Lube Oil and Bearing Thermal Management System

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Outline

- Objectives
- Roller and Ball Bearings
- Heat Load Modeling
- Heat Load Calculation
- Simulink Modeling
- Petroff’s Law
- Results and Discussion
- Conclusion

Objectives

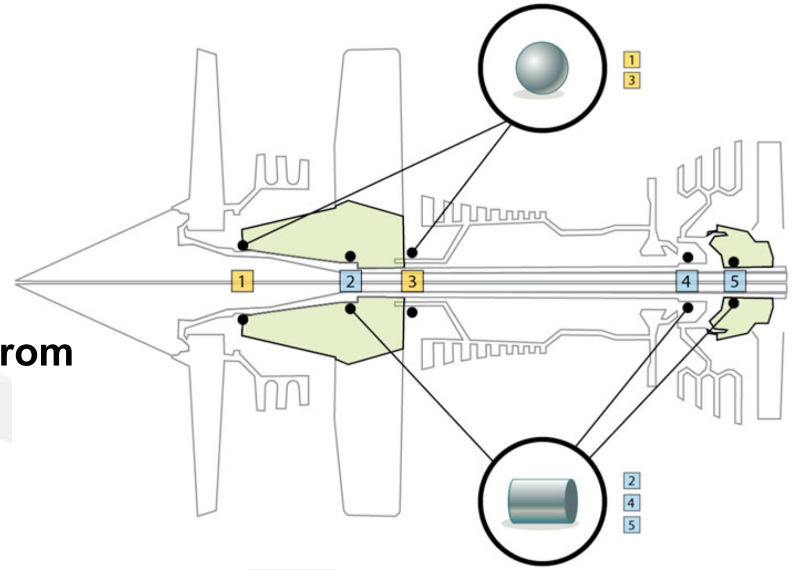
- Identify key components & thermal processes associated with the lubrication system of a gas turbine engine
- Capture the complex situation in a simple yet reliable model
- Identify & model the heat loads using the *MATLAB/Simulink* environment
- Verify the model by applying it to a realistic engine & mission

Types of Bearings & Locations

- No. 1 Ball Bearing
  - Thrust bearing between LP spool and engine main frame
- No. 2 Roller Bearing
  - Carries radial loads from the fan and booster
  - Centers the rotating LP assembly
- No. 3 Ball Bearing
  - Thrust bearing that centers HP spool in static assembly
  - Transfers axial loads
- No. 4 Roller Bearing
  - Located rear of HP shaft, separating it from LP spool
  - Centers both HP and LP spools
- No. 5 Roller Bearing
  - Supports LPT rotor inside turbine rear frame

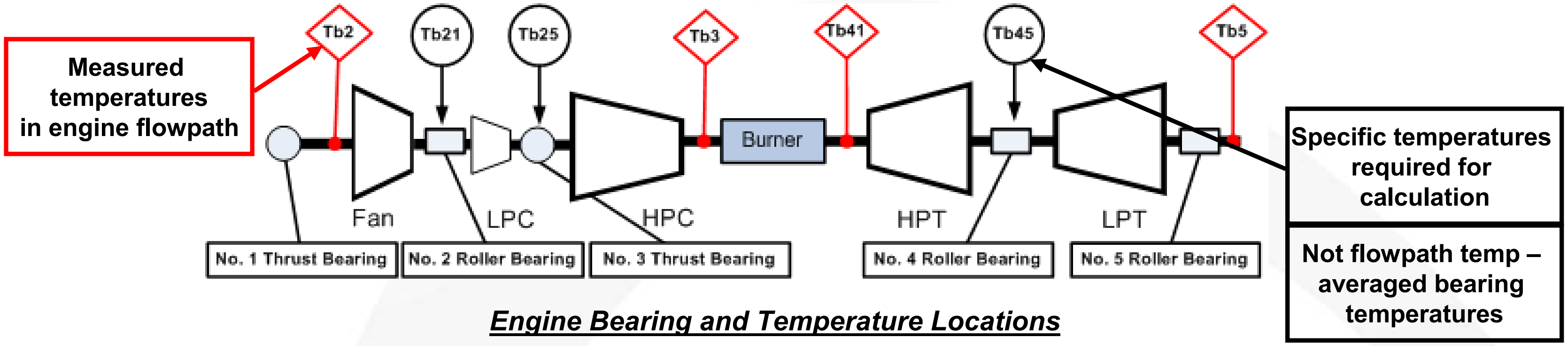
- **Ball Bearings**
- Centering mechanism for rotating assembly and radial loads
- Transmit axial loads from blade row forces

- **Roller Bearings**
- Located at the ends of the turbine and compressor shafts
- Mounted in housings with a thin layer of oil



Bearing Locations in a High Bypass Ratio Turbofan Engine

Heat Load Modeling



Measured temperatures in engine flowpath

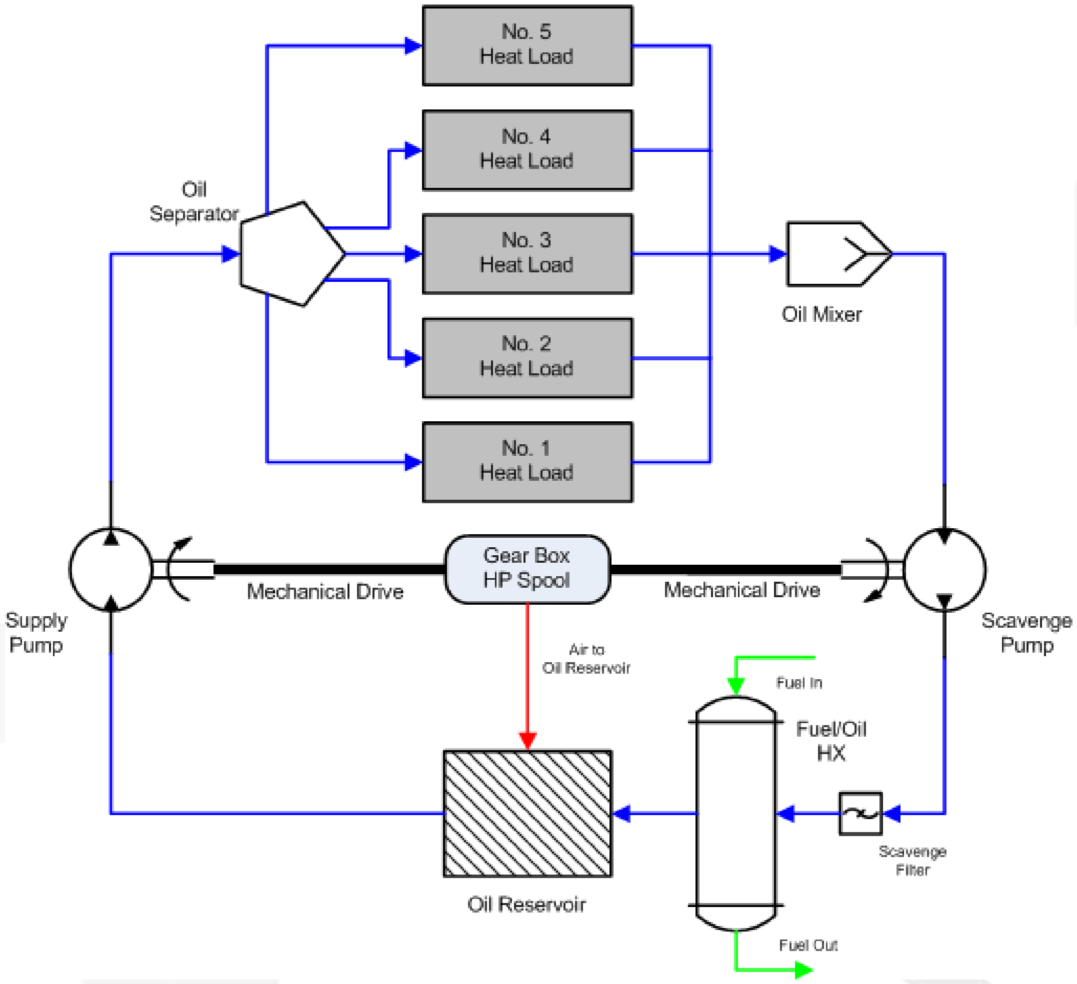
Specific temperatures required for calculation

Not flowpath temp – averaged bearing temperatures

Engine Bearing and Temperature Locations

- **Assumptions**
  - **Heat load for each bearing is a function of LP spool speed (N_1), HP spool speed (N_2), and engine station temperature (T)**
  - **Fuel properties into heat exchanger are known**
  - **Initial oil temperature increase, ΔT, of 150 to 250ºR (80 to 140ºK)**
  - **Oil temperature within supply tank typically kept between 700 and 800ºR (~200 and 350ºF, ~390 to 445ºK)**

Heat Load Modeling



Oil Loop Representative of the* MATLAB/Simulink *Model

Heat Load Modeling

- Single eqn. for solving heat load on each bearing element

$$\dot{Q}_{L,tot} = \dot{m}_{oil,tot} c_p \Delta T = \sum_{i=1}^{5} \dot{Q}_{Li}$$

1st Law Representation

$$\dot{Q}_{Li} = \alpha_{1i}(N_2 - N_1)^{\beta_{1i}} + \alpha_{2i} N_2^{\beta_{2i}} + \alpha_{3i} N_1^{\beta_{3i}} + \alpha_{4i} T^{\beta_{4i}}$$

Parametric Representation

  - ***cp*, *ΔT*, initial total flow rate are known for the 1st Law representation**
  - ***β* exponents are assumed unity, scaling coefficients, *α_i*, are the unknown in the parametric representation**

- 1st Law heat rate distribution

$$\dot{Q}_{L,tot} = \gamma_1 \dot{m}_{oil} c_p \Delta T + \gamma_3 \dot{m}_{oil} c_p \Delta T + \gamma_3 \dot{m}_{oil} c_p \Delta T + \gamma_4 \dot{m}_{oil} c_p \Delta T + \gamma_5 \dot{m}_{oil} c_p \Delta T$$

  - **Flow rate distribution based on location, environment, loading, size and quantity of bearings**

Initial Flow Rate Distribution

$$\begin{bmatrix} \gamma_1 \\ \gamma_2 \\ \gamma_3 \\ \gamma_4 \\ \gamma_5 \end{bmatrix} = \begin{bmatrix} 0.02 \\ 0.18 \\ 0.64 \\ 0.12 \\ 0.04 \end{bmatrix}$$

Bearing Heat Loads

$$\dot{Q}_{Li} = \alpha_{1i}\left(N_2 - N_1\right)^{\beta_{1i}} + \alpha_{2i} N_2^{\beta_{2i}} + \alpha_{3i} N_1^{\beta_{3i}} + \alpha_{4i} T^{\beta_{4i}}$$

Parametric Representation

- **No. 1 – front of fan, st. 2**
  - **LP spool and fan inlet temp**

$$\dot{Q}_{L1} = \gamma_1 \dot{m}_{oil} c_p \Delta T$$

$$\dot{Q}_{L1} = \alpha_{13} N_1 + \alpha_{14} T_{b2} \ \textit{where}\ T_{b2} = T_2$$

- **No. 2 – fan and booster (LPC), st. 21**
  - **LP spool and fan exit temp**

$$\dot{Q}_{L2} = \gamma_2 \dot{m}_{oil} c_p \Delta T$$

$$\dot{Q}_{L2} = \alpha_{21}\left(N_1\right) + \alpha_{22} T_{b21}$$

$$T_{b21} = \frac{T_2 + T_{b25}}{2}$$

- **No. 3 – booster and HPC, st. 25**
  - **HP spool, booster exit temp, HPC exit temp**

$$\dot{Q}_{L3} = \gamma_3 \dot{m}_{oil} c_p \Delta T$$

$$\dot{Q}_{L3} = \alpha_{32} N_2 + \alpha_{34} T_{b25}$$

$$T_{b25} = \frac{T_{b21} + T_{b3}}{2}$$

- **No. 4 – between HPT and LPT, st. 45**
  - **LP spool, HP spool and HPT exit temp**

$$\dot{Q}_{L4} = \gamma_4 \dot{m}_{oil} c_p \Delta T$$

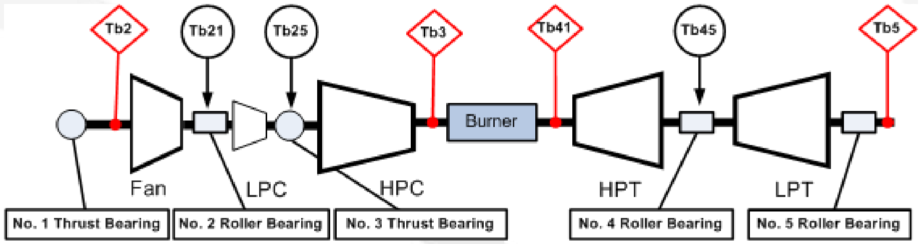
$$\dot{Q}_{L4} = \alpha_{41}\left(N_2 - N_1\right) + \alpha_{42} N_2 + \alpha_{43} N_1 + \alpha_{44} T_{b45}$$

$$T_{b45} = \frac{T_{b41} + T_{b5}}{2} \ \textit{and}\ T_{b41} = T_{41},\ T_{b5} = T_5$$

- **No. 5 – LPT and exhaust nozzle, st. 5**
  - **LP spool and LPT exit temp**

$$\dot{Q}_{L5} = \gamma_5 \dot{m}_{oil} c_p \Delta T$$

$$\dot{Q}_{L5} = \alpha_{53} N_1 + \alpha_{54} T_{b5}$$



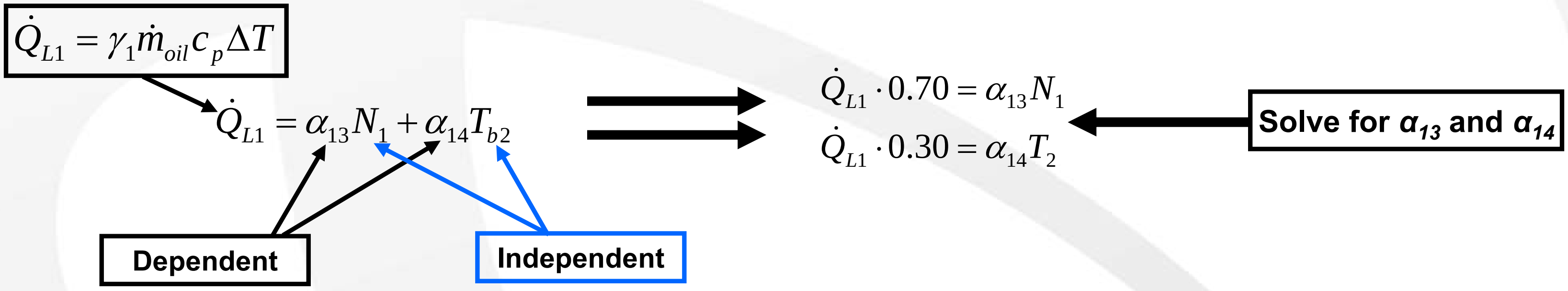
Engine Bearing and Temperature Locations

Heat Load Calculation

- Individual heat load on each bearing assembly
  - Oil flow rate provides initial guess for heat load generation – assumed 0.4 lbm/s (0.181 kg/s)
  - Initial total heat addition – 137,000 to 171,000 Btu/hr (40 to 50 kW)
    - From assumed Δ*T* of 150 to 250ºR (80 to 140ºK)

- For initial analysis
  - 70% – greatest heat load from friction – spool speed dependent
  - 30% - environment effects – engine temperatures

- Example: No. 1 Ball Bearing



$$\boxed{\dot{Q}_{L1} = \gamma_1 \dot{m}_{oil} c_p \Delta T}$$

$$\dot{Q}_{L1} = \alpha_{13} N_1 + \alpha_{14} T_{b2}$$

Dependent

Independent

$$\dot{Q}_{L1} \cdot 0.70 = \alpha_{13} N_1$$

$$\dot{Q}_{L1} \cdot 0.30 = \alpha_{14} T_2$$

Solve for α_{13} and α_{14}

Heat Load Calculation

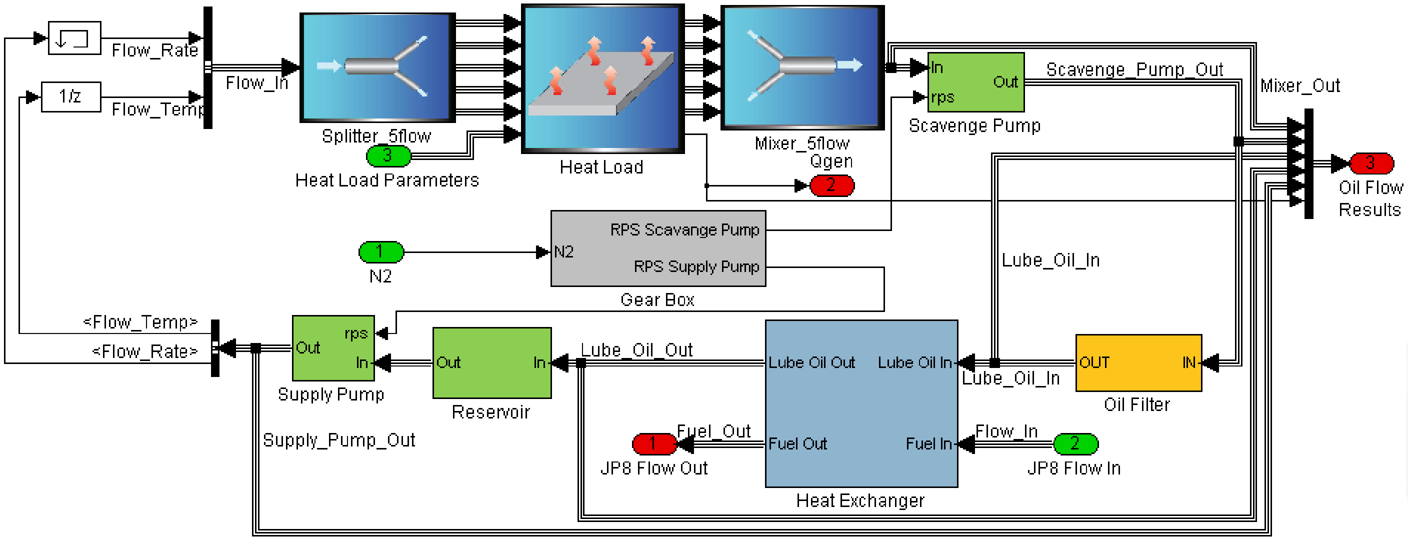
- **Applying this approach to all 5 bearing assemblies provides the scaling coefficient parameter array:**

$$\left.\begin{array}{l}\dot{Q}_{L1}\cdot 0.70=\alpha_{13}N_1\\ \dot{Q}_{L1}\cdot 0.30=\alpha_{14}T_2\\ \vdots\\ \vdots\\ \dot{Q}_{L4}\cdot 0.30=\alpha_{41}\left(N_2-N_1\right)\\ \dot{Q}_{L4}\cdot 0.20=\alpha_{42}N_2\\ \dot{Q}_{L4}\cdot 0.20=\alpha_{43}N_1\\ \dot{Q}_{L4}\cdot 0.30=\alpha_{44}T_{45}\\ \vdots\end{array}\right\}\boxed{\alpha_{13},\ \ldots,\ \alpha_{41},\ \ldots}\rightarrow\begin{bmatrix}\alpha_{11}&\alpha_{12}&\alpha_{13}&\alpha_{14}\\ \alpha_{21}&\alpha_{22}&\alpha_{23}&\alpha_{24}\\ \alpha_{31}&\alpha_{32}&\alpha_{33}&\alpha_{34}\\ \alpha_{41}&\alpha_{42}&\alpha_{43}&\alpha_{44}\\ \alpha_{51}&\alpha_{52}&\alpha_{53}&\alpha_{54}\end{bmatrix}=\begin{bmatrix}0&0&0.3021&1.827\\ 0&0&2.719&10.83\\ 0&7.673&0&28.71\\ 2.989&0.4111&0.5179&2.547\\ 0&0&0.6043&1.176\end{bmatrix}$$

- **169,100 Btu/hr [49.55 kW] heat load**

- **Sets the initial heat load parameters that determine the scalability of the *MATLAB/Simulink* model**

Simulink Modeling



__Simulink *Model of Thermal Management System*__

- **Heat load parameters supplied from look-up tables formulated using *GasTurb11***
- **Heat exchanger typically shell-and-tube or plate-fin configuration**
  - **Model assumes variation on a counter-flow, primary surface heat exchanger, typical run - 240ºR drop**
  - **Hot fluid – *Mil Spec MIL-PRF-7808L***
  - **Cold fluid – *JP-8***
- **Gearbox drives supply and scavenge pumps**
  - **Gear reduction – scavenge 10%, supply 30%**
  - **1.5 atm pressure rise, minimal temperature increase**
- **From previously defined heat load scaling parameters, the final temperature exiting each bearing assembly can be determined by:**

$$\dot{Q}_{L1} = \alpha_{13}N_1 + \alpha_{14}T_2 = \gamma_1 \dot{m}_{oil} c_p \left(T_f - T_{oil}\right)$$

Petroff's Law

- **Method of lubrication analysis – defines a uniformly distributed energy release in the lubricant due to shearing**

$$T_f = T_{oil} + \Delta T$$

- **Heat generation for a steady state system**

$$\dot{Q}_{gen} = TN = (\tau A r)N = \left(\frac{2\pi r\mu N}{c}\right)(2\pi r l)rN \rightarrow \dot{Q}_{gen} = TN = 4\pi^2 \frac{r^3 l}{c}\frac{\mu}{g_c}N^2$$

- **Environmental effects – convection and radiation**

$$\dot{Q}_{env} = U_0 A_0 \left(T_w - \tilde{T}_\infty\right) = \frac{U_0 A_0 \left(T_f - \tilde{T}_\infty\right)}{2} \quad where \ A_0 = 2\pi r l \ and \ U_0 = 3.0\frac{Btu}{h\cdot ft^2\cdot R}$$

- **Determine total heat addition, Δ*T***

$$\dot{Q}_{env} = \frac{U_0 A_0\left(T_{oil} + \Delta T - \tilde{T}_\infty\right)}{2} = 4\pi^2\frac{r^3 l}{c}\frac{\mu}{g_c}N^2 = \dot{Q}_{gen} \rightarrow \Delta T = \frac{8\pi^2}{U_0 A_0}\frac{r^3 l}{c}\frac{\mu}{g_c}N^2 + \tilde{T}_\infty - T_{oil}$$

- **Total heat generation from one bearing element**

$$\dot{Q}_{gen} = \dot{m}_{oil} c_p \Delta T \quad where \ \Delta T = \frac{8\pi^2}{U_0 A_0}\frac{r^3 l}{c}\frac{\mu}{g_c}N^2 + \tilde{T}_\infty - T_{oil} \quad and \ \dot{m}_{oil} = lcr\rho_{oil}\pi N$$

- **final oil temperature, *T*$_f$**

$$T_f = \tilde{T}_\infty + \frac{8\pi^2 \mu N^2 l r^3}{U_0 A_0 c g_c}$$

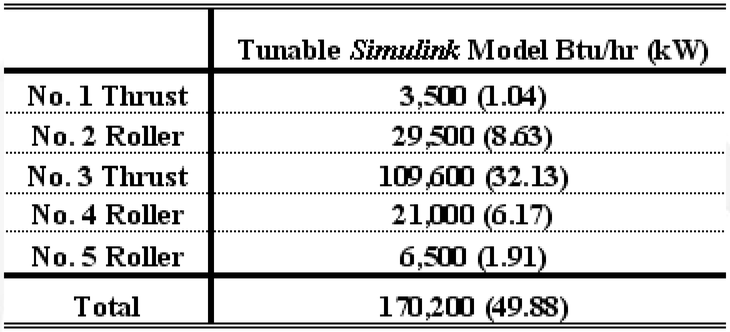
Results and Discussion

- *Petroff's Law* predicted heat loads higher than initial calculations
  - Initial – Δ*T* = 242ºR (134ºK)
  - *Petroff* – Δ*T* = 550ºR (306ºK)

- 70% from friction and 30% from surroundings, does not hold true for all cases

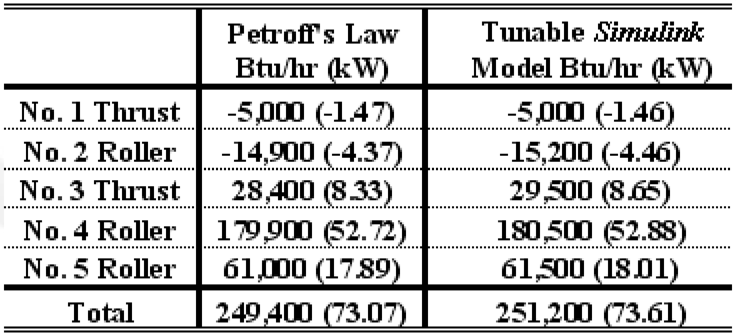
- Modifications to reflect local environment
  - Current study suggested a reversal in heat addition contributions for local temperatures > 1800ºR (1000ºK)

Heat Generation from Initial Analysis of the Tunable* Simulink *Model



	Tunable *Simulink* Model Btu/hr (kW)
No. 1 Thrust	3,500 (1.04)
No. 2 Roller	29,500 (8.63)
No. 3 Thrust	109,600 (32.13)
No. 4 Roller	21,000 (6.17)
No. 5 Roller	6,500 (1.91)
Total	170,200 (49.88)

Heat Generation from Petroff's Law Prediction and the Tunable* Simulink *Model



	Petroff's Law Btu/hr (kW)	Tunable *Simulink* Model Btu/hr (kW)
No. 1 Thrust	-5,000 (-1.47)	-5,000 (-1.46)
No. 2 Roller	-14,900 (-4.37)	-15,200 (-4.46)
No. 3 Thrust	28,400 (8.33)	29,500 (8.65)
No. 4 Roller	179,900 (52.72)	180,500 (52.88)
No. 5 Roller	61,000 (17.89)	61,500 (18.01)
Total	249,400 (73.07)	251,200 (73.61)

Individual components of heat generation are at steady state, sea-level-static conditions

Results and Discussion – 2^{nd} Look

- Heat addition predicted by *Petroff's Law* too high
    - Initial – ΔT = 242ºR (134ºK)
    - *Petroff* – ΔT = 550ºR (306ºK)

- Adjusted Environmental Parameters
    - Bearing size and location scaled from accurate representation of a two-spool, mid sized turbofan engine

    - Adjusted the heat transfer coefficient to better portray numbers found in literature
        - [U_o = 5.5 BTU/hr-ft^2-R (31 W/m^2-K)]

    - Incorporated cooling flow to location Tb_{45} and Tb_5

    - *Petroff* – ΔT = 300ºR (167ºK)

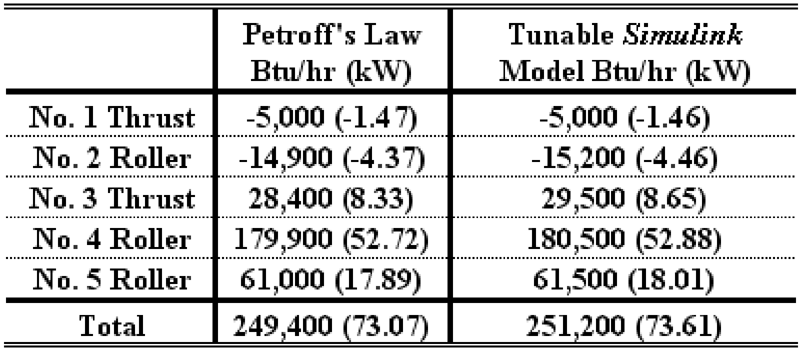
Results and Discussion

- Initial Analysis
  - 169,100 Btu/hr (49.55 kW)
  - 0.40 lbm/s (0.181 kg/s)
  - Δ*T* = 242ºR (134ºK)

- Original *Petroff's Law* Analysis
  - 251,200 Btu/hr (73.61 kW)
  - 0.40 lbm/s (0.181 kg/s)
  - Δ*T* = 550ºR (306ºK)

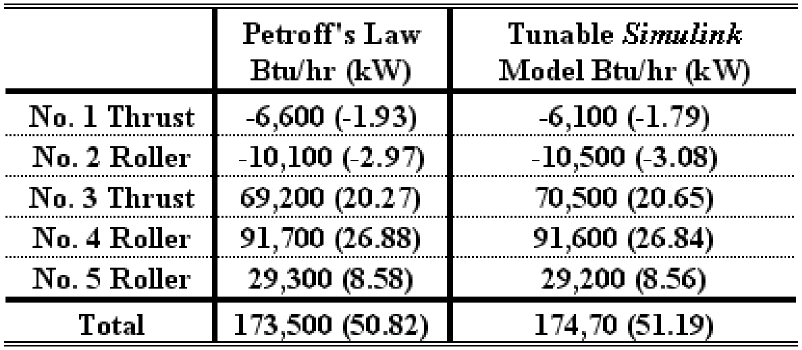
- 2nd Look *Petroff's Law* Analysis
  - 176,900 Btu/hr (51.82 kW)
  - 0.41 lbm/s (0.186 kg/s)
  - Δ*T* = 300ºR (167ºK)

Original *Petroff's Law* Heat Generation



	Petroff's Law Btu/hr (kW)	Tunable *Simulink* Model Btu/hr (kW)
No. 1 Thrust	-5,000 (-1.47)	-5,000 (-1.46)
No. 2 Roller	-14,900 (-4.37)	-15,200 (-4.46)
No. 3 Thrust	28,400 (8.33)	29,500 (8.65)
No. 4 Roller	179,900 (52.72)	180,500 (52.88)
No. 5 Roller	61,000 (17.89)	61,500 (18.01)
Total	249,400 (73.07)	251,200 (73.61)

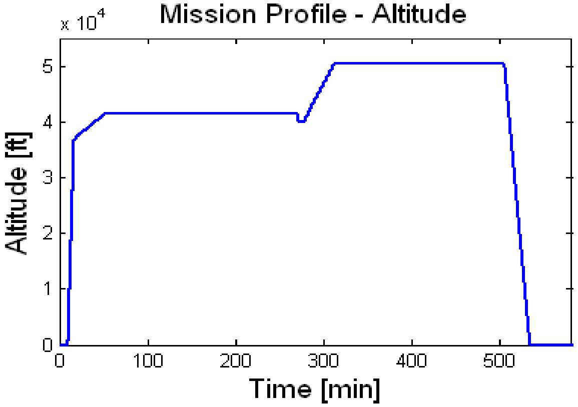
Adjusted *Petroff's Law* Heat Generation

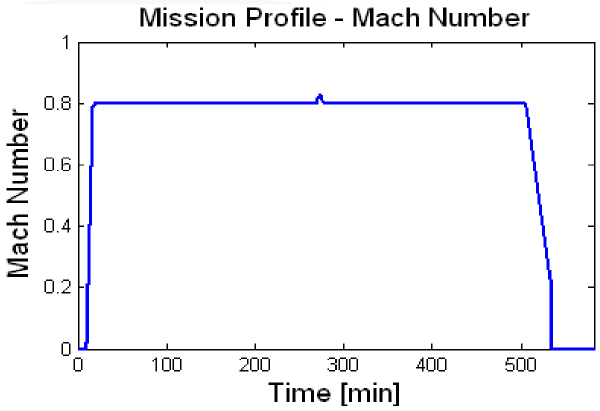


	Petroff's Law Btu/hr (kW)	Tunable *Simulink* Model Btu/hr (kW)
No. 1 Thrust	-6,600 (-1.93)	-6,100 (-1.79)
No. 2 Roller	-10,100 (-2.97)	-10,500 (-3.08)
No. 3 Thrust	69,200 (20.27)	70,500 (20.65)
No. 4 Roller	91,700 (26.88)	91,600 (26.84)
No. 5 Roller	29,300 (8.58)	29,200 (8.56)
Total	173,500 (50.82)	174,70 (51.19)

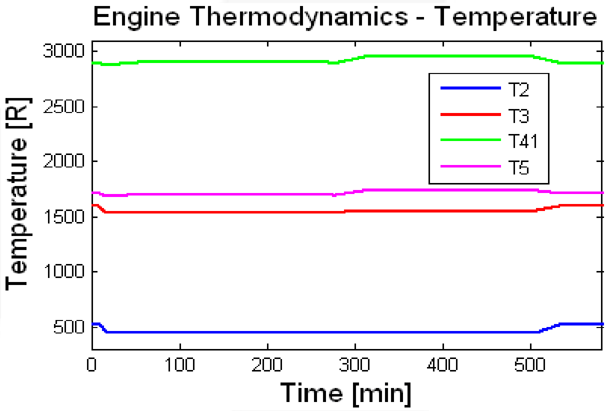
Mission Profile and Engine Parametrics

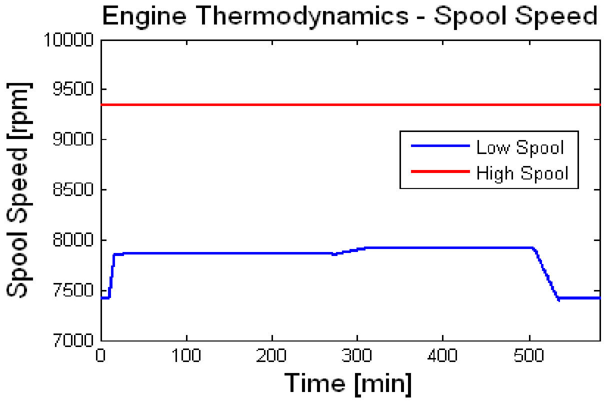
- **Mission Profile**





- **Engine Parametrics – *GasTurb11* engine simulation model**

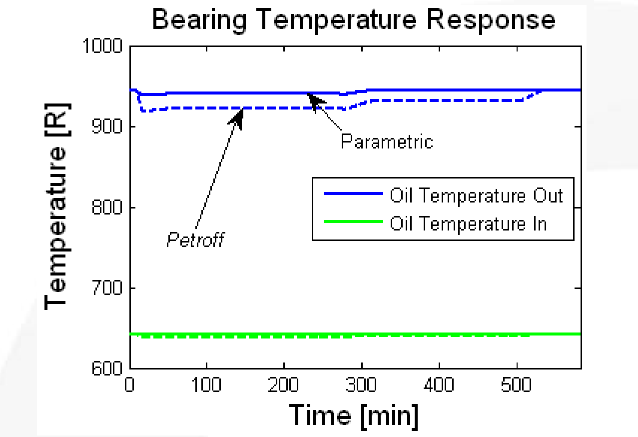




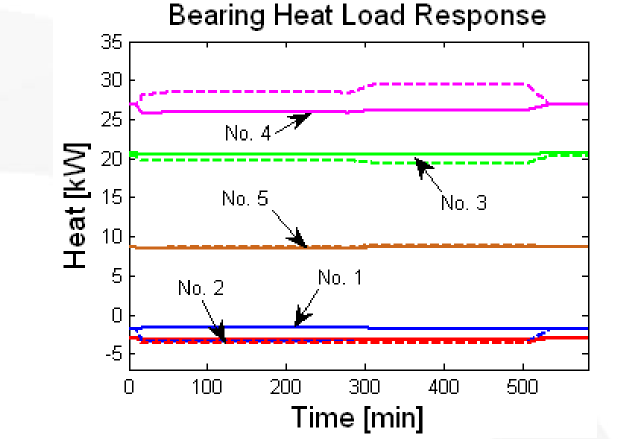
- **Fuel properties into the heat exchanger provided by an industry project team**

2nd Look Results and Discussion

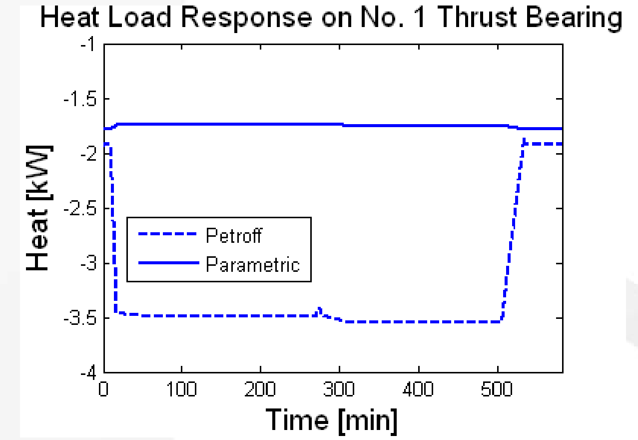
- **Changes match mission specifics**
- **Temperature gradients between oil splitter and mixer and across oil-to-fuel HX**
- **Negligible pump temperatures**
- **Δ*T* in lubrication cycle of 330ºR (~167ºK)**
- **~642 to 942ºR which is ~357 to 523ºK**



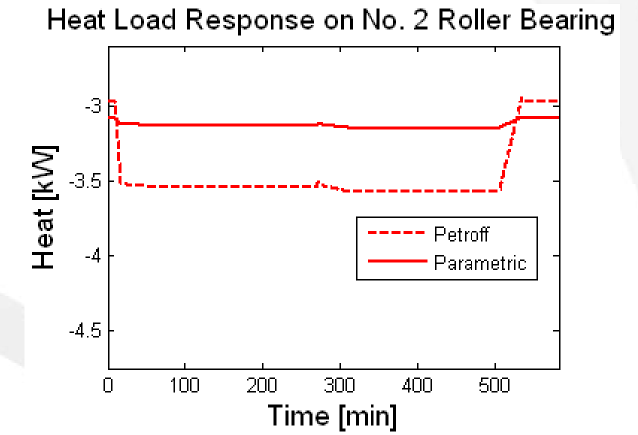
***Simulink* Oil Temperature Response**



***Simulink* Heat Load Response**



***Simulink* Heat Load Response on the No. 1 Thrust Bearing**



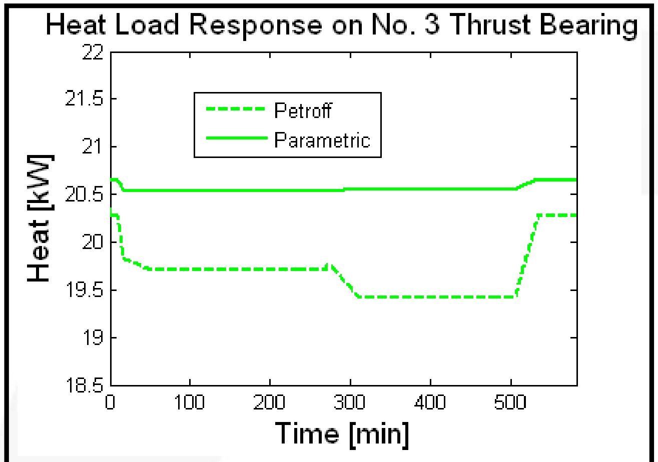
***Simulink* Heat Load Response on the No. 2 Roller Bearing**

2nd Look Results and Discussion

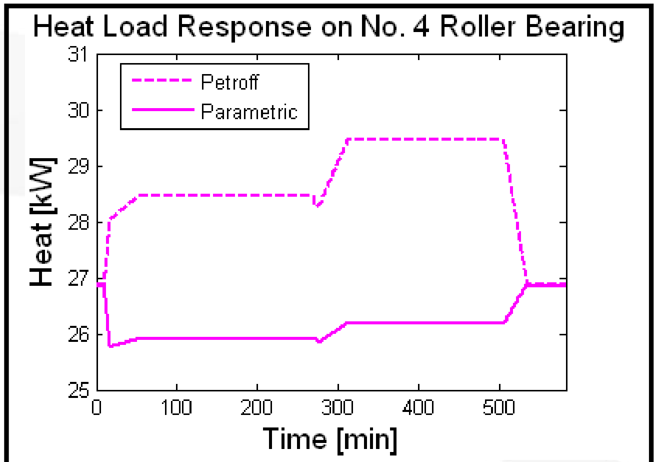
- **Heat load response gives similar behavior – more severe gradient seen with *Petroff's Law* model**

- **Parametric**
  - **Single Point Optimization**

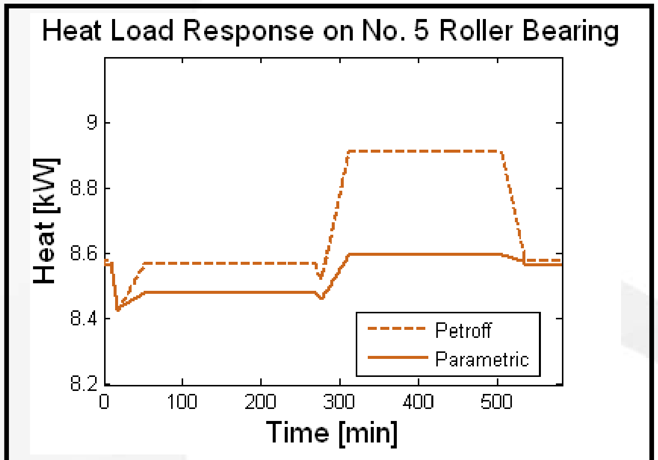
- ***Petroff's Law***
  - **Off-Design Mission**



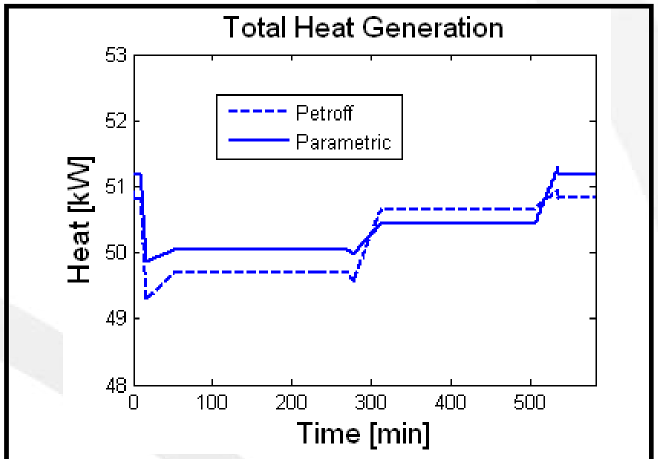
***Simulink* Heat Load Response on the No. 3 Thrust Bearing**



***Simulink* Heat Load Response on the No. 4 Roller Bearing**



***Simulink* Heat Load Response on the No. 5 Roller Bearing**



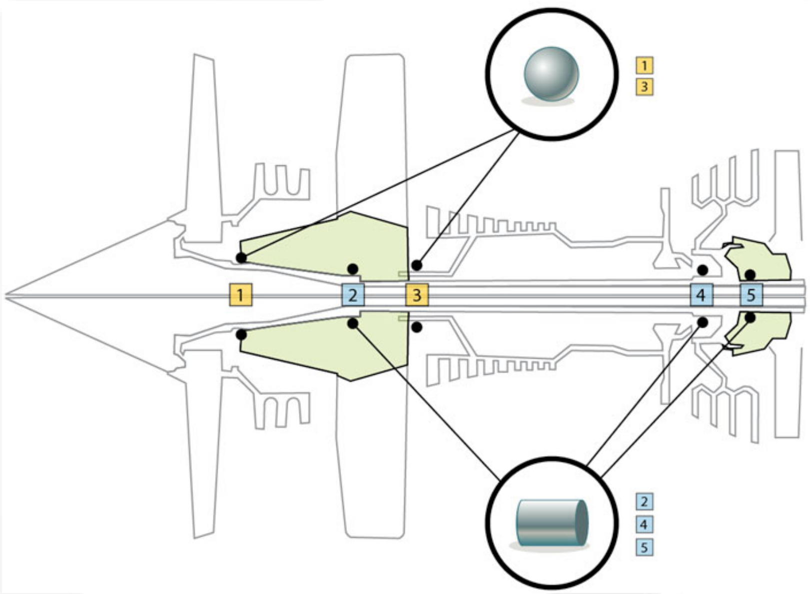
***Simulink* Total Heat Generation Response**

Conclusions

- A general approach has been developed to simulate a lubrication system that includes the heat loads that are part of more sophisticated engine TMS.
- The simple representation is an intermediate, yet realistic, step towards a more detailed thermal management system model.
- General assumptions have been made to greatly simplify, yet adequately capture, the definition of the engine lubrication circuit.
- A simulation capability based on the *MATLAB/Simulink* environment provides a general and tunable approach to examining the behavior of thermal management systems in real gas turbine engines.
- Results from this elementary model provide system level designers with a capability to conduct preliminary design and trade studies in new advanced systems, where relatively little information is available.

Acknowledgements

Avetec would like to acknowledge the assistance of Boeing and P.C. Krauss and Associates (PCKA), West Lafayette, IN for their joint support of this research



ASME Turbo Expo 2009: Power for Land, Sea and Air: Paper GT2009-60048
June 8-12, 2009, Orlando, FL USA