Methods to Measure, Predict and Relate Engine Friction, Wear and Fuel Economy

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Overview

Timeline

- Start date: 15 December 2014
- End date: 14 December 2017
- Completion: 20%

Budget

- Total funding (80/20): $1.32M
  - Cost share: $280K (> 20%)
  - DOE share: $1,040K
  - $390K to ANL over 3 yrs
- BY1: 12/2014 – 12/2015
  - $162K to Ricardo
- BY2: 12/2015 – 12/2016
  - $427K to Ricardo

Barriers

- Barriers to Friction Reduction Technology Adoption
  - **Risk aversion** → New technologies are not very well understood in regards to their durability and long-term benefits
  - **Cost** → The time and financial investment to screen technologies is prohibitive
  - **Computational models, design and simulation methodologies** → Analytical methods lack sufficient validation

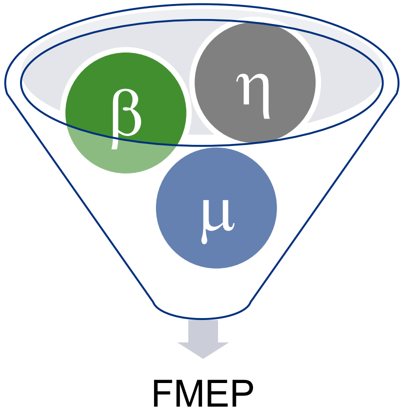
Partners

- Ricardo, Inc. (Lead)
- Argonne National Lab
- Isuzu
- ZYNP
- Infineum

Relevance

- To overcome the barriers to adoption of advanced vehicle technologies that improve fuel economy, in particular friction reduction technologies, this research effort has been designed with the following objective:
  - To develop methods capable of predicting the impact of friction reduction technologies on engine fuel economy and wear. The methods of prediction will be both empirically and analytically based.
- Empirical correlations will be established that allow for estimating changes in engine FMEP or fuel consumption based only on the engine speed, power and tribological parameters such as oil viscosity (η, β) and coefficient of friction (μ) which can be determined *a priori* in a lab-scale test.
- In a similar way, the same tribological parameters will be used as input into advanced simulation methods to predict changes in FMEP.

Key Idea: If one knows how a particular friction reduction technology changes η, β and μ then the methods developed during this project can be used to predict the impact on fuel consumption and wear. It is more cost effective to measure η, β and μ in a lab-scale test than conduct motored or fired engine tests.



Funnel = empirical correlation or advanced simulation methods

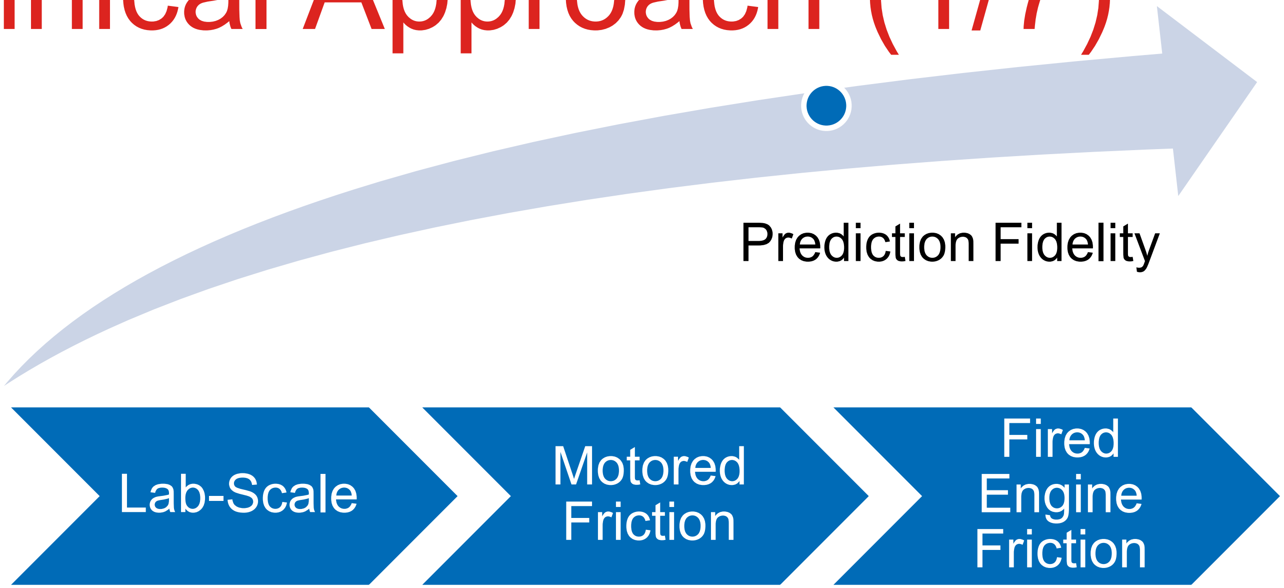
Milestones (2015)

Description	Type	Date	Status
Delivery of base components from Isuzu	Milestone	6 May 15	Complete
Preliminary RINGPAK Model of AART	Milestone	27 May 15	Complete
Final determination of FM	**GO/NO GO**	**14 Oct 15**	**Complete**
Engine first fire and de-green	Milestone	30 Oct 15	Complete
Delivery of final oils from Infineum for testing	Milestone	16 Nov 15	Complete

Milestones (2016)

Description	Type	Date	Status
Completion of testing at ANL with base Isuzu components	Milestone	15 Feb 16	Complete
Completion of testing at EMA with Isuzu base components	Milestone	31 March 16	Complete
Completion of engine thermal survey	Milestone	3 May 2016	On Track
Delivery of coated Isuzu components	Milestone	10 June 16	On Track
Completion of motored engine friction tests	Milestone	22 July 16	On Track
Completion of fired engine friction tests	Milestone	29 Aug 16	On Track
Final RINGPAK model of AART including validation	Milestone	16 Sep 16	On Track
Completion of accelerated wear testing	Milestone	4 Oct 16	On Track
Determination of hardware for long-term wear testing	**GO/NO GO**	**15 Oct 16**	**On Track**
Completion of RINGPAK/PISDYN model of motored & fired engine including validation	Milestone	22 Nov 16	On Track

Technical Approach (1/7)



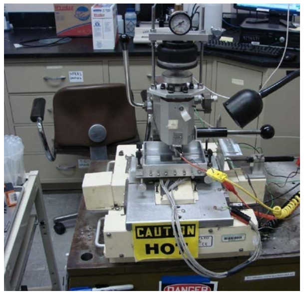
- Friction reduction technologies will be chosen for evaluation not necessarily because they are commercial viable but because of their usefulness in developing and validating prediction methodologies.
- These technologies will be tested in a progression of controlled test methods each with its own pros and cons for quantifying friction and wear.
- Data obtained from these experiments will be used to develop and validate empirical correlations and CAE methods.

Key Deliverables:
- Empirical correlations which relate tribological parameters to engine friction and wear
- CAE best practices for predicting friction and wear

Technical Approach (2/7)

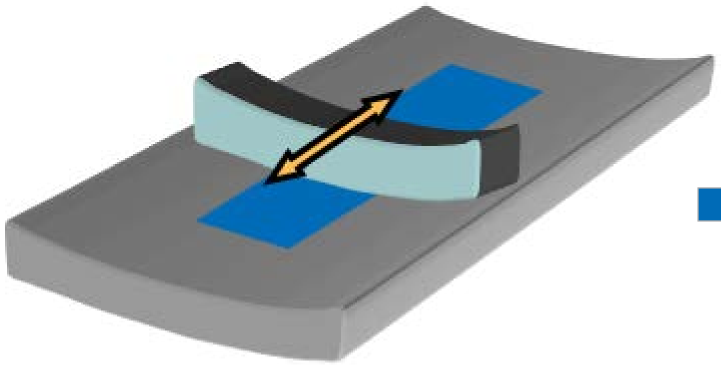
- Isuzu 4H engine will be the testbed.
- Advanced lab-scale test capabilities from Argonne National Lab will be leveraged, e.g., reciprocating tribometer (AART), white light interferometry.
- Dyno facilities at Ricardo will be used for motored and fired engine friction tests
- Friction reduction technologies will consist of advanced lubricants and surface treatments of the piston ring and skirt, i.e., power cylinder components only.
- Technologies will tested at two levels and in various combinations to generate the data set for regression modeling and CAE validation.

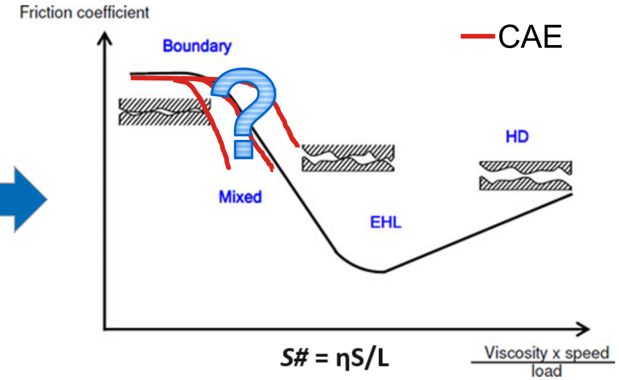
Build	Ring	Piston	Oil
1	Base	Base	A
2	New	Base	A
3	Base	New	A
4	New	New	A
5	Build 1		B
6	Build 2, 3, or 4		B



Technical Approach (3/7)

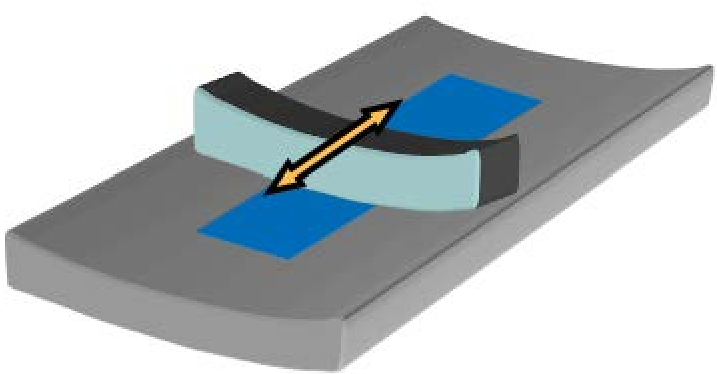
- Lab-scale measurements can’t measure FMEP or fuel economy impacts of friction reduction technology directly.
- However, lab-scale measurements can be precisely controlled so that very few noise factors are introduced. Thus, they are the first step in developing CAE methods to predict friction.
  - RINGPAK model of ANL's reciprocating tribometer (AART) will be developed and used to validate model setup, e.g., surface characteristics and other tunable parameters
  - Lab-scale measurements will be over a range of S# that includes both boundary lubrication and mixed lubrication. The data will be used to validate the ability of CAE methods to capture the transition between the regimes.





Technical Approach (4/7)

- Motored dyno tests allow for direct measurement for FMEP.
- However, motored dyno tests don’t duplicate a fired engine conditions due the absence of ring loading from cylinder pressure and thermal effects.
- They are an intermediate step to progress the modeling effort without adding too many new noise factors.
  - RINGPAK and PISDYN models of the motored engine configuration will be developed based on lab-scale models and validated against motored engine dyno data → **fundamental model parameters shouldn’t change to achieve correlation.**
- Motored friction tests will <u>also</u> be done at very low engine speeds so as to match the S# used in the lab-scale tests. The purpose of the very low engine speed tests is to identify limitations in the lab-scale tests (if any):



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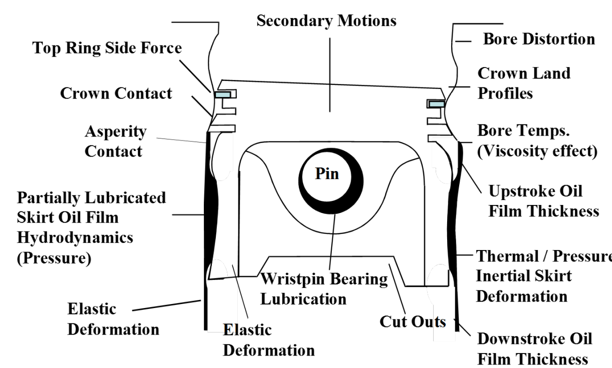
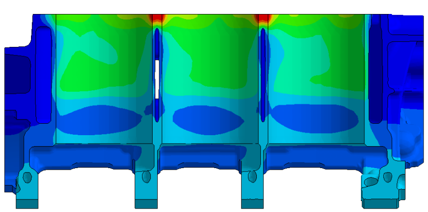
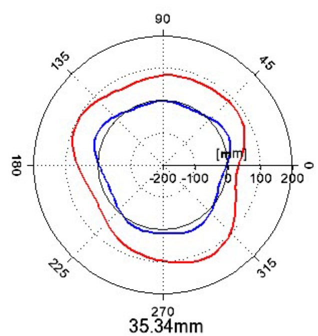
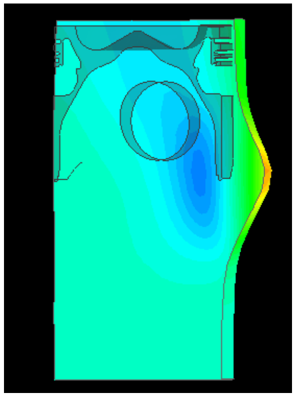


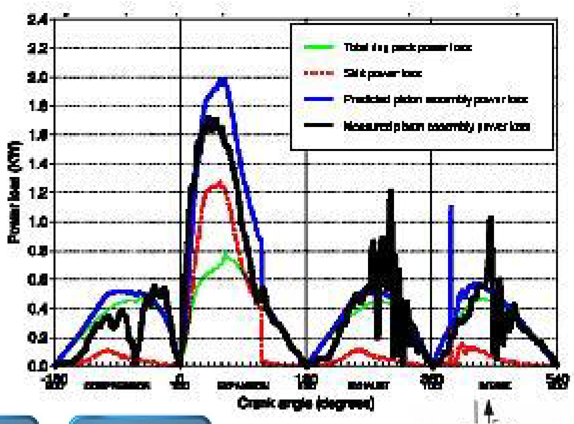
Technical Approach (5/7)

- Fired engine dyno tests best model real world operation by introducing the cylinder pressure and thermal effects missing in motored friction tests.
  - Fuel consumption will be measured.
  - FMEP will be inferred from cylinder pressure measurements.
- Motored engine RINGPAK and PISDYN models will be updated with fired engine boundary conditions and validated against fired engine dyno data.
  - Thermal-FEA will be used to provide bore distortion and local surface temperatures of components.
- Key deliverable will be analytical best practice for predicting friction.



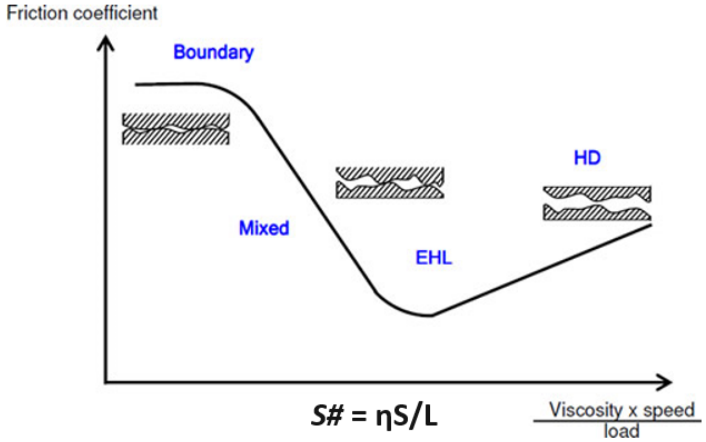


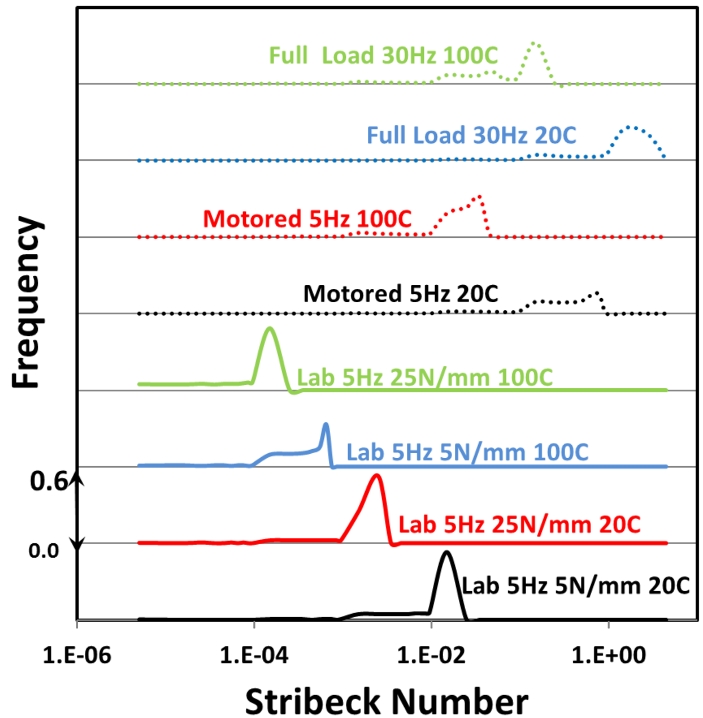




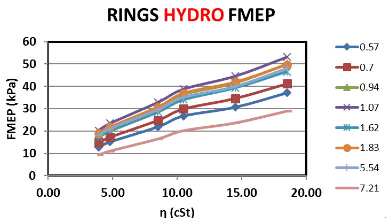
Technical Approach (6/7)

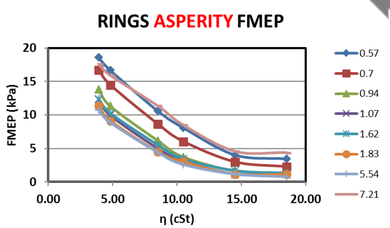
- Stribeck Number – S# = ηS/L, where
  - η = dynamic viscosity (Pa-s)
  - S = speed (m/s)
  - L = Load (N) (N)
  - = Load (N per unit length) (N/m)
  - = Load (avg contact pressure) (Pa)
- Engines and lab-scale measurements don't operate in the same range.
- Using an analytical model of a *representative* engine (developed under a CRADA), S# probability plots will be generated as a function of speed and load.
- This will be used to identify the friction regime in which the engine is operating as a function of speed and load. It will guide the selection of engine test conditions bridging the gap between lab-scale and engine tests.

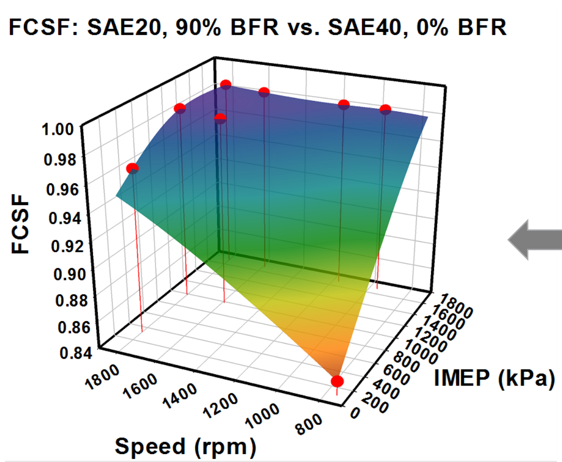




Technical Approach (7/7)







- Linear regression models of predicted FMEP from RINGPAK/PISDYN simulations will be made (model of a model to expand rage of applicability).
  - FMEP predictions will be validated against motored and fired engine dyno testing*.
- Possible model forms used to calculate ΔFMEP:

$FMEP_{hydro} = y_{h0} + a_h\eta$
$FMEP_{asp} = y_{a0} + a_a e^{(-b\eta)}$
$FMEP = FMEP_{hydro} + FMEP_{asp}$
$(y_{a,h0}, a_a, a_h, b) = A_0 + A_1 N + A_2 N^2 + A_3 IMEP + A_4 IMEP^2$
η = viscosity
N = Speed (rpm)
MEP = Mean Effective Pressure (kPa); F → Friction I → Indicated

- Through fuel consumption scaling factors (FCSF), fuel economy improvements relative to a baseline fuel economy map from friction reduction technologies will be calculated and validated against actual fuel economy improvements from fired engine dyno tests*.

$$FCSF = \frac{(IMEP + \Delta FMEP)}{IMEP}$$

- Interrogate FCSF over drive cycle to obtain real world fuel economy improvements.

*Corrected to account for lubricant effects on non power cylinder components contributing to FMEP or fuel consumption

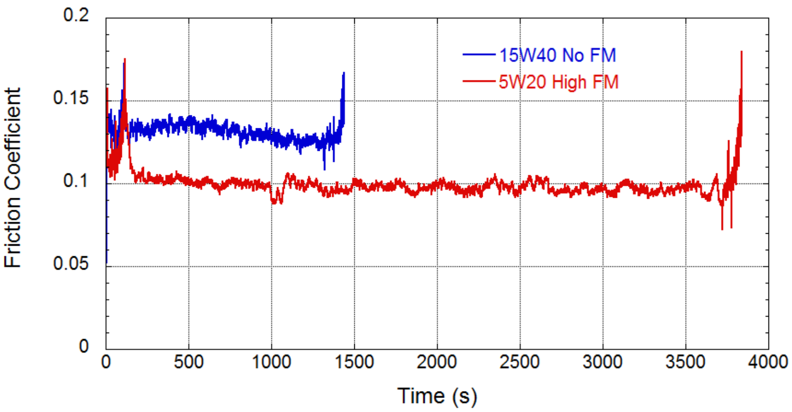
Oil Candidate Down-Select

- Objective: to identify a low friction oil candidate for lab-scale and dyno testing which maximizes friction reduction
- Over 22 variants of a low viscosity and/or high FM oil were tested.
- From all of the oils tested, we have identified a 15W40 (no FM) and a 5W20 (High FM) to be used for all the ring-on-liner and skirt-on-liner tests.

Oil	Description	Tests performed	Comments
Batch 1 oils	2 high vis HDDO oils (15W/40) - one with no Fm, one with a high FM treat	Ring-on-Liner and Ball-on-Flat	Preliminary blends exhibited minimal impact of FM on friction response - decision to reblend oils using PCMO add package
	2 low-vis HDDO oils (5W/20) - one with, one without FM	Ring-on-Liner and Ball-on-Flat	
Batch 2 oils	4 PCMO 5W/20 oils (1-4) with different FMs	Ball-on-Flat	Preliminary results exhibit measurable differentiation between 15W/40 HDDO and 5W/20 PCMO oil 3 - decision to optimize treat rate of oil 3
	4 HDDO 5W/20 oils (5-8) with different FMs	Ball-on-Flat	
Batch 3 oils	2 PCMO variants of PCMO oil 3 (3a and 3b from Batch 2) with different FM chemistry and treat rate	Ball-on-Flat	Optimization of oil 3 from batch 2 (3a and 3b from Batch 2) exhibited marginal impact of treat rate of oil 3 (3, 3a, 3b)
Batch 4 oils	6 PCMO 5W/20 oils (1, 2, 3, 3a, 3b, 4) - confirmtaion blends	Ball-on-Flat	Re-blend of batch 2 oils 1, 2, 3, 3a, 3b, 4, 5, 6, 7, 8 (from Batch 2 and 3) and test. Significant differentiation of friction response between HDDO high-vis (15W/40) No FM oil (baseline) and HDDO (5W/20) High FM2 oil 7
	4 HDDO 5W/20 oils (5-8) - confirmation blends	Ball-on-Flat	

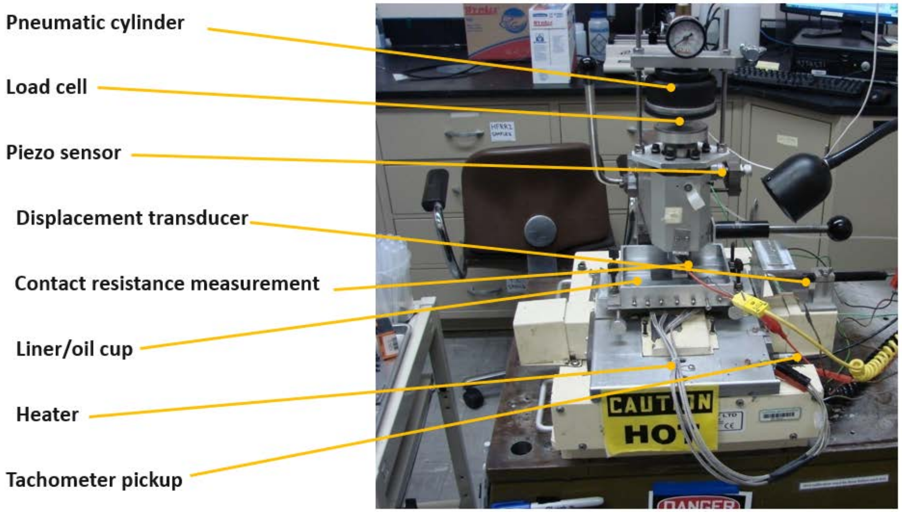
Oil Candidate Down-Select

Blend	15W40 no FM	5W20 High FM
Code main name	IM1502322-A-001	E00365-708-7
Description	No FM	High FM
Estimated Viscosity Grade	15W-40	5W-20
Formulation	HDD	HDD
kv100 (cSt)	14.30	7.46
CCS temperature deg C	-20	-30
CCS (cP)	3890	5018



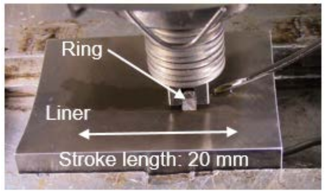
- Project was targeting ≥ 25% reduction
- Asperity friction coefficient, μ_{asp}, reduced approximately 25%
- Oil viscosity reduced approximately 48% at 100 °C.
- This fulfills the first GO/NO GO requirement

Lab-Scale Testing Summary



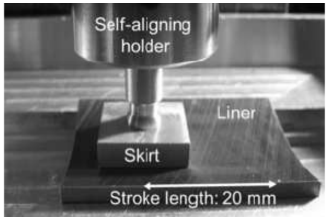
- Isuzu 4H base ring-on-liner and skirt-on-liner tests completed using reciprocating tribometer at ANL.
- Tests used to quantify μ_{asp} and provide raw data for model validation.
- FM has a similar impact on μ_{asp} with actual engine components as it did with ball-on-flat.

Ring/liner



Temperature (°C)	70		100		130	
Load (N)	**50**	**250**	**50**	**250**	**50**	**250**
15W40 no FM	0.130	0.126	0.133	0.136	0.138	0.132
5W20 high FM	0.088	0.092	0.102	0.097	0.097	0.098

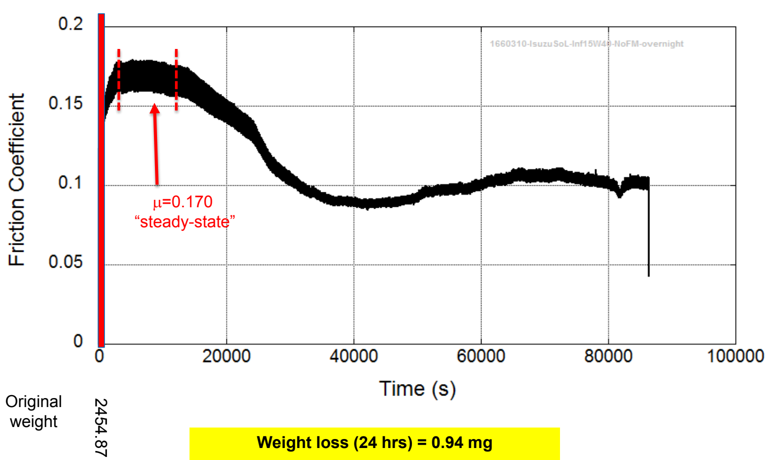
Skirt/liner

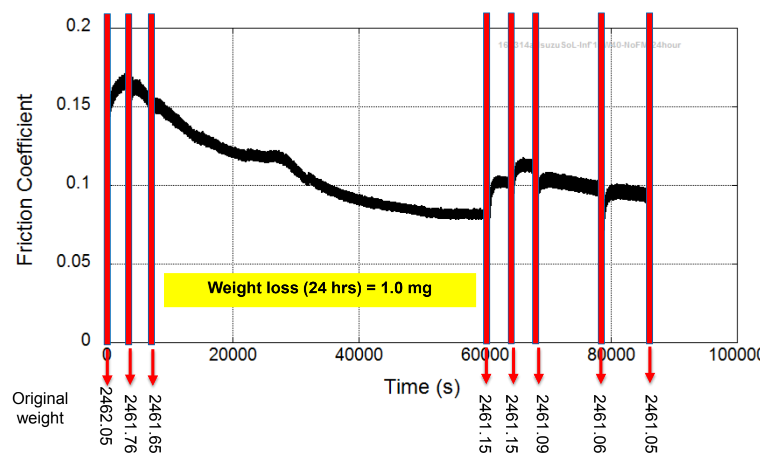


Temperature (°C)	70		100		130	
Load (N)	**50**	**250**	**50**	**250**	**50**	**250**
15W40 no FM	0.177	0.174	0.168	0.177	0.177	0.170
5W20 high FM	0.092	0.093	0.086	0.086	0.084	0.089

Lab-Scale Testing Summary

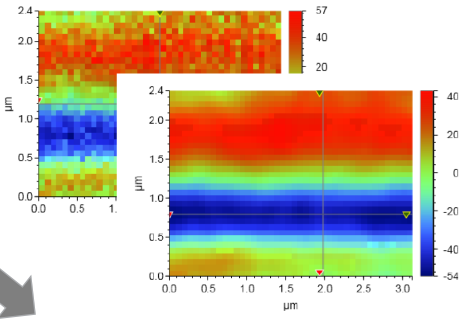
- Skirt-on-liner tests showed that steady-state was not reached by the end of tests of typical duration.
- Longer tests (24 hrs) were conducted.
  - An initial steady-state was reached (but then μ_{asp} fluctuates).
  - Boundary friction appears to vary according tribochemical film formation (a very dynamic process based on additive chemistry interactions), and material changes (graphite/resin exposure).
  - These changes are driven at rates proportional to temperature and load.
- This effect must be considered when validating models against engine tests.

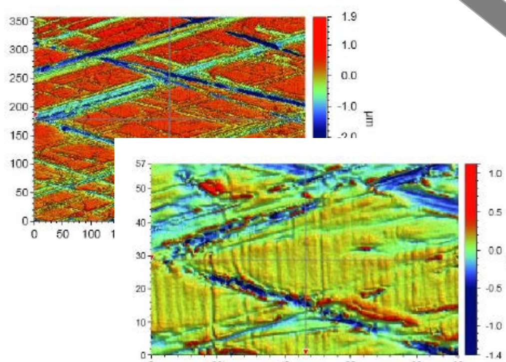


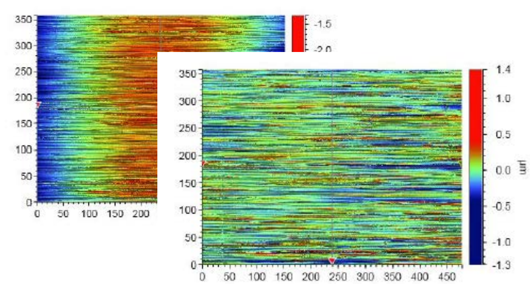


Surface Measurements

- A vital input to RINGPAK or PISDYN modeling is surface roughness characterization.
  - Greenwood-Trip (GWT) parameters are calculated from surface profiles.
  - GWT parameters are used to predict the contribution of the surface asperities to friction force.
- Considerable effort has been expended to determine
  - the best way to acquire surface profiles using white light interferometry
  - the best way to post-process surface images to get GWT parameters
- The following effects were studied using WLI:
  - Low pass filter cut-off
  - High pass filter cut-off/curvature removal
  - Magnification



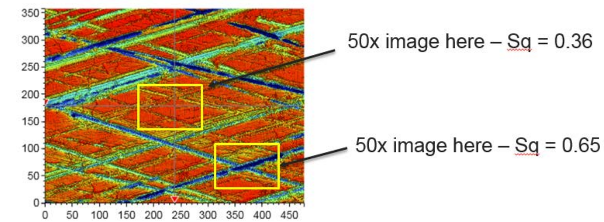




- Recommended Settings:
  - 10X magnification
  - 2.5 μm low pass filter
  - Data not sensitive to method for curvature removal but curvature must be removed nonetheless

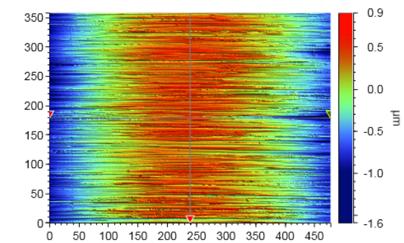
Surface Measurements

- The 2.5 μm Gaussian filtering limit is based on the expected size of asperity summits so that random noise within an individual pixel <u>is not</u> identified as summit.
    - A perfectly smooth surface will have an erroneously high summit count due to pixellation and the method by which a "summit" is defined.
    - A filtering scale F must be chosen to remove pixel-scale noise while revealing the true summit.
    - A rough estimate of the minimum size of a "real" summit is about 6 pixels = 2.5 μm.



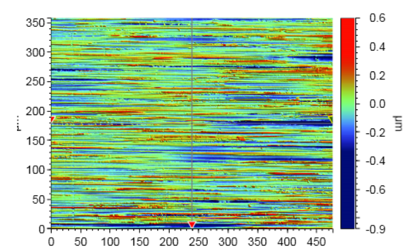
10x WLI image – Sq = 0.55 μm

- The 10x1 WLI magnification balances the need to retain detail yet obtain representative surface parameters for inhomogeneous surfaces such as honed cylinder liners.
    - A 50x magnification image sample provides an insufficient area to calculate representative surface parameters as shown in this example and a 10x is preferred.



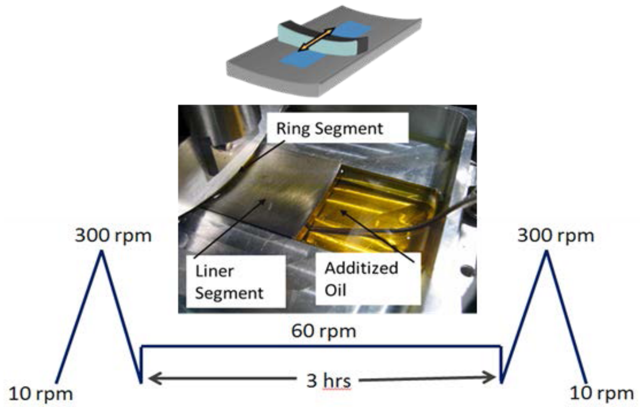
Label	Value	Units
Ssdn	45969.9264	1/mm²
Ssdn5	15131.4976	1/mm²
Ssmh	0.3593	μm
Ssmh5	0.1311	μm
Ssmr	40.2203	μm
Ssmr5	19.4719	μm
Sstd	0.176	μm
Sstd5	93.5016	nm

- The decision to remove curvature from WLI scans was arrived at based on the difference in surface parameters w/ and w/o curvature removed.
    - Without surface curvature removed the surface mean height is estimated as larger than actual and thus the algorithm "misses" some asperity summits artificially lowering the summit density.

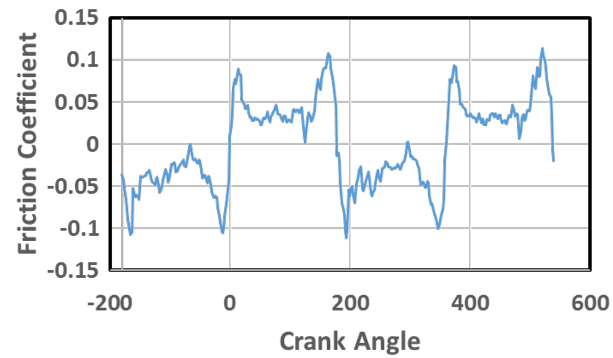


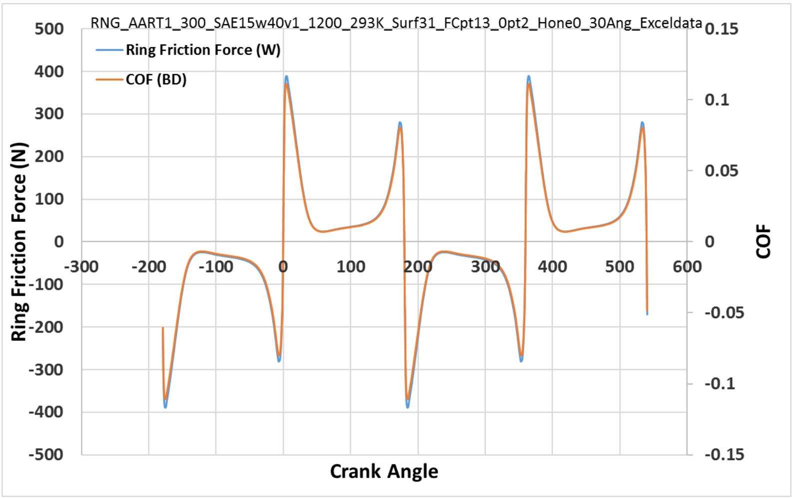
Label	Value	Units
Ssdn	64313.2672	1/mm²
Ssdn5	18114.3472	1/mm²
Ssmh	0.1374	μm
Ssmh5	74.5822	nm
Ssmr	39.5264	μm
Ssmr5	12.4778	μm
Sstd	0.1044	μm
Sstd5	60.0282	nm

RINGPAK Modeling



AART Case (50 N, RT, 300 rpm)





- AART RINGPAK model is being developed to predict experimental data and to understand sensitivity to
  - Temperature, Viscosity, Speed
  - Ring Tension
  - Surface Finish (GWT Parameters)
  - Honing grooves
- Comparing plots shows
  - COF moderately matches test data at min/max stroke (low velocity)
  - COF under predicted at mid-stroke (peak velocity)

RINGPAK Modeling

- Sensitivity study conducted around the following operating condition:
  - 50 N Load, 25 °C, 300 rpm, μ_{asp} = 0.13, S = 20 mm, 15W40
- Initial sensitivity study examined the impact of GWT parameters (surface roughness), ring tension (equivalent to 50 N normal load applied over different contact width), oil film thickness, honing parameters, μ_{asp}.
- Obvious trends are observed:
  - Rougher surfaces lead to higher COF at min/max stroke; smoother surfaces lead to more hydrodynamic friction at mid-stroke
  - Insufficient oil film thickness either has minimal effect or leads to asperity dominated friction across entire stroke; results are very sensitive!
  - Higher ring tension (50 N applied over a smaller contact patch) leads to higher COF at min/max stroke
- At this time best results are shown with larger than nominal ring tension, worn surface (smooth), sufficient film thickness, and larger than nominal μ_{asp}.
- Initial studies imply getting the oil film correctly during the mid-stroke where hydrodynamic friction dominates is key.

Challenges/Barriers

- CAE modeling is <u>confined</u> in scope to power cylinder components: ring and piston
- However, lubricant changes affect the friction contributions from all components wetted by engine oil
- Thus, FMEP measurements account for all contributors unless they are explicitly removed from the test
- This requires that an appropriate means to separate out the impact of lubricant changes on engine friction and fuel consumption realized through other components, e.g., main bearings, valve train, etc. be developed

Next Steps

- Start accelerated wear tests in support of next GO/NO GO decision point
  - Procure coated rings and pistons
- Perform engine thermal survey at key operating points
  - Instrument cylinder block and piston with thermocouples/templugs
- Develop and execute test plan for motored and fired engine friction tests
- Investigate Ricardo's FAST friction tool as possible method for estimating friction contributions from non power cylinder components

Summary

- Oil candidates for subsequent lab-scale and dyno testing have been identified per first GO/NO point.
- μ_{asp} and surface parameters (which are vital inputs for the RINGPAK/PISDYN simulations) for the base components have been obtained.
- RINGPAK model of the reciprocating tribometer has been created and validation of the model setup is in progress.

Partners/Collaborators

RICARDO

RICARDO	Technical lead responsible for project management and engine dyno testing, modeling and simulation.
Argonne NATIONAL LABORATORY	Responsible for lab-scale testing, data analysis and interpretation, modeling and simulation.
ISUZU	Partner providing in-kind contributions including the engine test platform, components for lab-scale and dyno testing, and consultation
Electro-Mechanical Associates	Sub-contractor providing additional lab-scale testing
ZYNP	Partner providing in-kind contributions of components for lab-scale testing