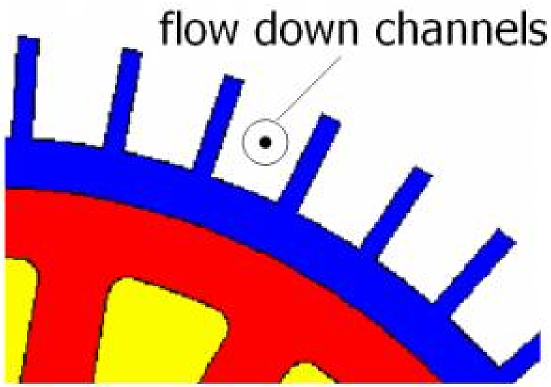
Heiles [1] calculates h for a forced cooled open fin channel:

$$h = \sigma_{Air} \times Cp_{Air} \times D \times \text{Air Velocity} / (4 \times L) \times [1 - \exp(-m)]$$

$$m = 0.1448 \times L^{0.946} / D^{1.16} \times \{k_{Air} / (\sigma_{Air} \times Cp_{Air} \times \text{Air Velocity}])\}^{0.214}$$

D = Hydraulic Diameter = 4 x channel area / channel perimeter (including open side)
L = Axial length of cooling fin

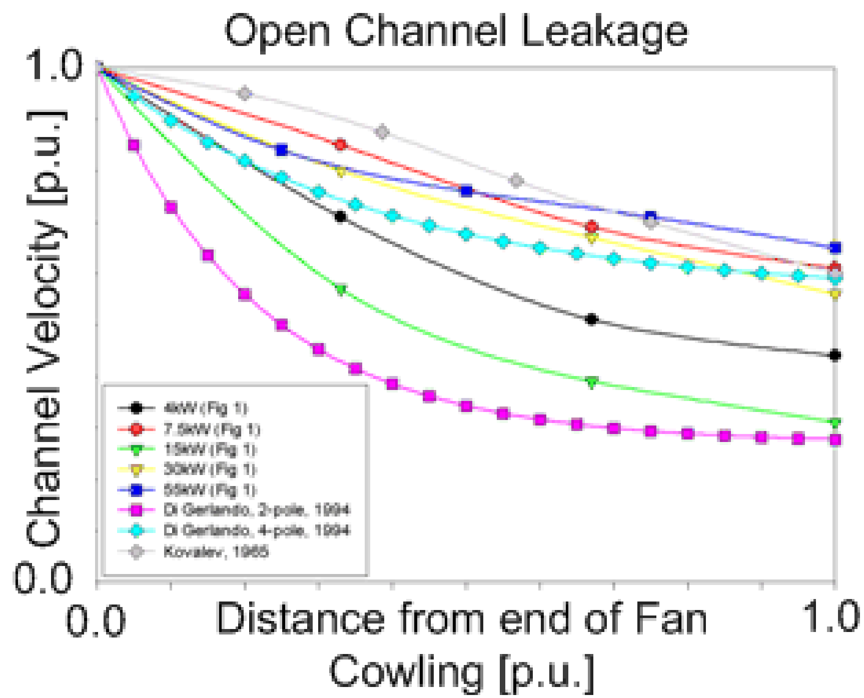
- This assumes isothermal wall, laminar air flow with air properties calculated at the film temperature = $(T_{free\text{-}stream} + T_{wall})/2$.
- Heiles recommends the use of a Turbulence Factor to directly multiply h by – his tests indicate typical turbulence factors in the range 1.7 - 1.9 which seem independent of the flow velocity



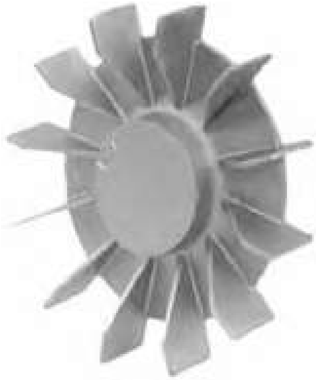

[1] Heiles, F. : Design and Arrangement of Cooling Fins, Elecktrotecknik und Maschinenbay, Vol. 69, No. 14, July 1952.

# Axial Fin Channel Air Leakage

- Typical form of open axial channel air leakage shown
- Complex function of
  - Cowling design
    - Gap
    - Fin overhang
  - Fan design
  - Speed
  - Motor size
  - Fin design
  - Fin roughness
  - Restrictions
  - etc.







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# Mixed Convection

- The total heat transfer coefficient due to convection ($h_{MIXED}$) is the combined free and forced convection heat transfer coefficients.
- $h_{MIXED}$ is calculated using:

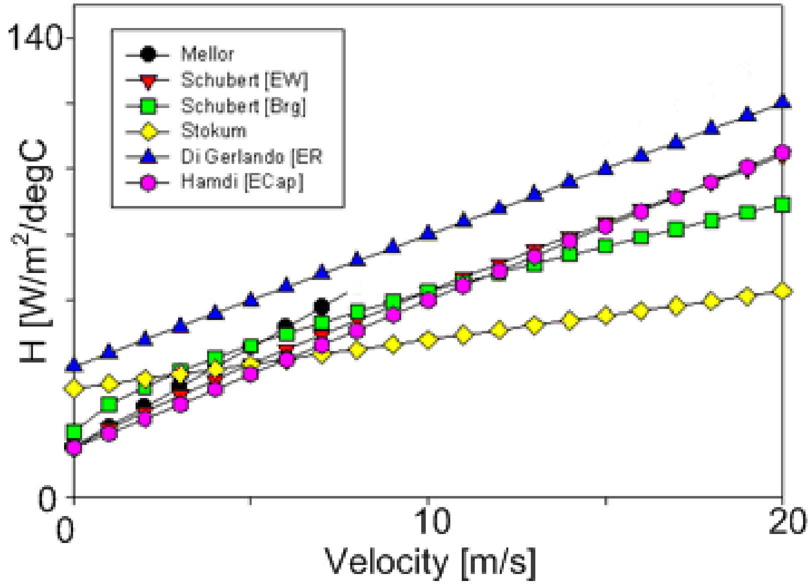
$$h^3_{MIXED} = h^3_{FORCED} + h^3_{NATURAL}$$

- The Motor Orientation determines the sign (±):
  - \+ in assisting and transverse flows
  - – in opposing flows

# End Space Cooling

- Complex area of machine cooling due to flow around end-windings
- Turbulent flow which depends upon:
  - Shape & length of end-windings
  - Fanning effects of internal fans, end-rings, wafters/rotor wings, etc.
  - Surface finish of the end sections of the rotor
- Several authors have studied such cooling and most propose the use of the following formulation:

$$h = k1 \times [1 + k2 \times \text{velocity}^{k3}]$$

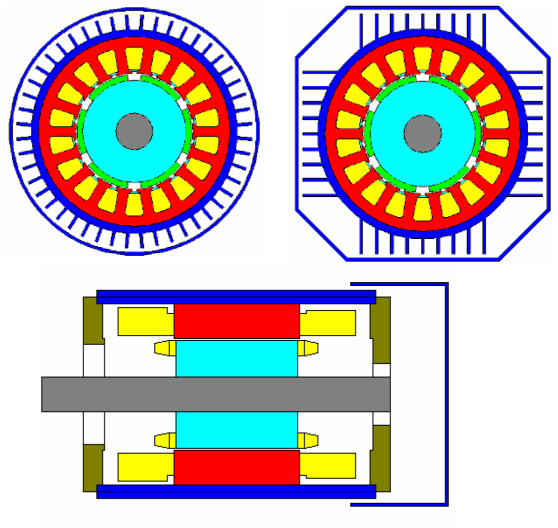


# Airgap Heat Transfer

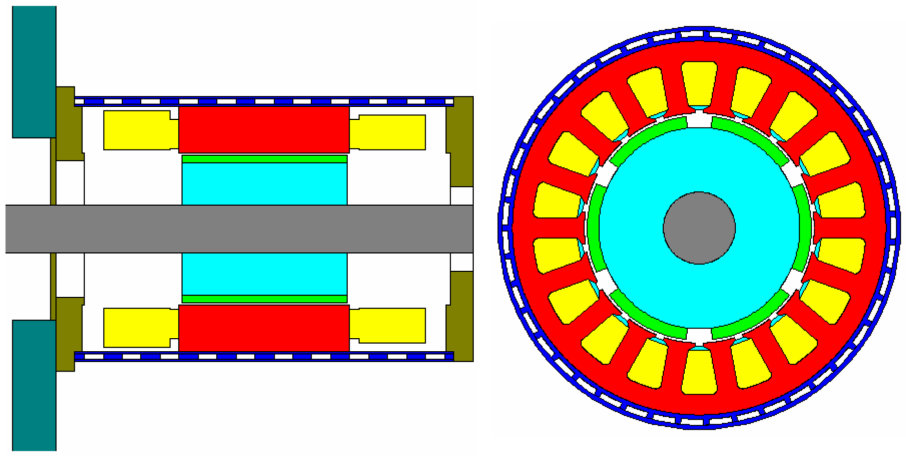
- Airgap heat transfer is often calculated using a formulation developed from Taylor's work on concentric cylinders rotating relative to each other
  - Heat transfer by conduction when flow is laminar
  - Increase in heat transfer when the airgap $R_e$ number increases above a certain critical value at which point the flow takes on a regular vortex pattern (vortex circular rotational eddy pattern)
  - Above a higher critical $R_e$ value the flow becomes turbulent and the heat transfer increases further (turbulent is more of a micro eddy flow)

Taylor, G.I.: 'Distribution of Velocity and Temperature between Concentric Cylinders', Proc Roy Soc, 1935, 159, PtA, pp 546-578

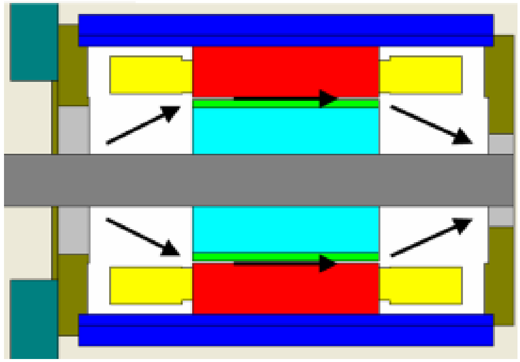
# Typical Air/Fluid Cooling Types



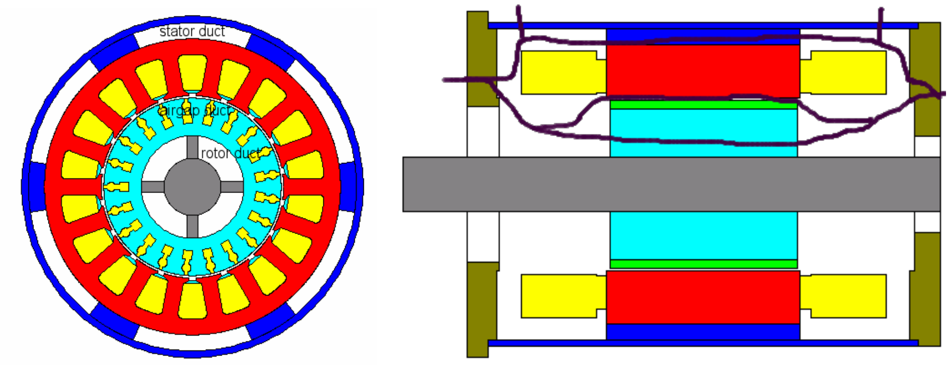
Blown Over



Water Jacket



Wet Rotor




Through Ventilation

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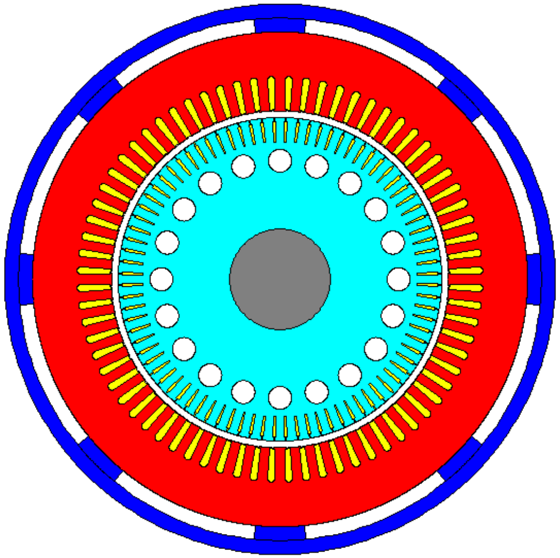
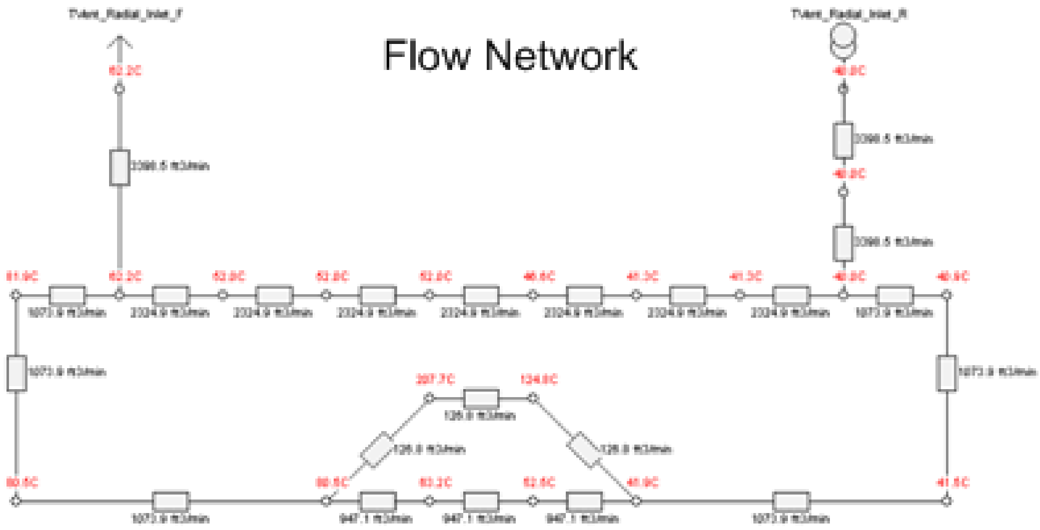
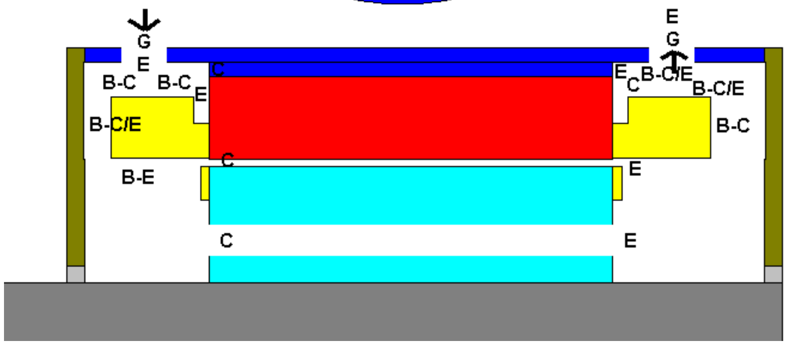
# Flow Network Analysis

- Separate topic not covered in this tutorial where a set of analytically based flow resistances are put together in a circuit to predict the pressure drops and flow
- Pressure drops due to duct wall friction and restrictions to flow (bend, expansion, contraction, etc.).






| | | |
|---|---|---|
| G | - | GRILL/GUARD |
| E | - | EXPANSION |
| C | - | CONTRACTION |
| B | - | BEND |
| C/E | - | CONTRACTION OR EXPANSION |
| B-C | - | MAX(BEND AND CONTRACTION) |

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# Radiation Heat Transfer

- The heat transfer mode from a surface due to energy transfer by electromagnetic waves
- Convection thermal resistance is calculated using:

$$R_R = 1/\ (A\ h_R)\ [C/W]$$

- $h_R$ can be calculated using the formula:

$$h_R = \frac{\sigma\ \varepsilon\ F_{1\text{-}2}\ (T_1^4 - T_2^4)}{T_1 - T_2} \qquad [W/m^2.C]$$

$h_R$ = radiation heat transfer coefficient $[W/m^2.C]$
$A$ = area of radiating surface $[m^2]$
$\sigma$ = Stefan-Boltzmann constant ($5.669 \times 10^{-8}\ W/m^2/K^4$)
$T_1$ = absolute temperature of radiating surface [K]
$T_2$ = absolute temperature of surface radiated to (ambient) [K]
$\varepsilon$ = emissivity of radiating surface ($\varepsilon \leq 1$)
$F_{1\text{-}2}$ = view factor ($F_{1\text{-}2} \leq 1$) – calculated from geometry

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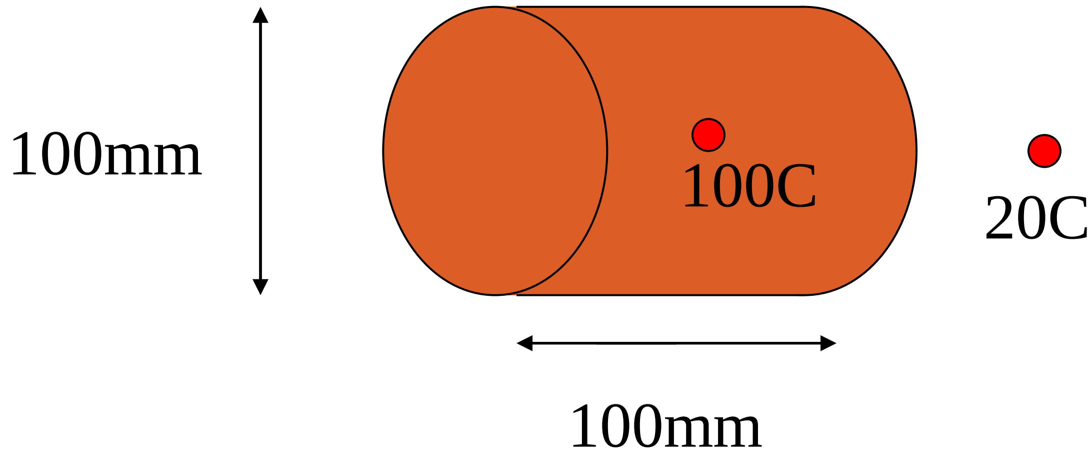
# Typical Emissivity ($\varepsilon$) Data

| Material | Emissivity | Material | Emissivity |
|---|---|---|---|
| Aluminium | | Iron | |
| • black anodised | 0.86 | • polished | 0.07 – 0.38 |
| • polished | 0.03 – 0.1 | • oxidised | 0.31 – 0.61 |
| • heavily oxidized | 0.20 – 0.30 | Nickel | 0.21 |
| • sandblasted | 0.41 | Paints | |
| Alumina | 0.20 – 0.50 | • white | 0.80 – 0.95 |
| Asbestos | 0.96 | • grey | 0.84 – 0.91 |
| Carbon | 0.77 – 0.84 | • black lacquer | 0.96 – 0.98 |
| Ceramic | 0.58 | Quartz, fused | 0.93 |
| Copper | | Rubber | 0.94 |
| • polished | 0.02 | Silver, polished | 0.02 – 0.03 |
| • heavily oxidized | 0.78 | Stainless Steel | 0.07 |
| Glass | 0.95 | Tin, bright | 0.04 |

From the Stokes Research Institute

# Convection/Radiation Example Calculations




- For the following conditions, calculate h for the above horizontal cylinder
  - Natural convection with air as the fluid
  - Forced convection with 5m/s of air flowing axially over the cylinder
  - Forced convection with 5m/s of water flowing axially over the cylinder
  - Radiation from cylindrical surface with emissivity of 0.9
- Not realistic problem as we already know the temperature
  - Usually calculate h to calculate temperature (non-linear system)

# Natural Convection (fluid = air)

- Laminar or turbulent flow? (Gr Pr), L= 0.1m

  $Gr = \beta\, g\, \theta\, \rho^2 L^3 / \mu^2$ $\qquad$ $Pr = c_p\, \mu / k$

- Air properties at mean film temperature = (20 +100)/2 = 60C

  $k = 0.0287$ W/m K

  $\rho = 1.06$ kg/m$^3$

  $\mu = 1.996\times10^{-5}$ kg/m s

  $c_p = 1008$ J/kg K

  $Gr = 1/(273 + 60) \times 9.81 \times (100 - 20) \times 1.06^2 \times 0.1^3 / (1.996\times10^{-5})^2 = 6.65\times10^6$

  $Pr = 1008 \times 1.996\times10^{-5} / 0.0287 = 0.701$

  $GrPr = 6.65\times10^6 \times 0.701 = 4.66\times10^6$

- Horizontal Cylinder correlation with Laminar Flow from $10^4 < GrPr < 10^9$

  $N_u = 0.525\,(Gr\,Pr)^{0.25}$ (laminar flow)

  $N_u = 0.525\,(4.66\times10^6)^{0.25} = 24.39$

- Heat Transfer Coefficient

  $h = N_u \times k / L = 24.39 \times 0.0287 / 0.1$

  **$h = 7.0$ W/m$^2$/C**

# Forced Convection (fluid = air)

- Laminar or turbulent flow? ($R_e$ at 5m/s, L = 0.1)

$$R_e = \rho \, v \, L / \mu$$

- Air properties at mean film temperature = (20 +100)/2 = 60C

$$k = 0.0287 \text{ W/m C}$$
$$\rho = 1.06 \text{ kg/m}^3$$
$$\mu = 1.996 \times 10^{-5} \text{ kg/m s}$$
$$c_p = 1008 \text{ J/kg C}$$
$$P_r = c_p \, \mu / k = 1008 \times 1.996 \times 10^{-5} / 0.0287 = 0.701$$
$$R_e = 1.06 \times 5 \times 0.1 / 1.996 \times 10^{-5} = 2.66 \times 10^4$$

- Flat Plate correlation (external flow) with Laminar Flow as Re < $5 \times 10^5$

$$N_u = 0.664 \, (R_e)^{0.5} \, (P_r)^{0.33}$$
$$N_u = 0.664 \, (2.66 \times 10^4)^{0.5} \, (0.701)^{0.33} = 96.32$$

- Heat Transfer Coefficient

$$h = N_u \times k / L = 96.32 \times 0.0287 / 0.1$$
$$\mathbf{h = 27.64 \; W/m^2/C}$$

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# Forced Convection (fluid = water)

- Laminar or turbulent flow? ($R_e$ at 5m/s, L = 0.1)

  $R_e = \rho\ v\ L\ /\ \mu$

- Water properties at mean film temp. = (20 +100)/2 = 60C

  $k = 0.651$ W/m C

  $\rho = 985.4$ kg/m$^3$

  $\mu = 4.70 \times 10^{-4}$ kg/m s

  $c_p = 4184$ J/kg C

  $P_r = c_p\ \mu\ /\ k = 4184 \times 4.70 \times 10^{-4} / 0.651 = 3.02$

  $R_e = 985.4 \times 5 \times 0.1 / 4.70 \times 10^{-4} = 1.05 \times 10^6$

- Flat Plate correlation (external flow) with Turbulent Flow as $R_e > 5 \times 10^5$

  $N_u = [0.037\ (R_e)^{0.8} - 871] \times (P_r)^{0.33} \quad (R_e > 5x10^5)$

  $N_u = [0.037\ (1.05 \times 10^6)^{0.8} - 871] \times (3.02)^{0.33} = 2242$

- Heat Transfer Coefficient

  $h = N_u \times k\ /\ L = 2242 \times 0.651 / 0.1$

  $\mathbf{h = 14595\ W/m^2 C}$

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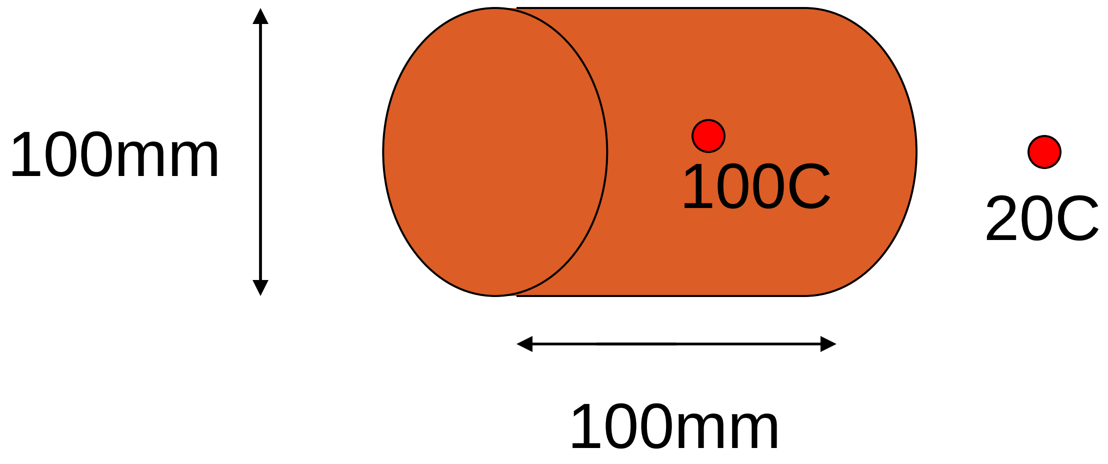
# Radiation




- Painted surface with emissivity = 0.9
- View factor = 1 as unblocked view of outside world by surface
- $h_R = \sigma \, \varepsilon_1 \, F_{1\text{-}2} \, (T_1^4 - T_2^4) \, / \, (T_1 - T_2)$

  $h_R = 5.669\text{x}10^{-8} \times 0.9 \times 1 \times [(100+273)^4 - (20+273)^4] \, / \, (100\text{-}20)$

  $\mathbf{h_R = 7.6\ W / m^2\, C}$

  – In this case the radiation is larger than that for natural convection (it was 7.0 $W/m^2\,C$)

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# Electrical Losses

- Most important that the losses and their distribution be known in order to obtain a good prediction of the temperatures throughout the machine
  - Some losses are associated with the electromagnetic design
    - Copper (can have high frequency proximity loss)
    - Iron (difficult to calculate accurately due to limitations of steel manufacturers data)
      - Loss data from material manufacturer may not have the same situation as real machine
        - Different waveforms
        - Stress of manufacture on laminations
        - Damage to interlamination insulation due to burs
        - Work underway to try and produce better loss data
        - Calibration using tests on actual machine best if available
    - Magnet (circumferential and axial segmentation to minimize)
      - Sm-Co 1-5 ($5 \times 10^{-8}$ ohm-m), Sm-Co 2-17 ($90 \times 10^{-8}$ ohm-m), Sintered Nd-Fe-B ($160 \times 10^{-8}$ ohm-m), Bonded Nd-Fe-B ($14000 \times 10^{-8}$ ohm-m)

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# Mechanical Losses

- Some losses are associated with the mechanical design
  - friction (bearings)
  - windage (air/liquid friction in gap/fan)
- Losses used in thermal model may be calculated and/or measured
- Losses are **inputs** in a thermal model and are not discussed in much detail in this tutorial

# Loss Variation with Temperature

- We can model the variation in copper loss with temperature directly in the thermal model by knowing the electrical resistance variation with temperature

$$\rho = \rho_{20}[1 + \alpha(T - 20)]\ \Omega\ m$$

where $\rho_{20} = 1.724 \times 10^{-8}\ \Omega m$

$\alpha = 0.00393\ /^\circ C$ for copper

  - 50° C rise gives 20% increase in resistance
  - 140° C rise gives 55% increase in resistance

- We can also account for the loss in flux due to magnet temperature rise directly in the thermal model
  - typical temperature coefficients of Br
    - Ferrite = -0.2 %/C (-20% flux for 100C rise, 1.56 x $I^2R$)
    - Sm-Co = -0.03 %/C (-3% flux for 100C rise, 1.06 x $I^2R$)
    - Nd-Fe-B = -0.11 %/C (-11% flux for 100C rise, 1.26 x $I^2R$)

- We can also account for the variation in windage loss using analytical windage formulations and the variation of viscosity with temperature

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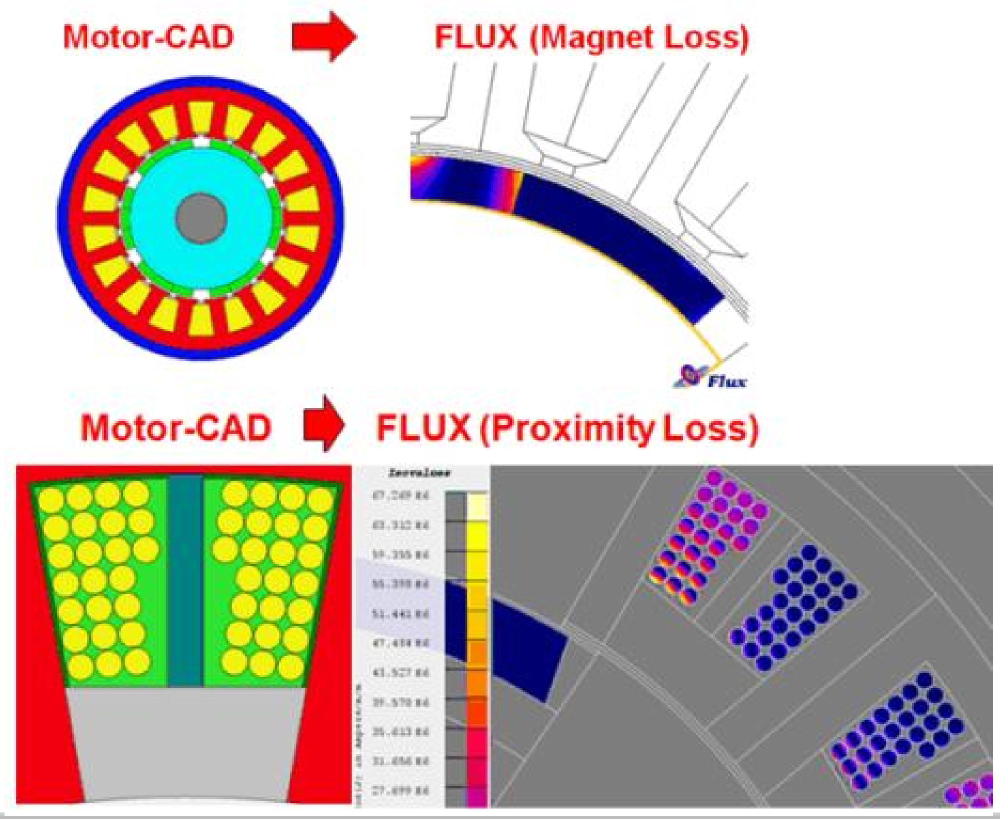
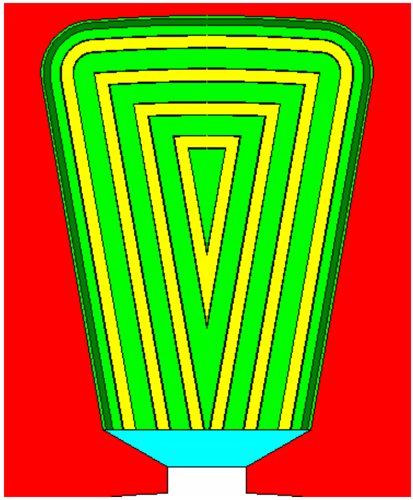
# Complex loss types and FEA

- More complex loss mechanisms benefit from FEA analysis
  - Automated links from lumped circuit thermal solver to the FEA code can speed up with this analysis
    - Often if a loss is going to take days/weeks to set up and calculate then it may be tempting to make estimates based on previous experience rather than by calculation (less accurate)

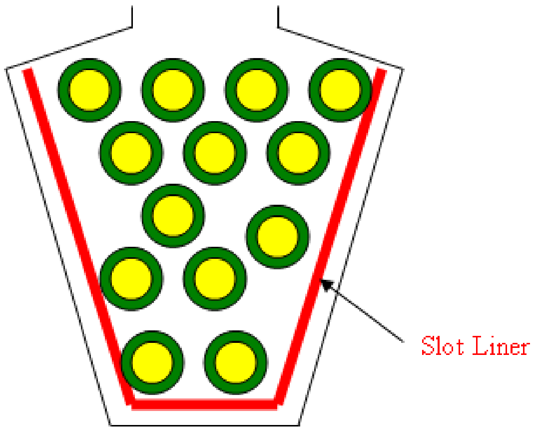


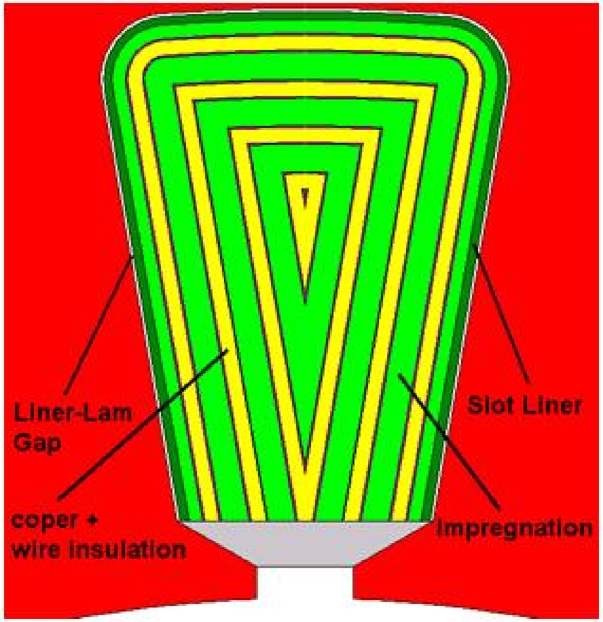
# Winding Heat Transfer

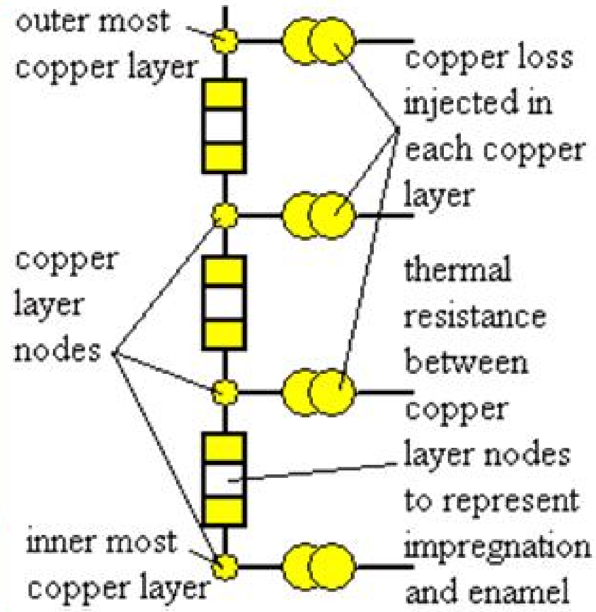
- The aim is to form a set of thermal resistances, power sources and thermal capacitances that model the thermal behavior of the winding
- Need to set area (A), length (L) and thermal conductivity (k) in R = L/kA for a complex slot shape holding a variety of components
- Various other modelling strategies have been developed to model the heat transfer within a winding but they usually are limited to one or more of the following:
  - A particular placement of conductors
  - A simple slot shape
  - A particular slot fill
- Such methods typically require test or FEA solutions to calibrate
- The layered winding model used by MDL has the advantage that:
  - It has some physical meaning and can be fully set up from slot geometry and known wire number and size



# Layered Winding Model




- Winding modelled using copper/insulation (liner, enamel and impregnation) layers
- To calculate a thermal resistance require layer thickness (L), layer area (A) and layer thermal conductivity (k) – $R = L/kA$, A is layer periphery x stack length
- Model has same quantity of components as in the actual machine
- The model gives details of temperature build up from slot wall to the hot-spot at the centre
- The copper loss is distributed between the layers according to their volume

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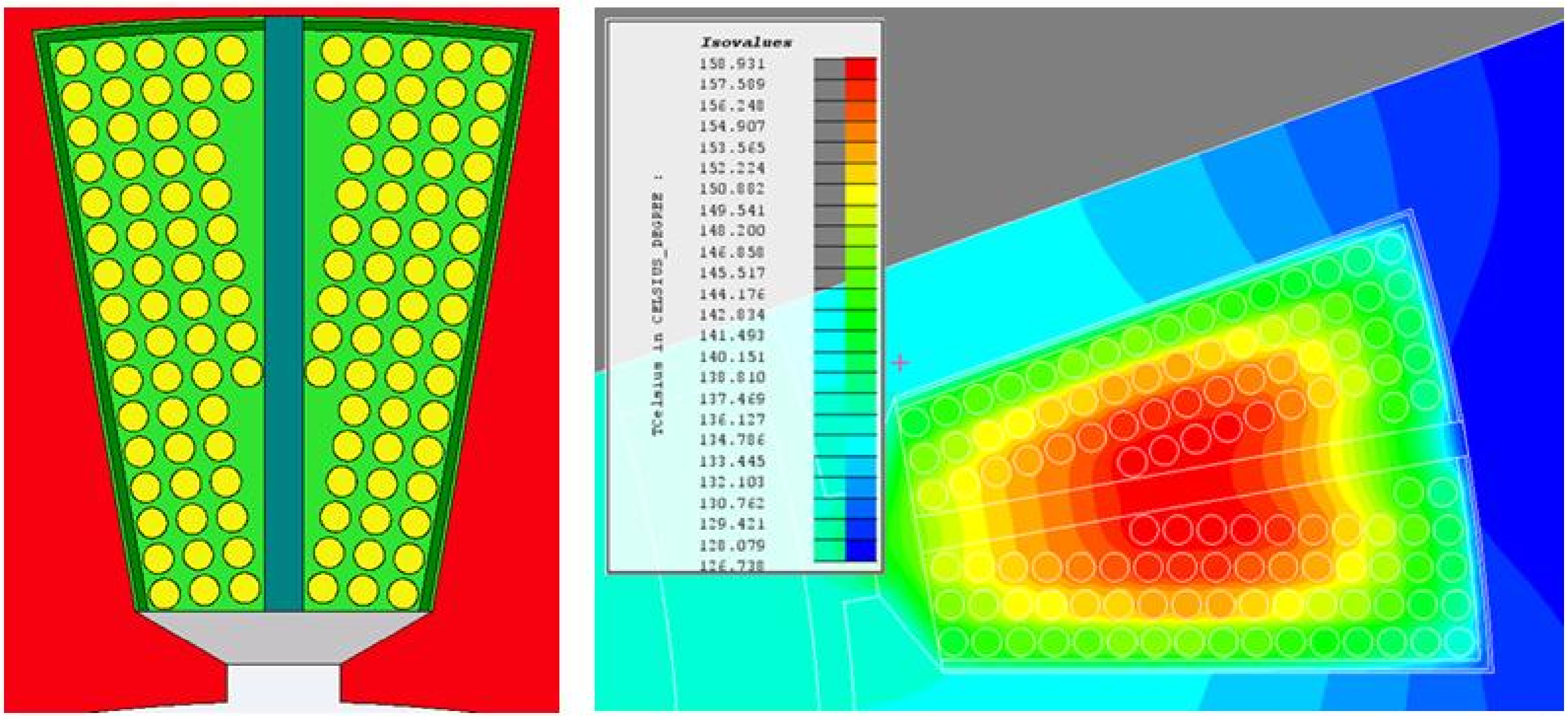
# Calibration of stranded winding models

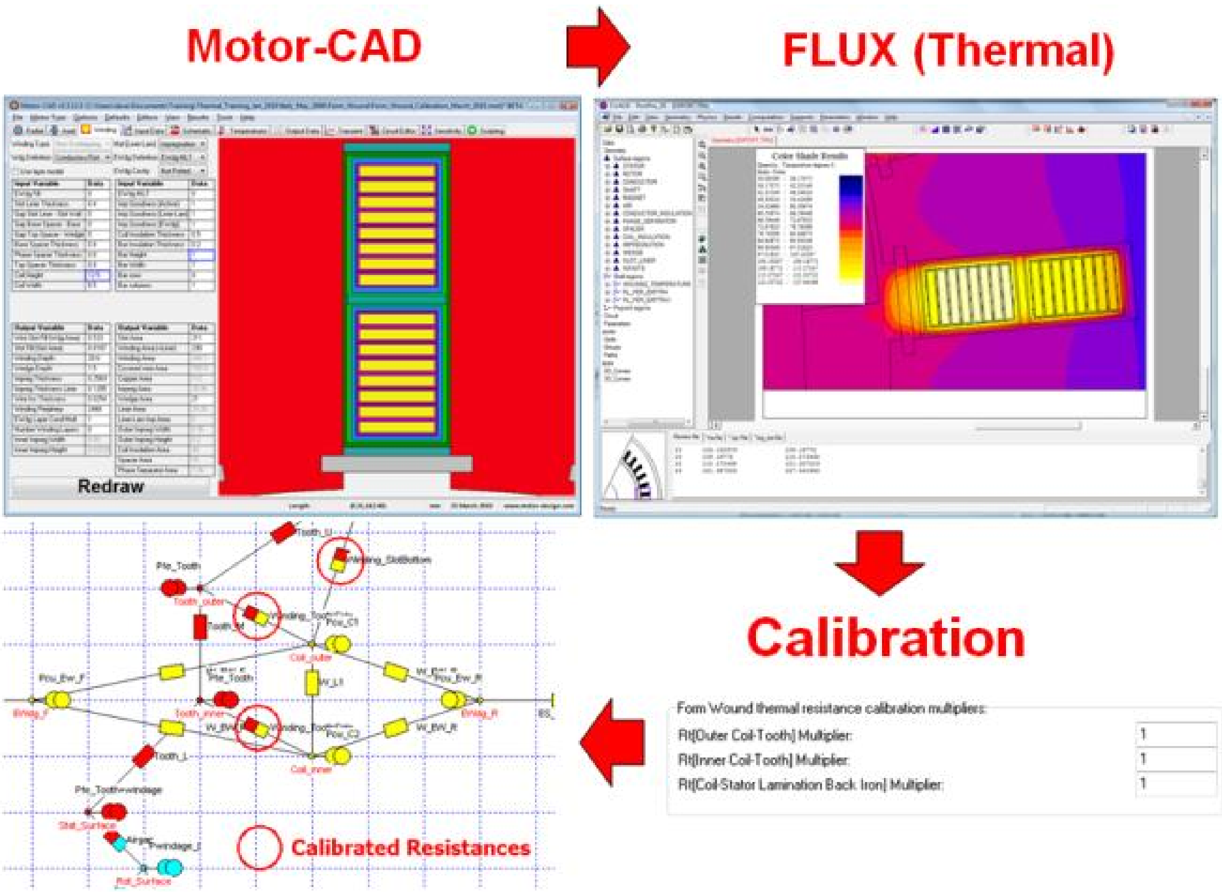
- If we can make an estimate of the placement of the conductors in a slot we can create a finite element thermal model and check the conduction heat transfer in the slot matches with the layered winding model

**Motor-CAD → FLUX (Thermal)**



# Calibration of form wound winding

- The form wound winding has rectangular wire with its associated layers of insulation
- Automated links to FEA for winding thermal resistance calibration are useful in this case

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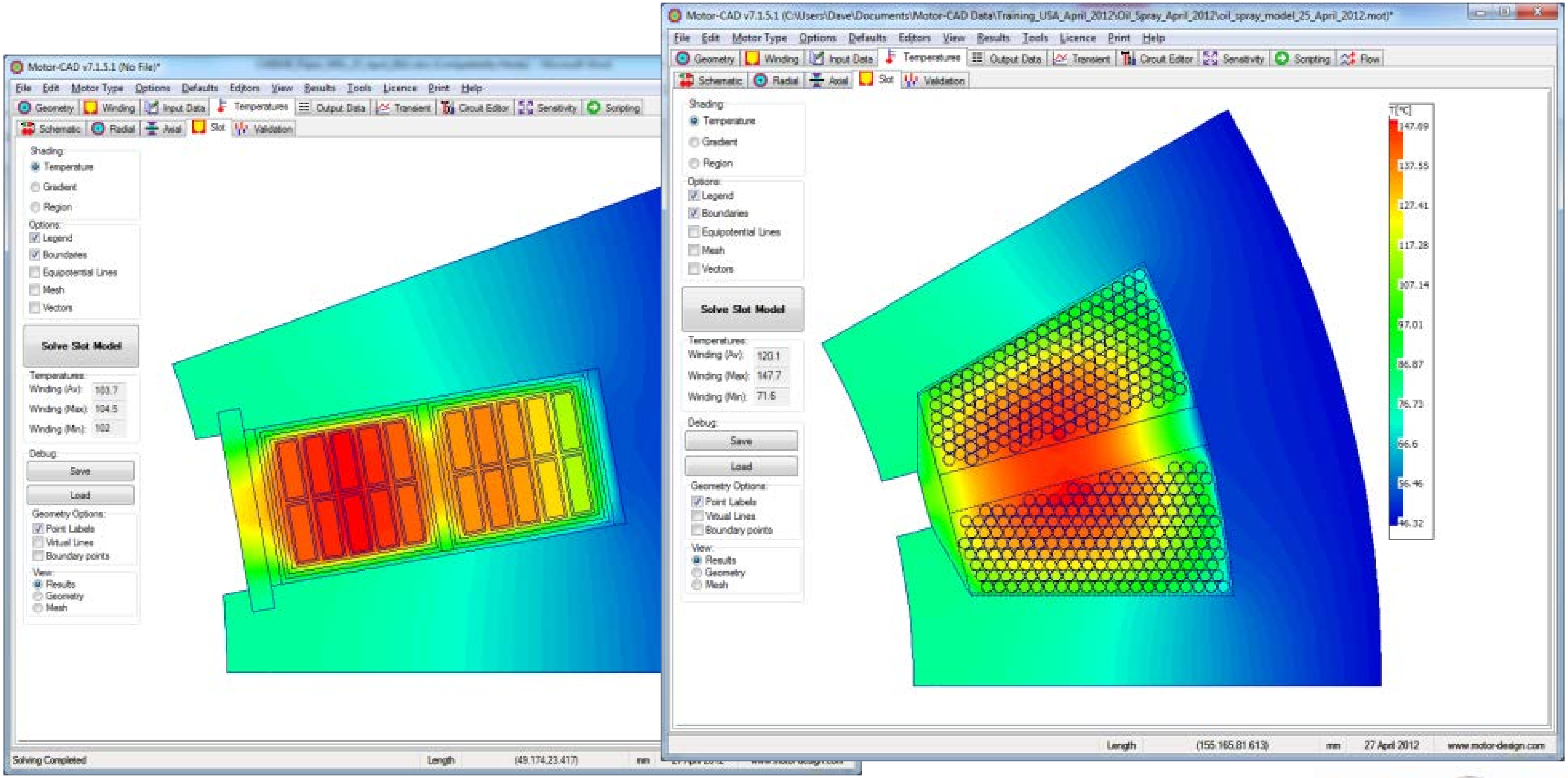
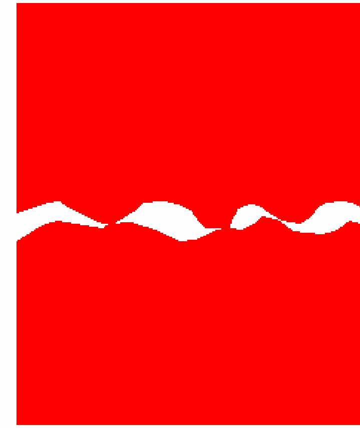
# Motor-CAD Integrated Thermal FEA Solver

- Just a few seconds to generate a mesh and calculate
- Help improve accuracy through calibration of analytical model



# Interface Thermal Resistance



- Touching surfaces have an effective thermal resistance:
  - Contact resistance due to imperfections in touching surfaces (uneven surfaces lead to voids) – contact area is typically small
- Two parallel resistances, conduction for touching spots & conduction/radiation for voids
- Can be crucial, especially in heavily loaded machines
- MDL model them using an effective airgap (R = Gap/kA)
  - Gap dependent upon materials and manufacturing processes used
  - Air assumed as interface fluid (can scale gap if other material)
  - Easy for non thermal expert to input as they soon get a feel for what is a good and what is a bad gap (physical dimension)
  - Alternative used by thermal experts is to input a value of Contact Resistance [m2.C/W] or Interfacial Conductance [W/C.m2]
    - More difficult for the non thermal expert to gain a feel for what is a good and what is a bad value
    - Values are displayed in Motor-CAD as useful for thermal experts

# Interface Thermal Resistance

- How rough is a surface? - published work gives a Mirror Finish = 0.0001mm and Rough Finish = 0.025mm (root-mean-square of deviations of a surface from a reference plane)
  - Average interface gap is not just a function of roughness
  - Softer materials have smaller average interface gaps as peaks are squashed and will fit together better
  - Complex function of material hardness, interface pressure, smoothness of the surfaces, air pressure, thermal expansion, etc.

# Interface Gaps – published data

- A book by Holman gives the following values for roughness and contact resistance with air as fluid medium

| materials | pressure | original roughness | resistance x $m^2$ |
|---|---|---|---|
| 416 ground Stainless | 3-25 atm | 0.0025 mm | 2.64 $m^2.C/W \times 10^4$ |
| 304 ground Stainless | 40-70 atm | 0.0011 mm | 5.28 $m^2.C/W \times 10^4$ |
| ground Aluminum | 12-25 atm | 0.0025 mm | 0.88 $m^2.C/W \times 10^4$ |
| ground Aluminum | 12-25 atm | 0.00025 mm | 0.18 $m^2.C/W \times 10^4$ |

- Lower value of contact resistance is better

- Conversion of Holman data to equivalent airgaps
  - $m^2.C/W$ x $W/m.C$ x 1000 = mm
  - k for air = 0.026 W/m/C

| materials | pressure | original roughness | effective gap |
|---|---|---|---|
| 416 ground Stainless | 3-25atm | 0.0025mm | 0.0069mm |
| 304 ground Stainless | 40-70atm | 0.0011mm | 0.0137mm |
| ground Aluminum | 12-25atm | 0.0025mm | 0.0023mm |
| ground Aluminum | 12-25atm | 0.00025mm | 0.0005mm |

- Materials used in electric machines have effective gap of between 0.0005mm and 0.014mm (average of around 0.007mm)
- softer material have smaller effective gap

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# Interface Gaps – published data

- A book by Mills gives values of interfacial conductances (at moderate pressure and usual finishes)
  - Stainless Steel – Stainless Steel 1700-3700 $W/m^2/C$
  - Aluminum – Aluminum 2200-12,000 $W/m^2/C$
  - Stainless Steel – Aluminum 3000-4500 $W/m^2/C$
  - Iron – Aluminum 4000-40,000 $W/m^2/C$

- Higher value of interfacial conductance is better
- Conversion of Mills data to equivalent airgaps
  - 1/(m2.C/W) x W/m/C x 1000 = mm
  - k for air = 0.026 W/m/C
    - Stainless Steel – Stainless Steel 0.0070-0.0153 mm
    - Aluminum – Aluminum 0.0022-0.0118 mm
    - Stainless Steel – Aluminum 0.0058-0.0087 mm
    - Iron – Aluminum 0.0006-0.0065 mm

  - Materials used in electric machines have effective gap of between 0.0006mm and 0.015mm (average of around 0.0075mm)
  - softer material have smaller effective gap

- Similar gaps to Holman data

# Stator Lamination – Housing

- A problem in gaining reliable results is that the laminated surface roughness is dependent upon the manufacturing processes used
- Larger interface gaps than usual are typical due to laminated surface
- Best to calibrate to suit motors, materials and manufacturing practices used

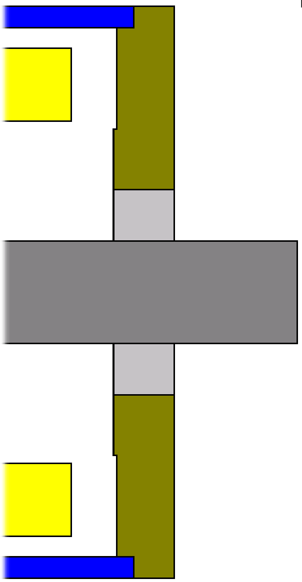
| Motor rated power[kW] | Interface Gap [mm] |
|---|---|
| 4.0 (MA112M4) | 0.042 |
| 7.5 (MA132M4) | 0.076 |
| 15.0 (MA160L4 ) | 0.077 |
| 30.0 (MA200L4) | 0.016 |
| 55.0 (MA250M4) | 0.037 |


Data from Boglietti - Politecnico di Torino, Italy

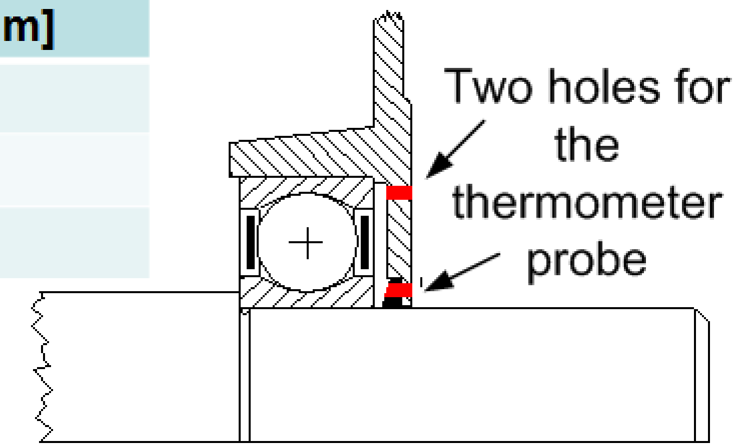
- Other Practical Results
  - 142mm diameter BPM servo motor - 0.01mm (cast aluminium housing)
  - 335mm & 500mm diameter IM's - around 0.015mm (cast iron frames)
  - 128mm diameter IM - around 0.02mm (aluminium housing)

# Bearings

- Bearings are a complex composite structure
  - Inner and outer race, balls, grease
  - Can be modeled by an effective interface gap
    - Problem is to know what this gap should be
      - Calibration testing has been done Boglietti showing a typical effective interface gap between 0.2mm and 0.4mm

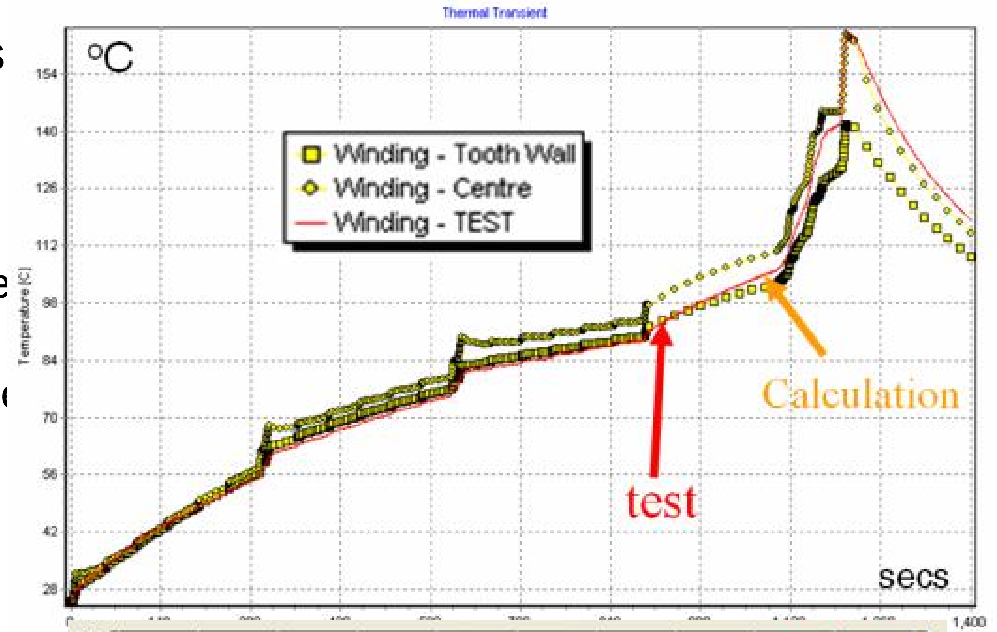


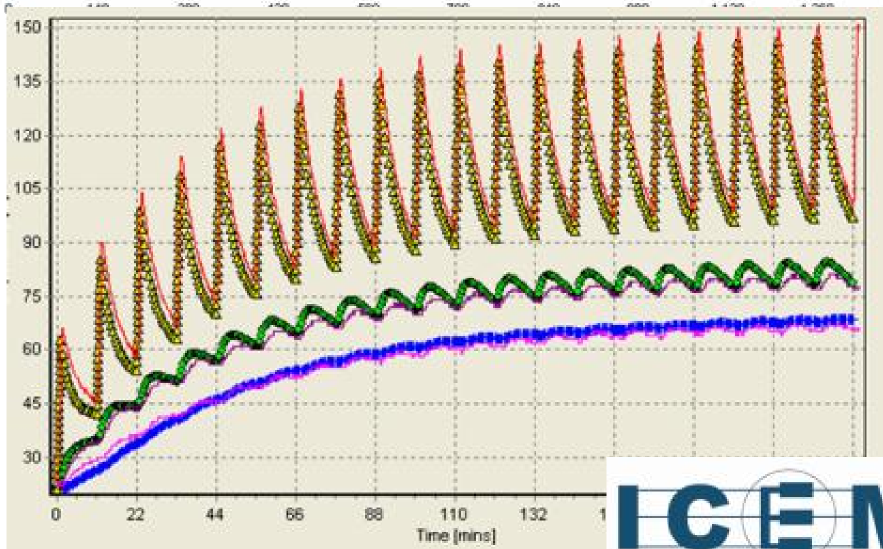
| Motor | Inner diameter [mm] | Outer diameter [mm] | Equivalent interface gap [mm] |
|---|---|---|---|
| 4 kW | 30 | 19 | 0.35 |
| 7.5 kW | 40 | 18 | 0.23 |
| 15 kW | 45 | 25 | 0.40 |



# Dealing with difficult areas of thermal analysis

- Many manufacturing uncertainties such as:
  - Goodness of effective interface between stator and housing
  - How well the winding is impregnate or potted
  - Leakage of air from open fin chann blown over machines
  - Cooling of the internal parts in a TENV and TEFC machine
  - Heat transfer through the bearings
  - etc.
- Test program with leading universities over the past 15 years to help develop data to help with such problems
  - Set default parameters in Motor-CAD giving good level of accuracy without the user having done extensive calibration using testing of their own machines



# Improved Accuracy Using Calibration

- It has been seen that there are many complex issues (some manufacturing issues) when setting up an accurate thermal model for a motor
- Calibration of models helps to increase accuracy
  - Also useful to gain an insight of how your machine compares to other machines in terms of manufacturing goodness and quality of design
  - Often the thermal model identifies a problem with the electromagnetic calculation, i.e. hot rotor temperature showing higher than expected magnet loss, etc.
- Various specialist tests can be made on a machine to help with model calibration
  - Fixed DC Current (known loss) into stator winding (all phases in series) with various temperatures measured
    - Help calibrate interface gaps and winding impregnation, etc.
- Sensitivity analysis is also a good way to identify the main design issues so concentration can be given to those

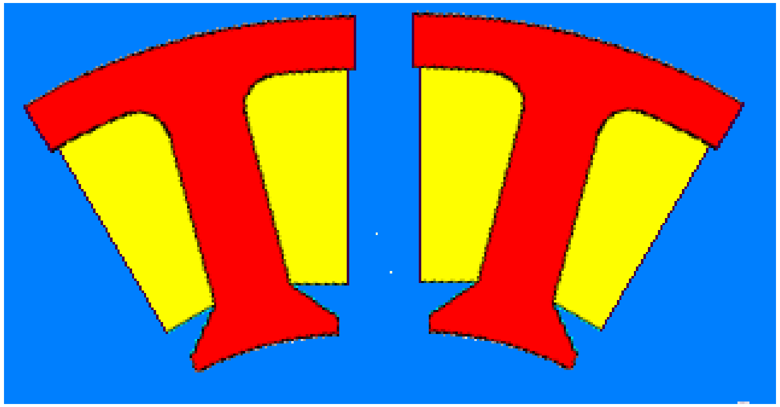
# Todays Topics

- Motor Design Ltd (MDL)
- Need for improved tools for thermal analysis of electric machines
- Important Issues in Thermal Analysis of Electric Motors
- **Examples of use of thermal analysis to optimise electric machine designs**

# Previous Thermal Project Examples

- Most projects details are provided by MDL customers
- Only examples are shown that have permission to publish from the user
  - Most have technical papers associated with them that can be examined for more details

# Segmented Motor Miniaturization



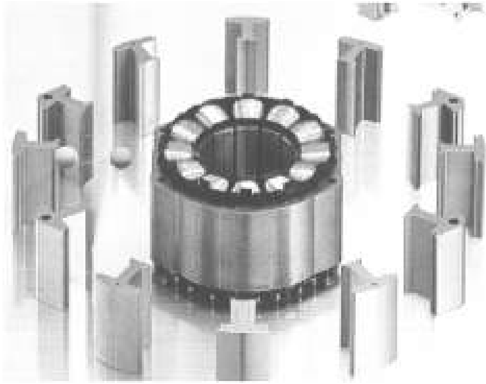
Papers published in 2001

Existing Motor:

- 50mm active length
- 130mm long housing
- traditional lamination
- overlapping winding

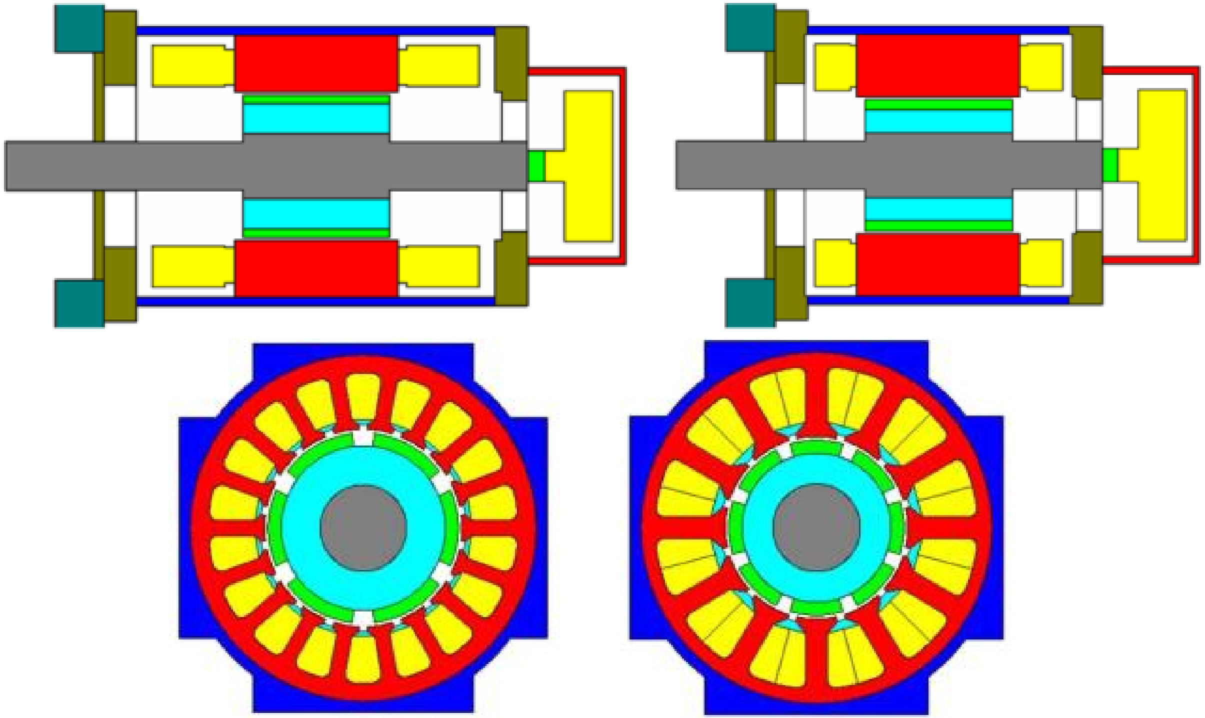
New Motor:

- 50mm active length
- 100mm long housing
- 34% more torque for same temperature rise
- segmented lamination
- non-overlapping winding



- In order to optimize the new design an iterative mix of electromagnetic and thermal analysis was performed

# Segmented Motor Miniaturization



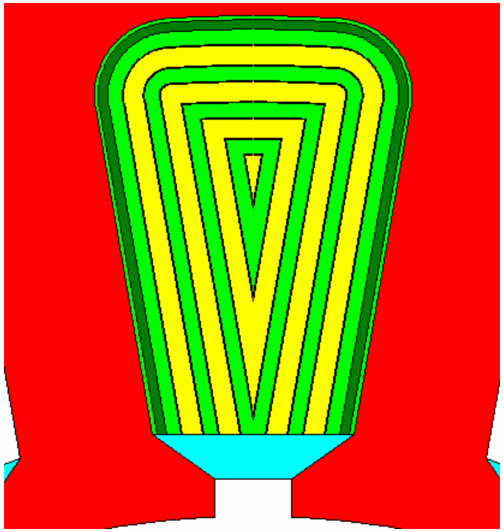
Old motor

- inserted mush winding
- 54% slot-fill
- 80mm diameter
- 18-slots, 6-poles
- large end windings which overlap each other

New Motor

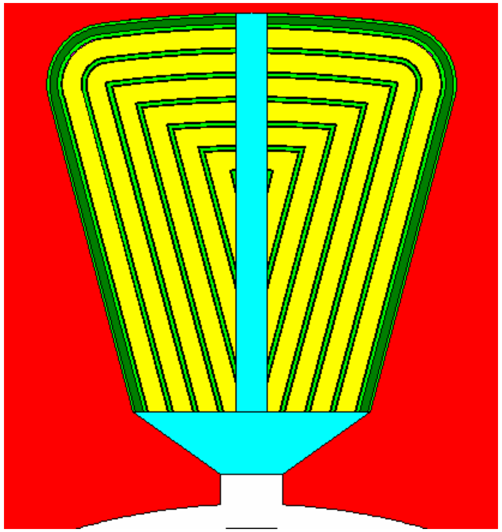
- precision bobbin wound
- 82% slot-fill
- 80mm diameter
- 12-slots, 8-poles
- short end windings that are non-overlapping

# Segmented Motor Miniaturization



Traditional Winding

- inserted mush winding
- 54% slot-fill



Concentrated Winding

- precision bobbin wound
- 82% slot-fill

Mixed EM/Thermal design:

- Iron losses have an easier path to the housing
- Optimum thermal design requires correct balance between copper & iron losses
  - complex function of size, speed, materials, etc.

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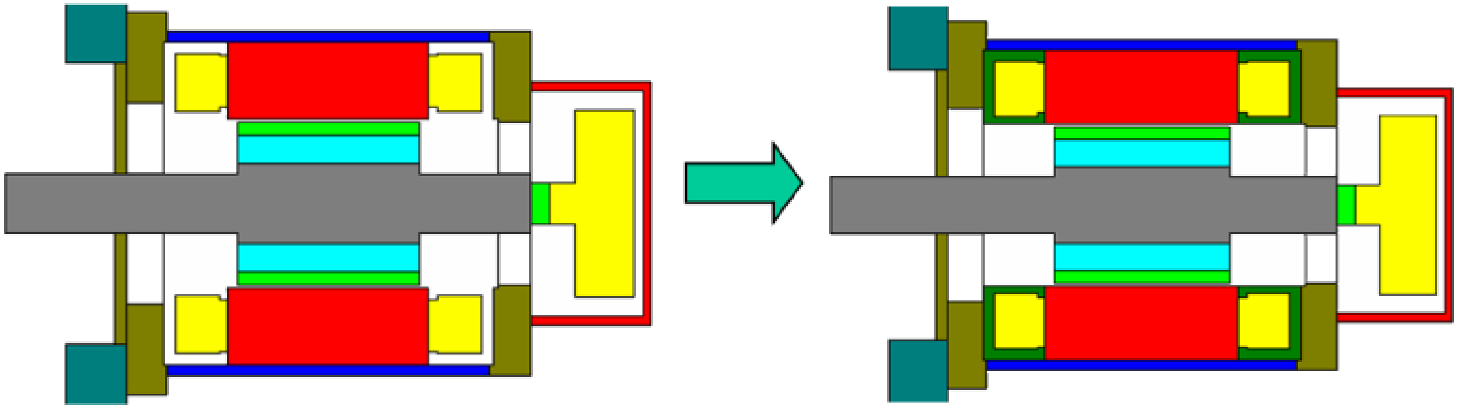
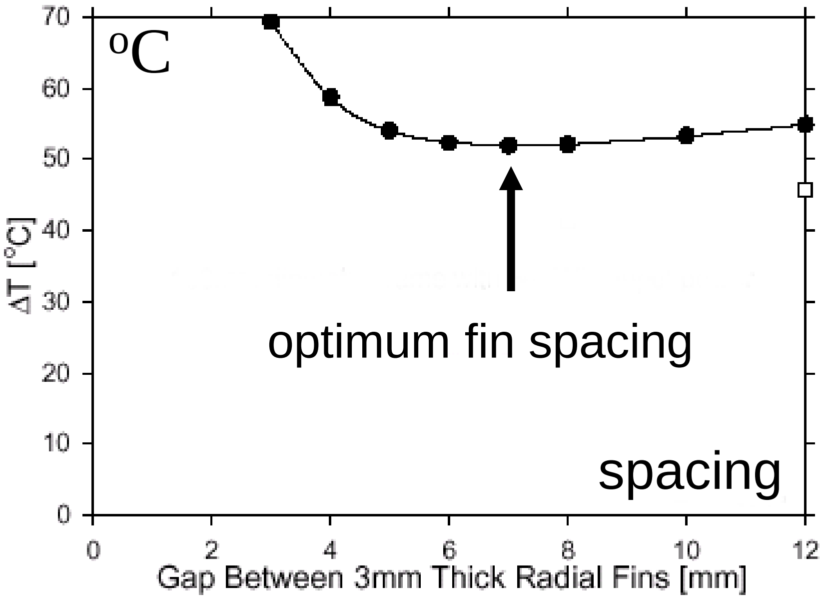
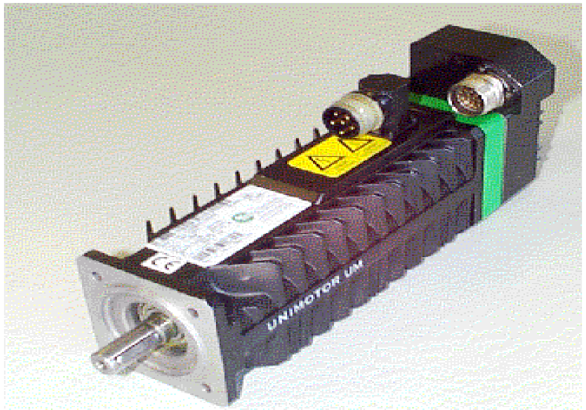
# Improved winding insulation system

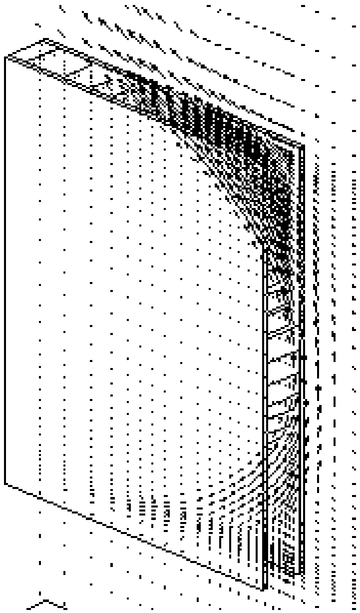


- Developed improved winding insulation in new designs
  - New potting/impregnation materials with k = 1W/m/C
    - previous materials have k = 0.2W/m/C
    - above designs show 6%-8% reduction in temperature
  - Above potted end-winding design shows a 15% reduced temperature compared to non-potted design
  - Vacuum impregnation can eliminate air pockets
    - above design shows 9% decrease in temperature in perfectly impregnated motor compared to old impregnation system

# Improved housing for TENV motor

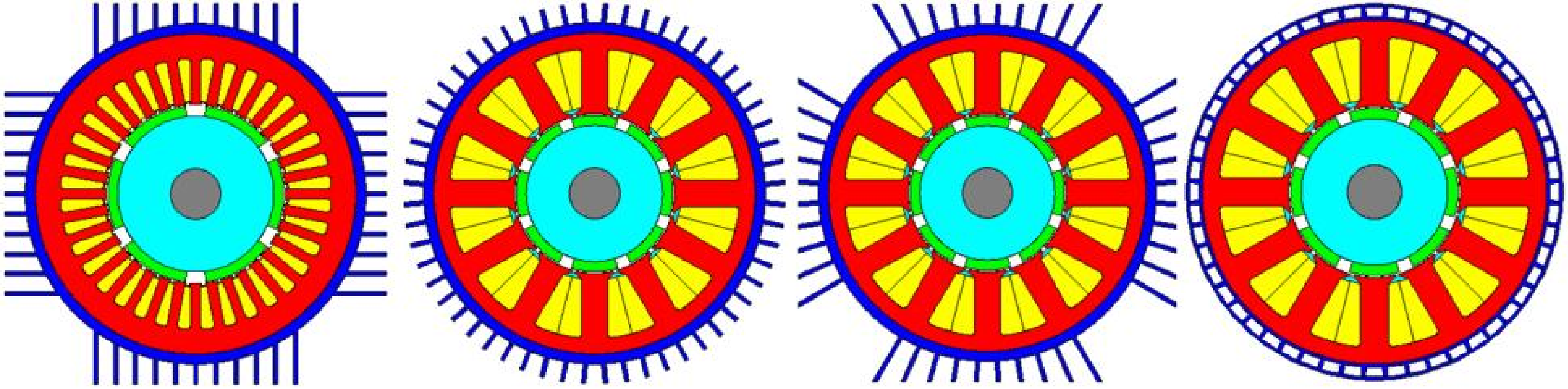


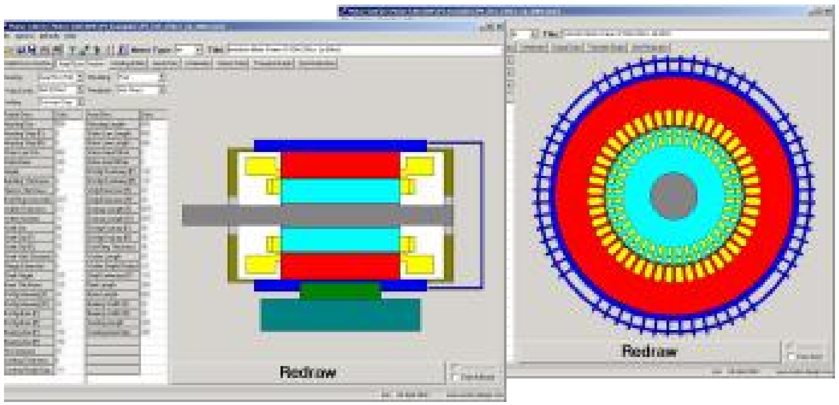
- Motor-CAD analytical formulations for convection used to optimise the fin spacing
  - small fin spacing has large surface area but reduced air flow
  - large fin spacing has maximum air flow but reduced surface area
- Accuracy validated using tests on series of rectangular and circular discs and internal heater
- Also did CFD which gave same optimum but took a lot longer to set up the model and calculate

# Axial Cooling Fin Optimization

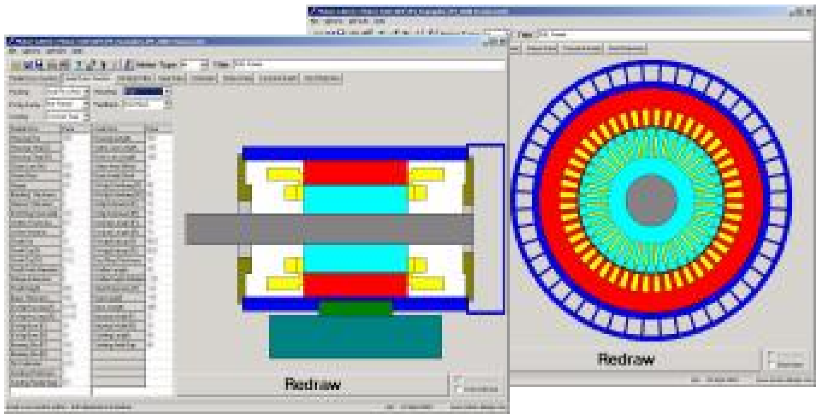


- motors with shaft mounted fan or blower unit
  - large thermal benefits possible with correct fin design
  - large selection of fin types available in Motor-CAD

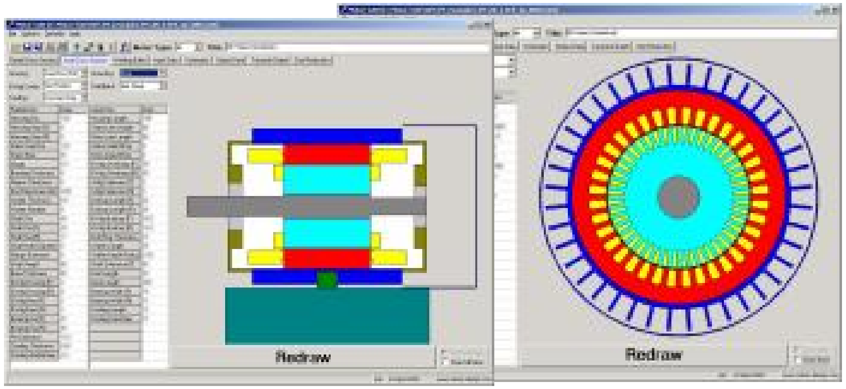
# Selection of TEFC machines where axial fins optimized using thermal analysis




315mm Shaft Height, Cast Housing




200mm Shaft Height, Cast Housing




80mm Shaft Height, Aluminium Housing

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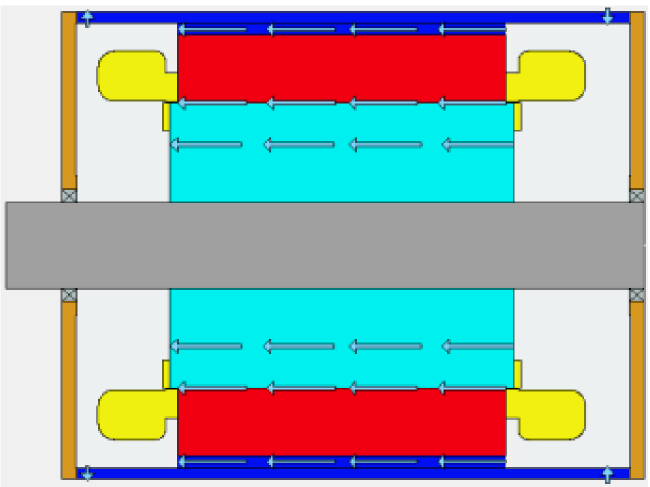
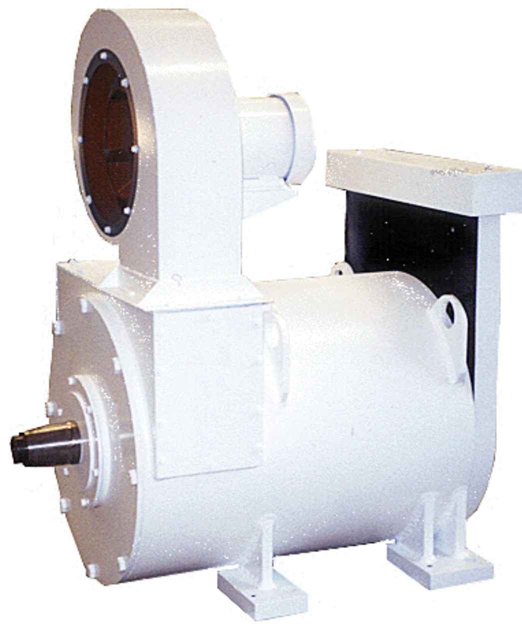
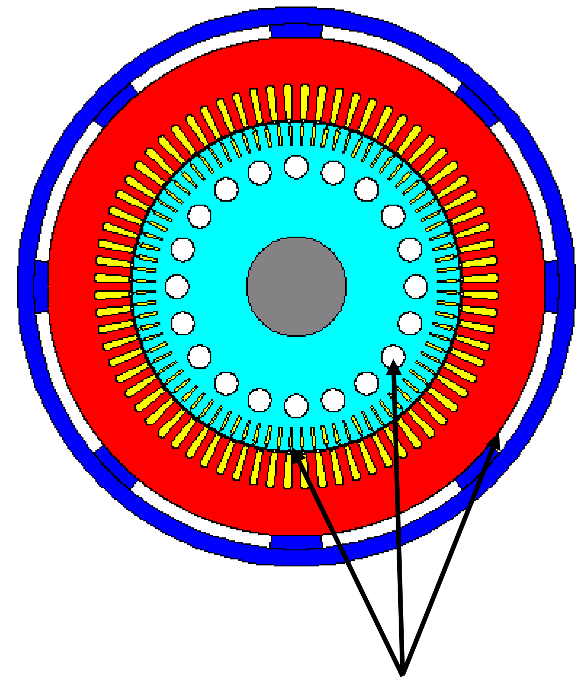
# Through Ventilation Motor







ventilation paths

1150hp IM

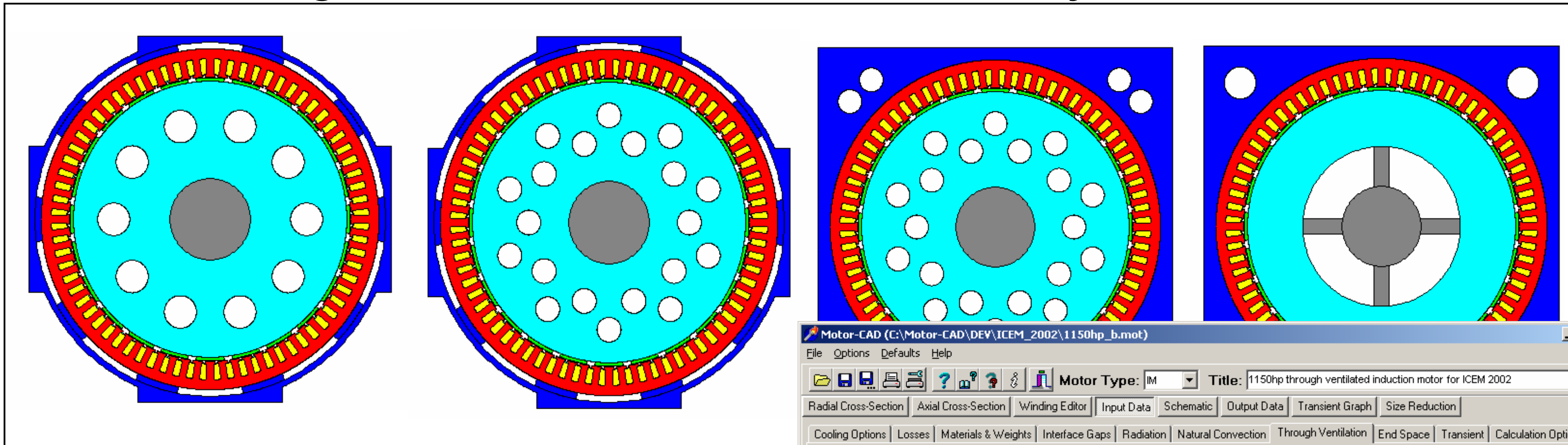
Tw(test) = 157ºC

Tw(calc) = 159ºC

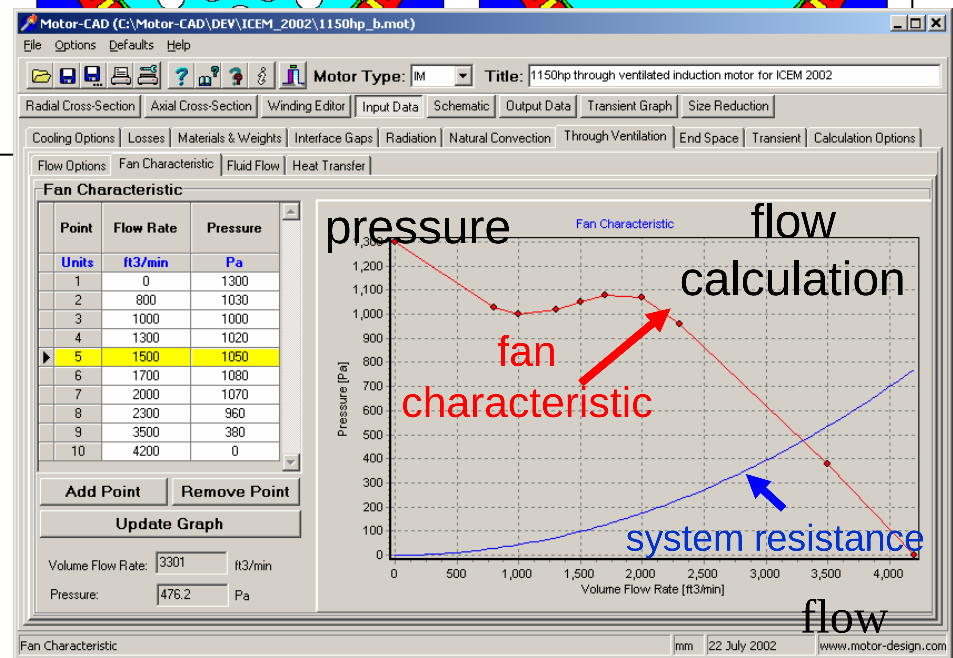
- Through ventilated motor optimised using a mix of heat transfer and flow network analysis
- More details in paper from ICEM 2002

# Through Ventilation

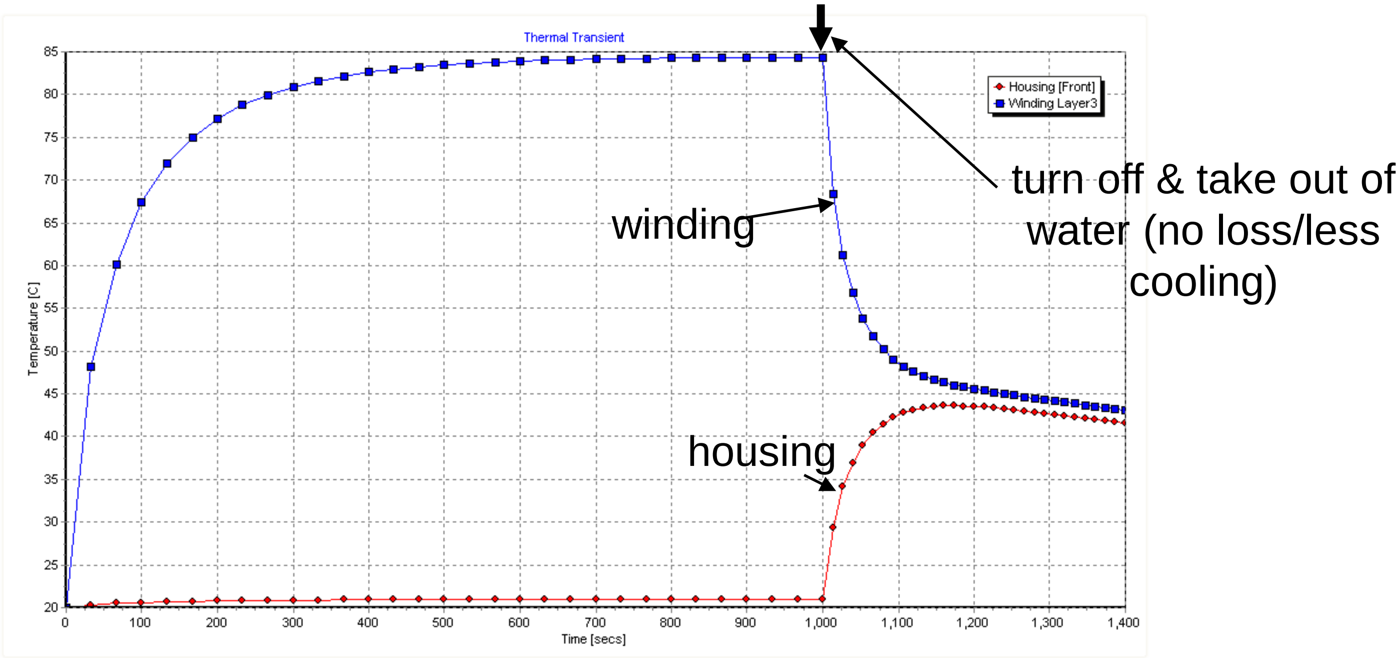
- Few of the many ducting systems available for use in through ventilation thermal analysis



- flow circuit automatically calculated
- User can input the fan characteristic and the system flow resistance is calculated

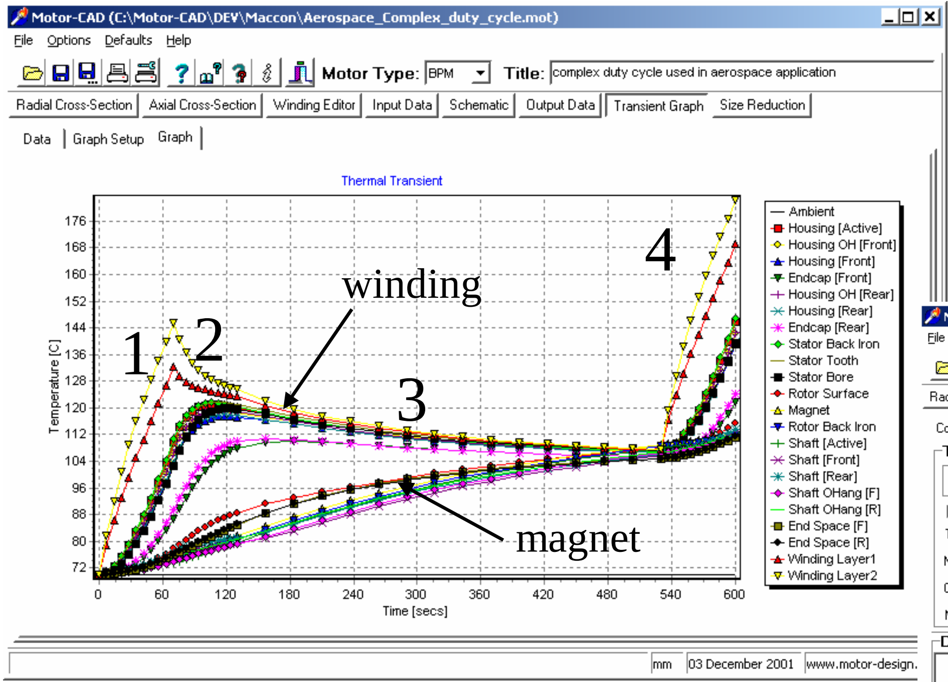


# Underwater Camera Motor

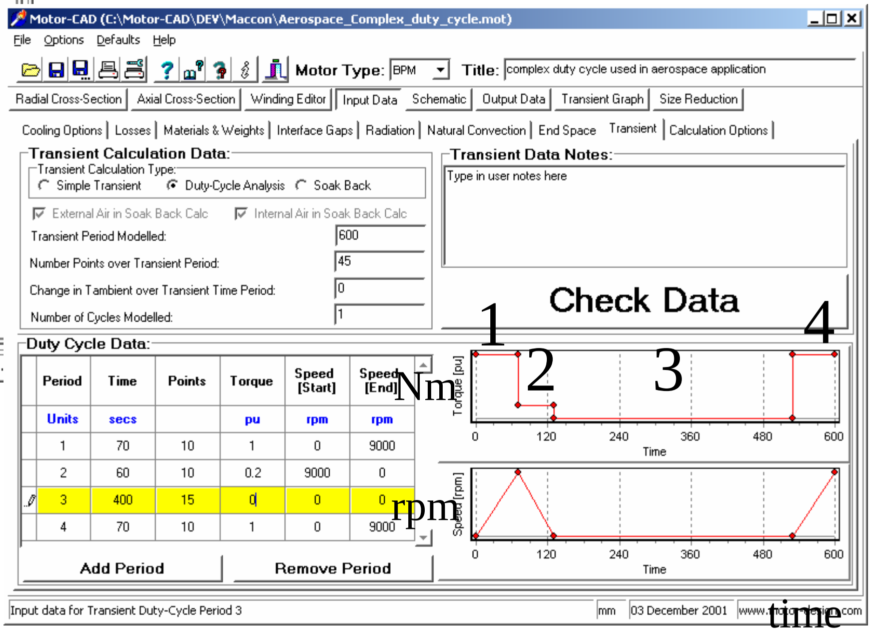



- Analysis carried out on motors driving propellers on a small submersible craft fitted with a camera
- Analysis to make sure that the housing temperature did not get too hot when the motor turned off and removed from the water
  - Losses now zero and winding cools down but this heats up the housing which is now not liquid cooled

# Minimization of Motor Size/Weight Aerospace Duty Cycle Analysis




duty-cycle analysis on an aerospace application with a short term load requirement




the motor needed to operate on the duty cycles and just get to 180C

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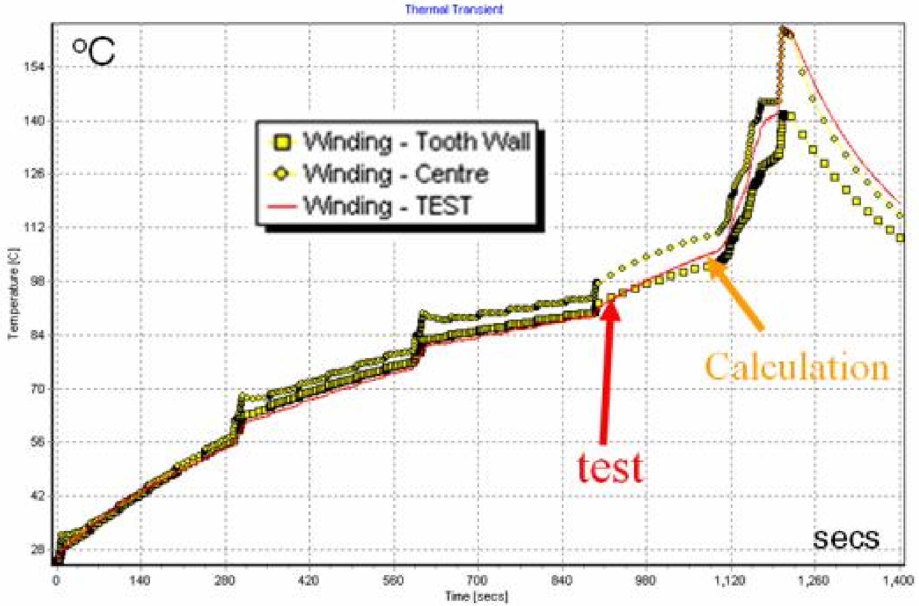
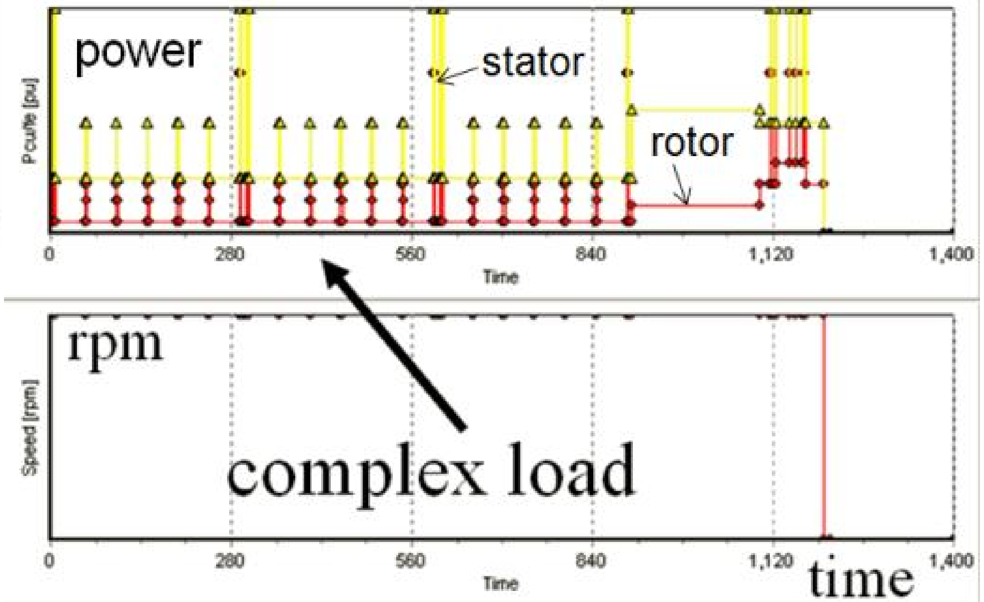
# Validation of Duty Cycle Thermal Transient Analysis (Aerospace)




- the complex load modelled in Motor-CAD is shown below:




- excellent agreement between the calculated thermal response and measured temperatures

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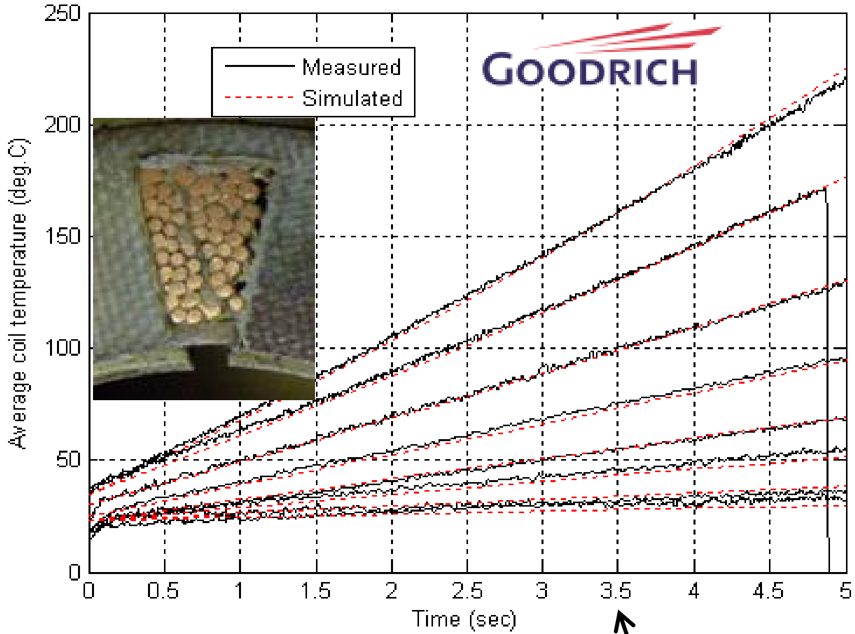
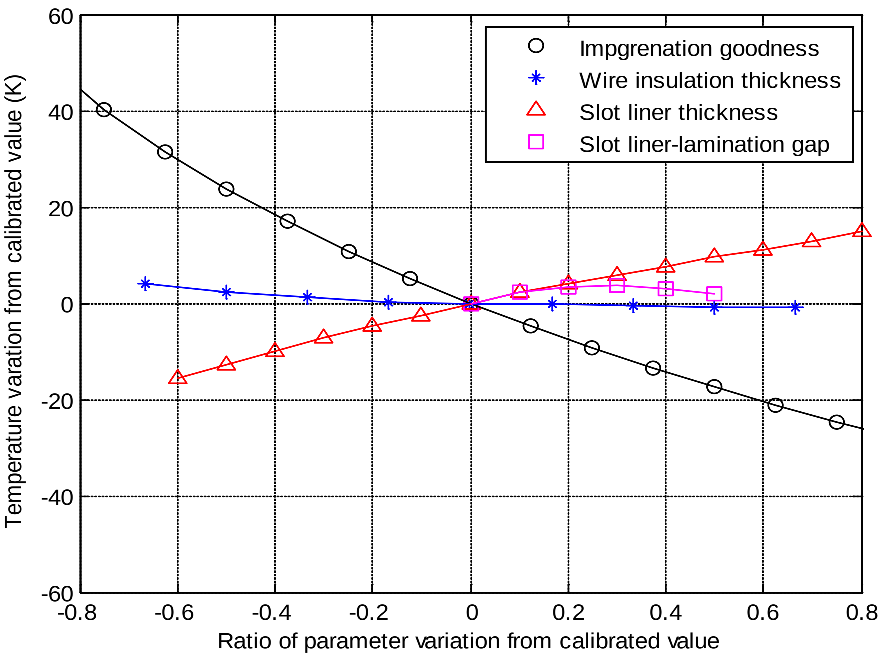
# Thermal Modeling of a Short-Duty Motor

- Motor designed to have minimum size and weight for a high performance short-duty cycle application.
- Optimization of thermal performance is critical for minimized weight and size
- Motor-CAD used for transient analysis and size optimisation
- Excellent agreement with test data when prototype built

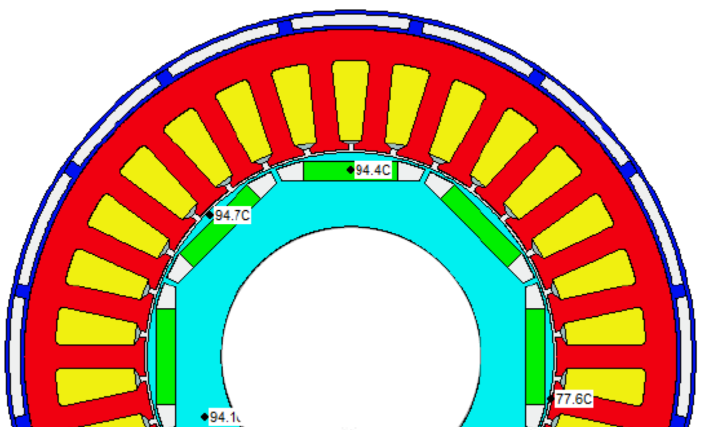



Winding temperature rise with DC currents (0.2 to 1.1 p.u.) - Test & Motor-CAD

# IPM Traction Motor

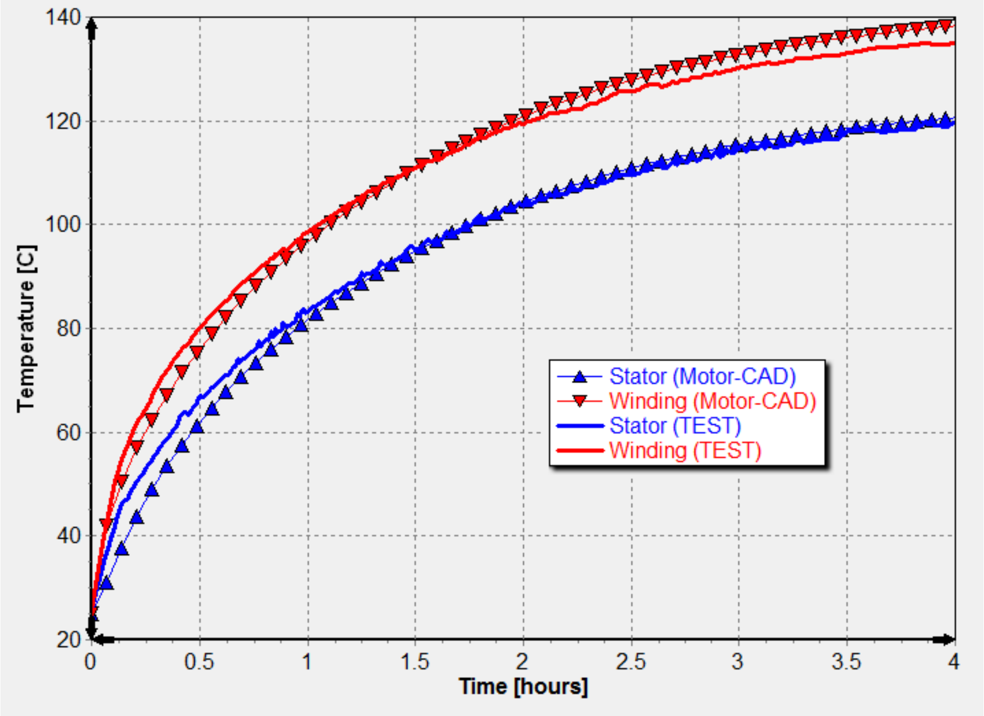
- Optimisation of water jacket for a traction motor
- Excellent agreement with test data when prototype built showing level of confidence in default values for manufacturing issues like the stator-lamination interface gap

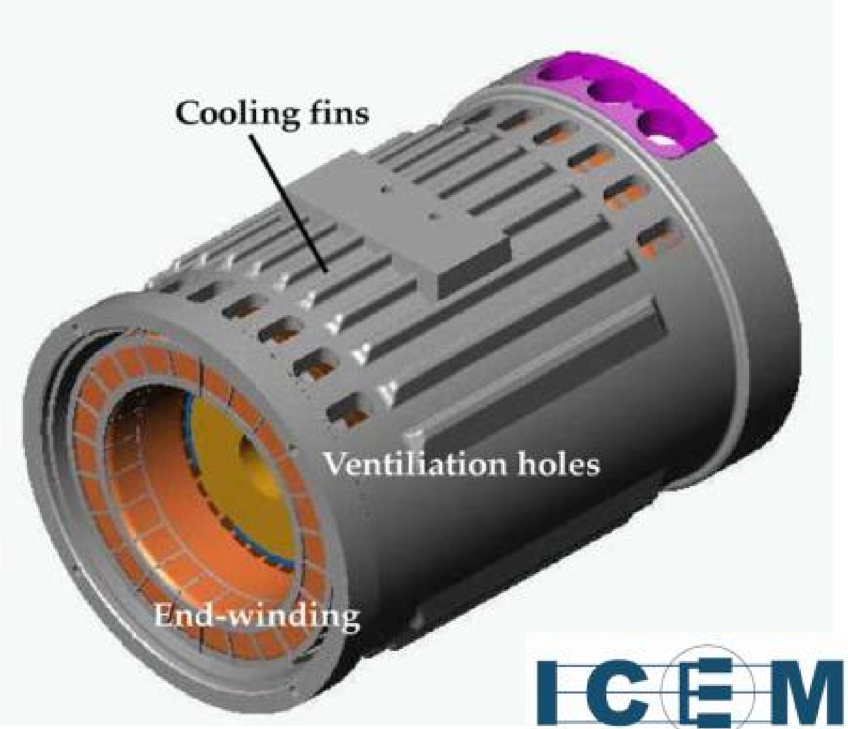



| | Casing | Stator tooth | | Winding | | Rotor magnets |
|---|---|---|---|---|---|---|
| | | Mid | Tip | Max. | Av. | |
| Thermal Model [°C] | 50 | 77 | 78 | 104 | 99 | 95 |
| Measurement [°C] | 50 | 78 | 82 | 101 | 96 | 100 |

# BPM Traction Motor

- Motor-CAD used to optimise cooling of traction motor
- Motor had open endcap with local convection cooling of end-windings
- Excellent accuracy shown with fast calculation times

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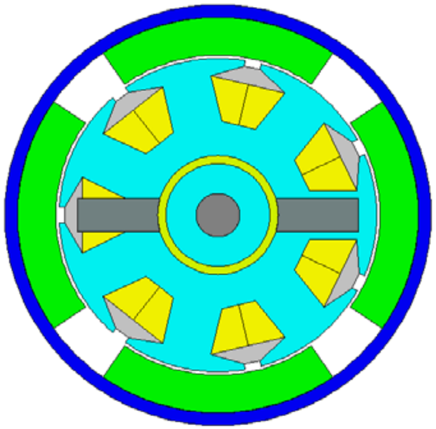
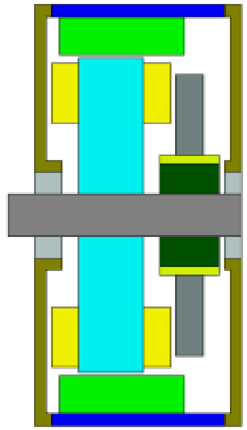
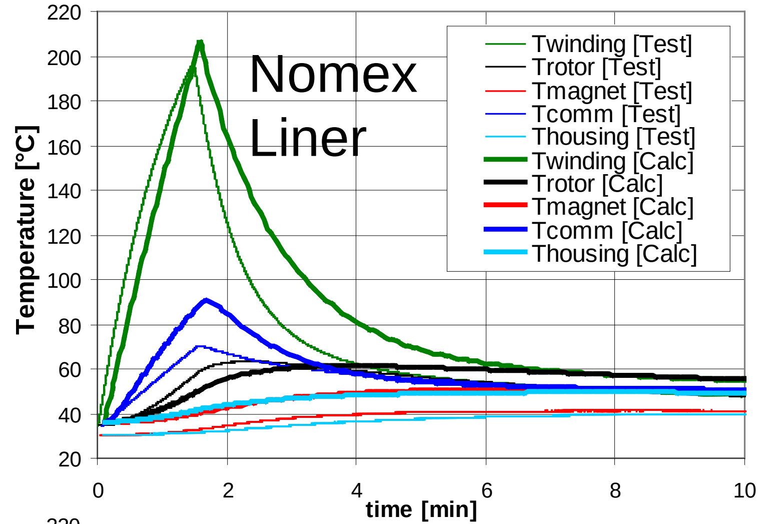
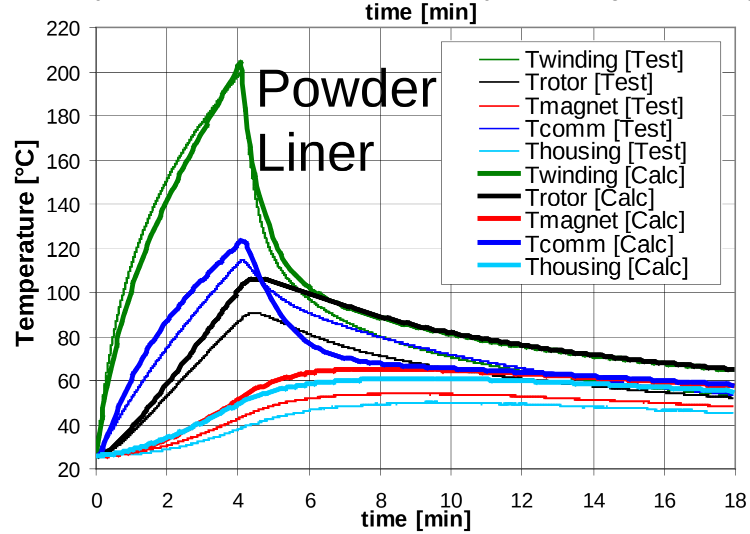
# Automotive PMDC - Improved Winding Insulation System



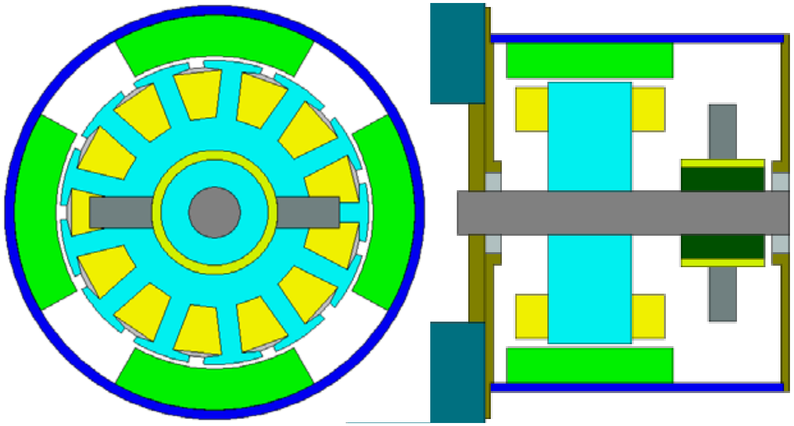


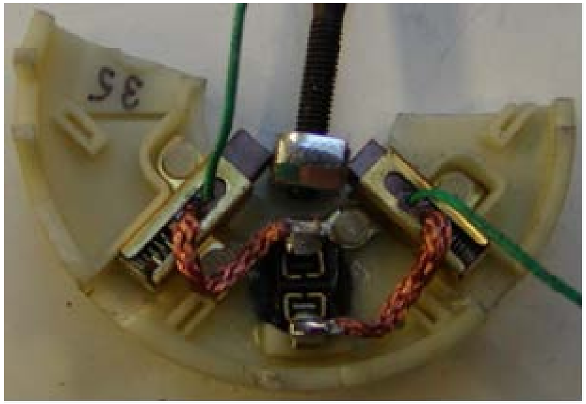
- ICEM 2008 – electro-hydraulic brake
- Optimisated impregnation process and slot liner to allow a longer operation time
- Two transients shown
  - Same load of 20A locked rotor
  - One has Nomex liner and the other a powder liner
  - Powder liner allows 4 minute load rather than 1.8 minutes
- Measured and simulated results shown

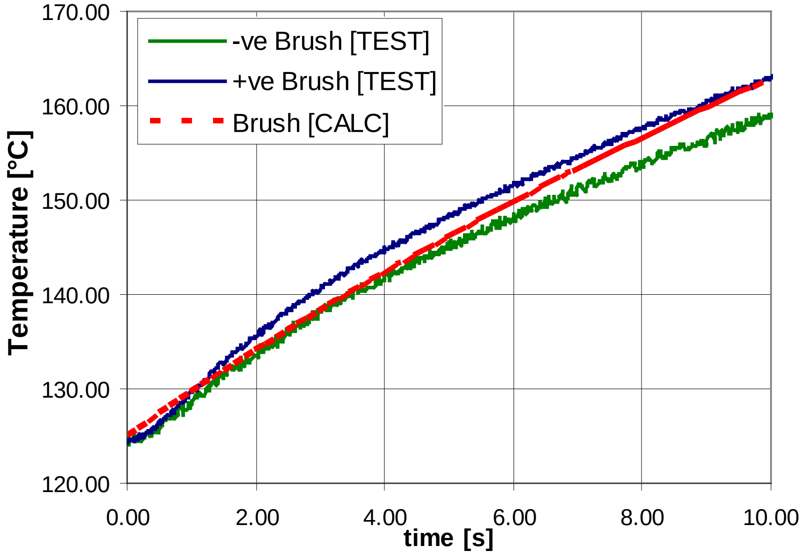


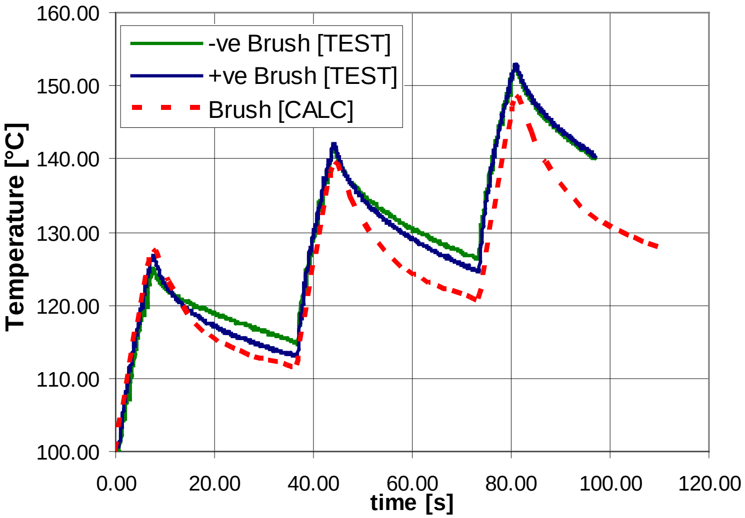



# Automotive PMDC Brush Model





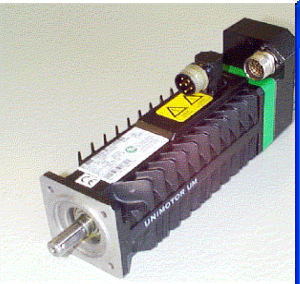


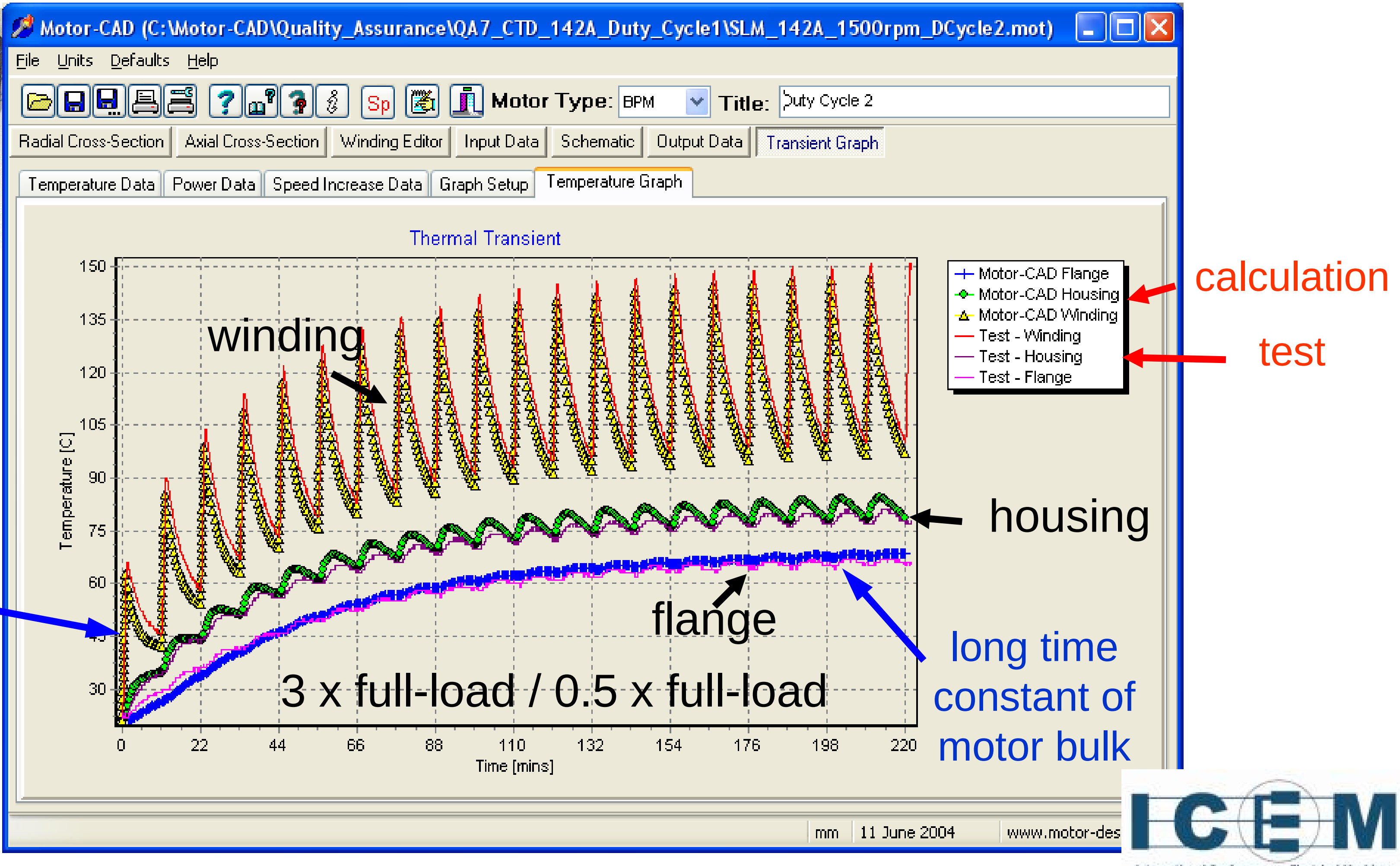




- Improved brush model developed with aid of extensive testing

# Servo Motor Duty Cycle Analysis

- Prediction of a motor thermal performance on a duty cycle load is important to match the motor to the load
  - excellent agreement between the prediction and test data shown



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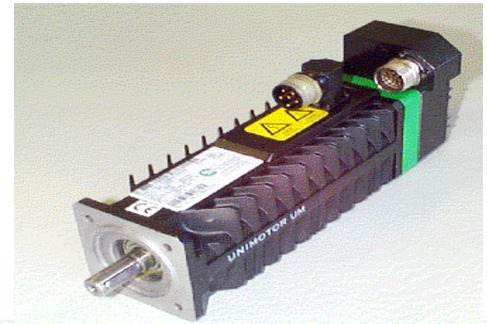
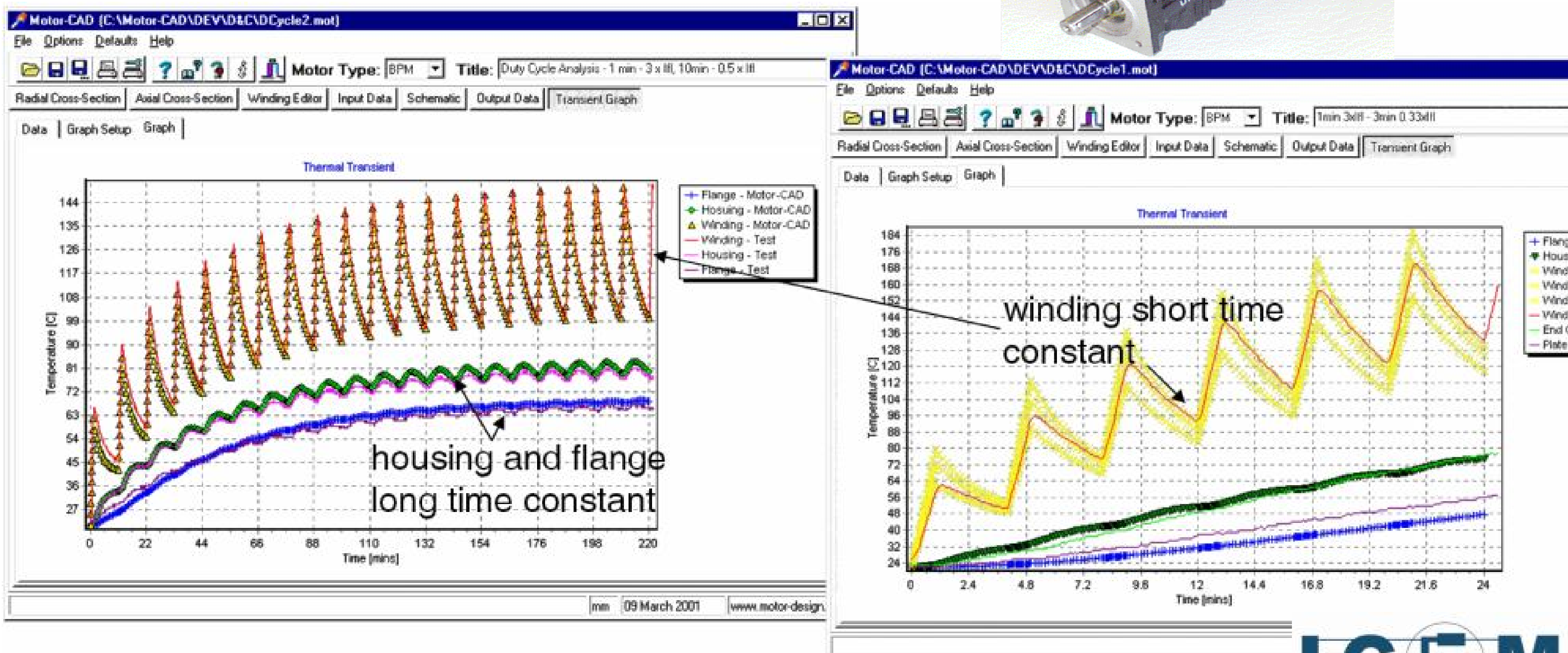
# Servo Motor Duty Cycle Analysis

- Range of tests with different duty cycles used to proved model
  - symbols = Motor-CAD model
  - lines = test data



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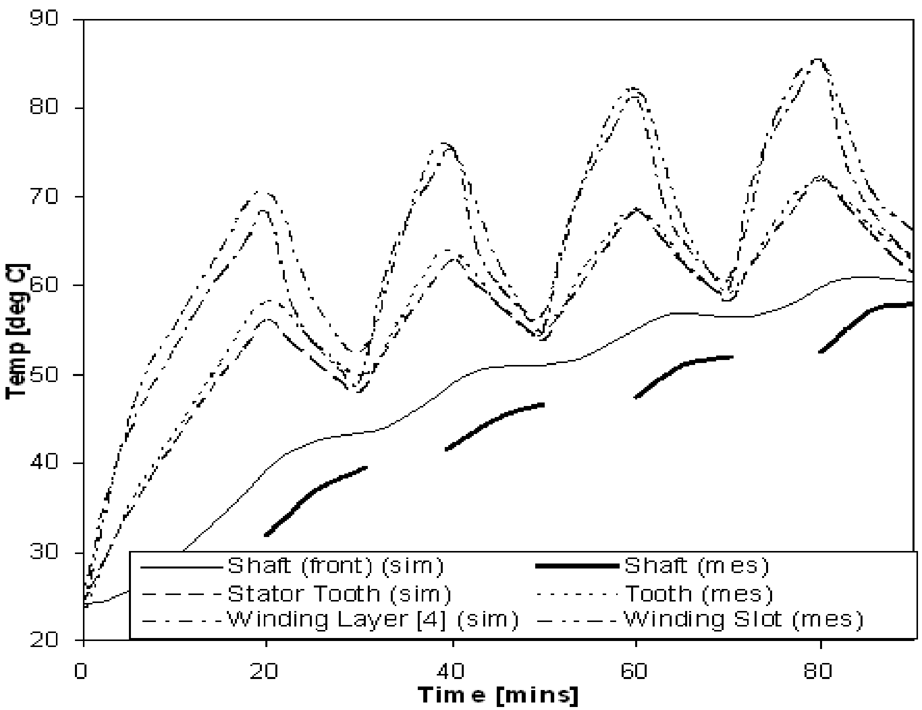
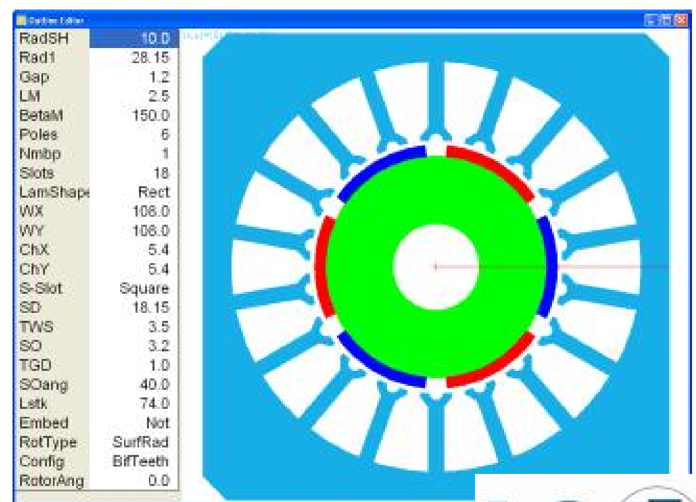
# Servo Motor Duty Cycle Analysis

- Again for another servo motor design excellent agreement between test and calculation




| Temp location | Measured (deg C) | Simulated (deg C) | Error (%) |
|---|---|---|---|
| Winding Hotspot | 144.3 | 148.1 | 2.6 |
| Housing | 117 | 117 | 0.0 |
| Tooth | 127.9 | 129.4 | 1.0 |
| End Winding | 134.1 | 126.8 | -5.5 |
| End Cap | 107.4 | 107.9 | 0.5 |

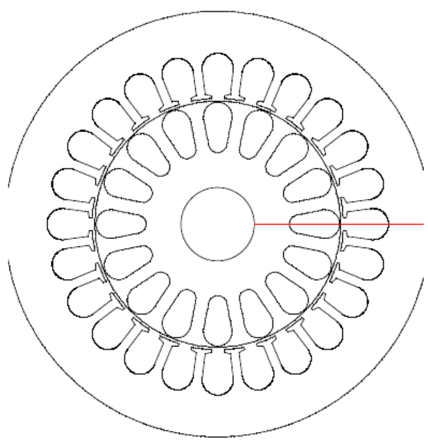
Comparison between measured and simulated steady-state temperatures

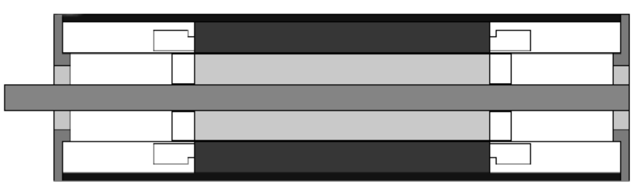


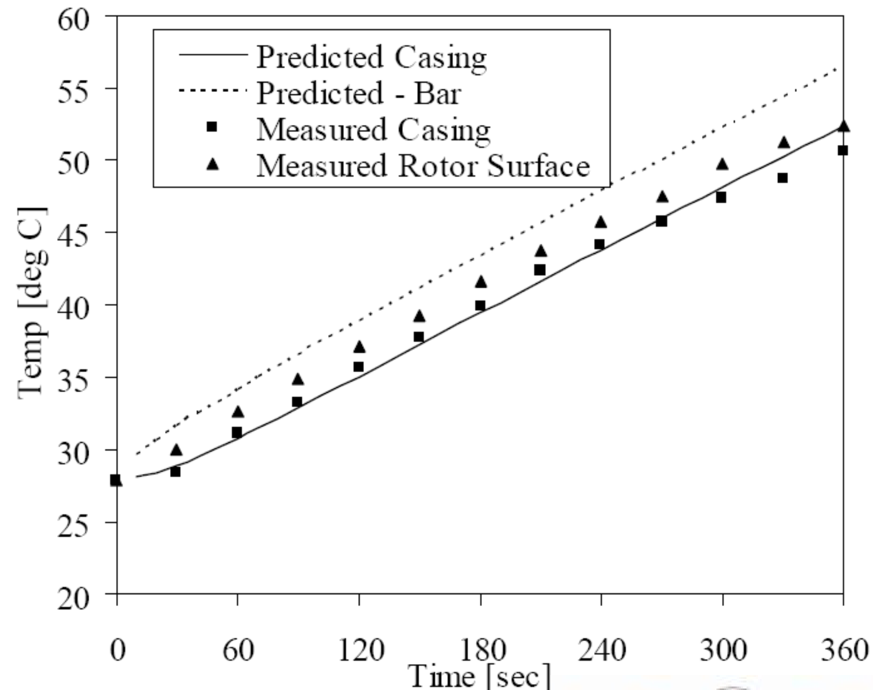

PEMD 2006

# Induction Motor Locked Rotor Analysis

- excellent agreement between test and calculation of induction machine transient on locked rotor
- Iterative calculation between SPEED and Motor-CAD
- Work done by Dave Dorrell at Glasgow University








IECON 2006

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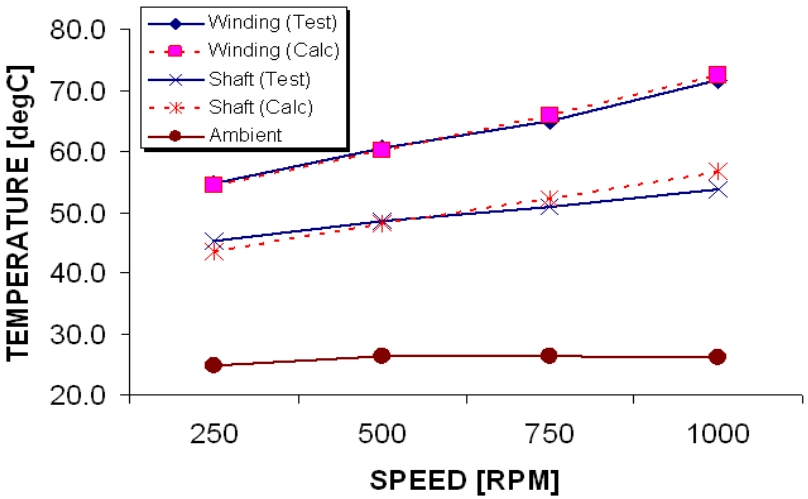
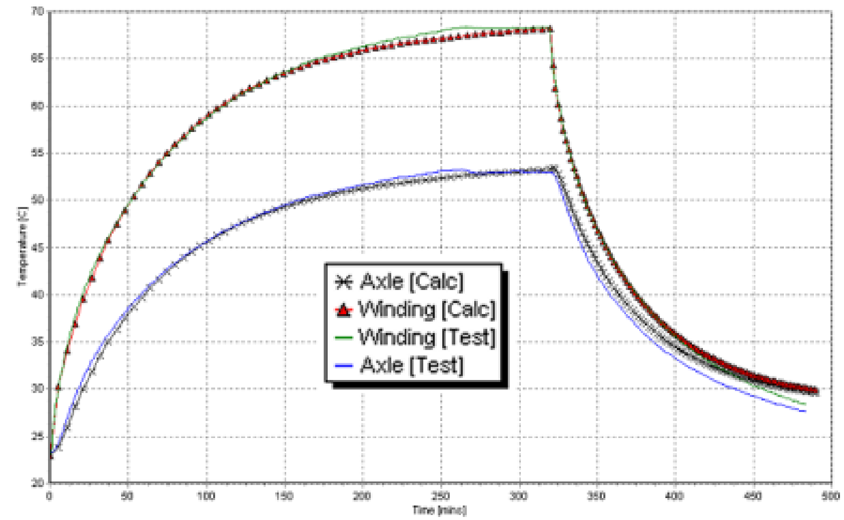
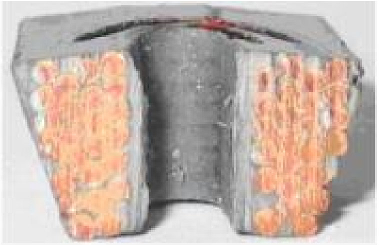
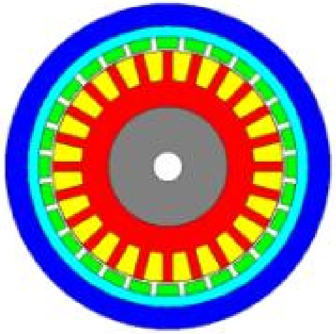
# Outer Rotor Brushless Motor

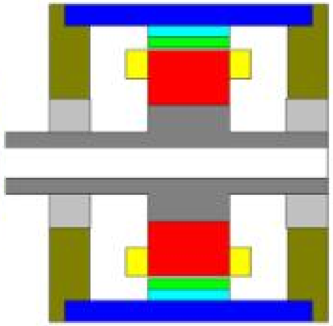



- Project with University of Bristol to optimise the cooling of this outer rotor BPM machine
- More complex cooling than traditional inner rotor machine







# TEFC & TENV Synchronous Gens.

- Range of machines tested and modelled with good level of agreement
  - Mostly <5% error
  - Cummins Generator Technologies and Edinburgh University

TENV Cooling:

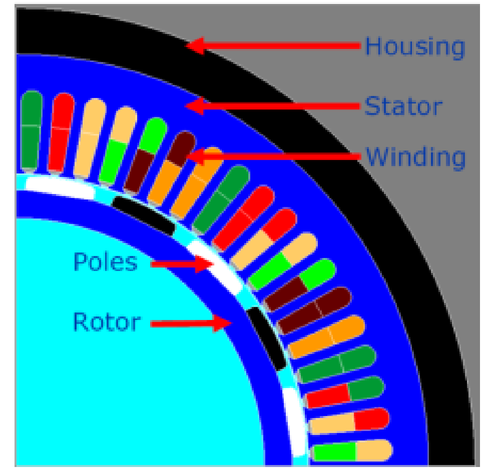
| 4.38 kVA | O/p Parameter | Exp. o/p (°C) | Motor-CAD o/p (°C) | ΔT (°C) [Exp. - Mot.] | % Error |
|---|---|---|---|---|---|
| | Housing - Front | 68.1 | 67.51 | 0.59 | 0.87 |
| | Housing - Active | 68 | 66.05 | 1.95 | 2.87 |
| | Housing - Rear | 64.8 | 63.94 | 0.86 | 1.33 |
| | Bearing - Front | 68.3 | 68.75 | -0.45 | -0.66 |
| | Bearing - Rear | 61.5 | 64.47 | -2.97 | -4.83 |
| | Stator | 74.1 | 74.86 | -0.76 | -1.03 |
| | Rotor | 83.7 | 82.05 | 1.65 | 1.97 |
| | Winding Average | 76.15 | 73.93 | 2.22 | 2.92 |
| | EndWinding (F) | 76.7 | 69.9 | 6.8 | 8.87 |
| | EndWinding (R) | 75.5 | 68.85 | 6.65 | 8.81 |
| | | | | Average % Error | 3.42 |

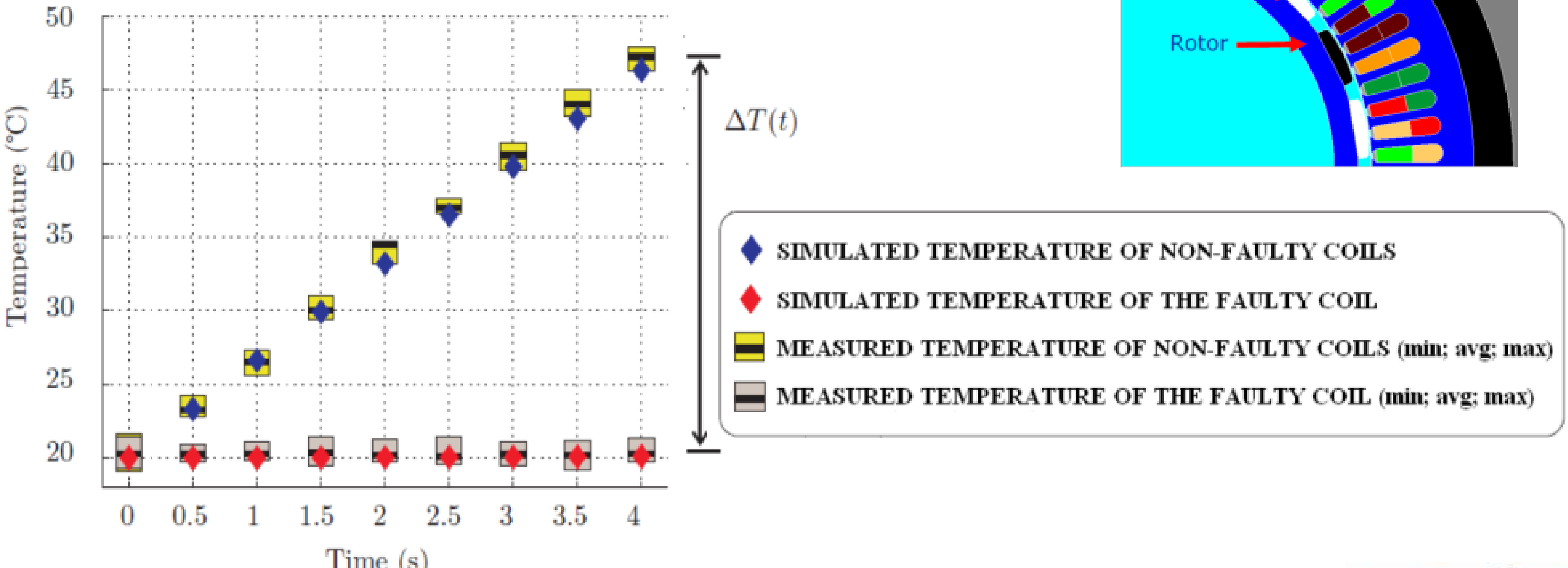
TEFC Cooling:

| 18 kVA | O/p Parameter | Exp. o/p (°C) | Motor-CAD o/p (°C) | ΔT (°C) [Exp. - Mot.] | % Error |
|---|---|---|---|---|---|
| | Housing - Front | 55.9 | 57.74 | -1.84 | -3.29 |
| | Housing - Active | 51.5 | 52.49 | -0.99 | -1.92 |
| | Housing - Rear | 48.1 | 48.14 | -0.04 | -0.08 |
| | Bearing - Front | 56.5 | 61.64 | -5.14 | -9.10 |
| | Bearing - Rear | 51.4 | 52.25 | -0.85 | -1.65 |
| | Stator | 69.2 | 74.68 | -5.48 | -7.92 |
| | Rotor | 98.4 | 101.2 | -2.8 | -2.85 |
| | Winding Av. | 81.8 | 82.45 | -0.65 | -0.79 |
| | EndWinding (F) | 90 | 94.07 | -4.07 | -4.52 |
| | EndWinding (R) | 88.3 | 92.21 | -3.91 | -4.43 |
| | | | | Average % Error | 3.66 |

# Wind Generator Thermal Analysis

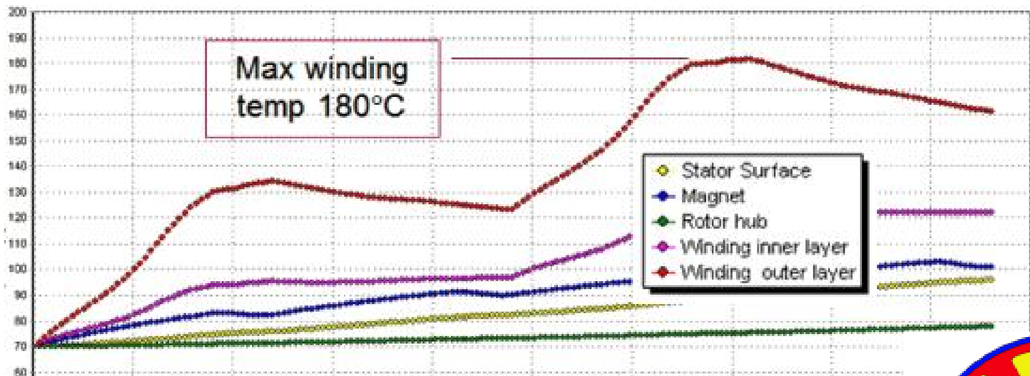
- Wind generator (University of Cantabria)
  - FLUX Users meeting 2010
  - Excellent results with Motor-CAD v Test
  - Fault Analysis also performed

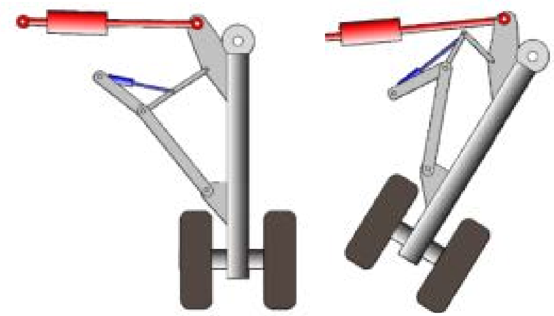


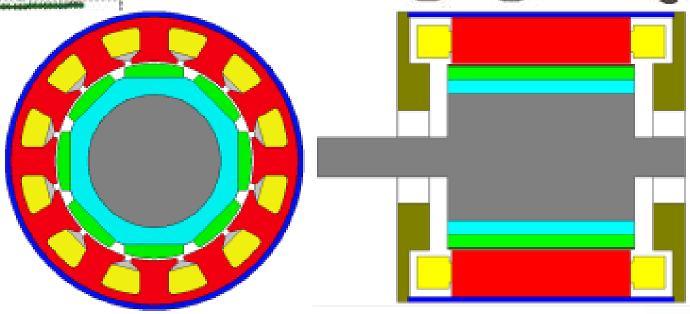



# Motor size and weight reduction

- Motor-CAD enabled a sizable reduction in size and weight compared to if no thermal analysis was done at the design stage
  - Weight reduction 31%
  - Volume reduction 39%





| Parameter | Unit | Before optimisation | After optimisation |
|---|---|---|---|
| Stator diameter | mm | 110 | 105 |
| Stator length | mm | 100 | 67 |
| Effective volume | $cm^3$ | 950 | 580 |
| Active mass | kg | 5.45 | 3.75 |

UK Magnetics Society Meeting 2008

Thank you for your attention!

- Are there any questions?