ANL-15/12

Argonne Engine Friction Study Phase 1 Final Report

Energy Systems Division

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Argonne Engine Friction Study Phase 1
Final Report

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ARGONNE ENGINE FRICTION STUDY PHASE 1

FINAL REPORT

EXECUTIVE SUMMARY

Argonne National Laboratory (ANL) has developed a process for making near frictionless carbon (NFC) coatings and depositing them on metal substrates. Friction reductions approaching an order of magnitude have been measured in laboratory tests. While there are many potential applications for such coatings, friction reduction in internal combustion engines is of particular interest due to the apparent fuel savings potential. Ricardo has performed a program of work to estimate potential fuel economy improvements due to the application of such a coating at key interfaces within a diesel engine typical of those found in large trucks.

Piston, ring pack, and valvetrain simulations have been performed, using existing models of representative engines, with various degrees of friction reduction applied at important interfaces. The simulations were run at 8 specific operating points to allow approximation of engine performance over the FTP test cycle. Reduction in fuel consumption over the cycle was calculated for each reduced friction case.

Results show that application of a friction-reducing surface treatment, like the NFC coatings, at the piston rings and skirt, and at key interfaces within the valvetrain, is expected to result in a reduction in fuel consumption of 0.43% to 0.81% over the FTP heavy duty test cycle. The piston skirt and piston rings are the interfaces where the coating can make the most difference, assuming no changes are made to the engine lubricant.

Hydrodynamic friction represents a very large fraction of friction losses within the interfaces considered, at all operating conditions, indicating that changes to the engine lubricant, such as reduced viscosity, can result in further improvement. Reduced oil viscosity may result in increased metal-to-metal contact and wear, unless a durable, low friction coating can be applied at key interfaces. Ricardo recommends an analytical evaluation of the potential benefits of reduced oil viscosity, which considers potential increases in wear loads at key interfaces.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

1. INTRODUCTION

Argonne National Laboratory (ANL) has developed a process for making near frictionless carbon (NFC) coatings and depositing them on metal substrates. The amorphous carbon coatings exhibit properties comparable to diamond. Friction reductions approaching an order of magnitude have been measured in laboratory tests. While there are many potential applications for such coatings, friction reduction in internal combustion engines is of particular interest due to the apparent fuel savings potential.

The U.S. Department of Energy's (DOE) Office of Heavy Vehicle Technology (OHVT) has estimated energy loss due to friction in various vehicles and specific engine components (pistons, rings, and connecting rods). For example, for heavy duty (HD) vehicles the estimated energy loss is equivalent to 160 million barrels of diesel fuel per year. It is assumed that this loss could be reduced through the selective use of NFC coatings on engine components, but there is relatively little guidance available on where to employ such coatings in a manner that is both practical and effective. To properly assess the potential value of Argonne's NFC coatings, their impact on engine friction needs to be quantified and their effect on engine design, production, and durability issues needs to be assessed.

Ricardo has performed a program of work to assess the potential for fuel savings through friction reduction at key engine interfaces. This program of work is outlined in Ricardo Statement of Work G0325, and the results of this work are presented in the sections below.

2. OBJECTIVES

The objective of this work was to estimate the potential for reduction in fuel consumption due to the reduction of friction at key engine interfaces within a heavy-duty diesel engine.

3. TECHNICAL APPROACH

A schematic representation of the overall technical approach is shown in Figure 1. The following paragraphs summarize the methodology.

3.1 Engine Type

The engine type studied was a diesel typical of those found in large trucks. In a valve line typical of these engines, each cam lobe is contacted by the rolling portion of a roller follower. The other end of the roller follower moves a pushrod, which actuates a rocker arm. The rocker arm pivots upon a rocker shaft, with the other end of the arm pressing against a valve bridge. Each valve bridge, and thus each cam lobe, operates two valves. Figure 2 shows an engine with a similar valvetrain layout, but without the roller follower and valve bridge. The engine type studied would typically use an articulated piston, comprised of a separate skirt and crown, with two compression rings to seal gases within the cylinder, and one oil retention ring to assist in the removal of excess oil from the cylinder wall.

3.2 Mean Effective Pressures

To compare the performance of engines of various sizes, it is common to use mean effective pressure, or MEP. Mean effective pressure is independent of engine size or speed. MEP is a representative pressure that is required within an engine, during the expansion stroke, to produce a particular power output at a particular speed. Cylinder pressure over the remaining strokes is assumed to be zero for this theoretical cycle. In other words, MEP is work per cycle divided by the volume displaced during the cycle. Several different types of MEP were used in the course of this study to simplify comparisons between engines:

- Indicated mean effective pressure (IMEP) is based solely on the cylinder pressure throughout the cycle. It is related to the maximum amount of work that could be produced in an engine with no mechanical losses.
- Brake mean effective pressure (BMEP) is based on the power output of an engine, and takes into account any mechanical losses within the engine. BMEP is a value that is often used to describe how highly rated an engine is, or how heavily it is loaded at a particular operating condition.
- Friction mean effective pressure (FMEP) is based on the difference between IMEP and BMEP, and represents the mechanical losses within the engine.

3.3 Ricardo 8-Mode FTP Simulation

For estimates of fuel savings to be useful, they need to be representative of what can be achieved in real-world operation. The FTP heavy-duty transient test cycle is a test used for emissions testing of engines in the USA. The test is intended to be representative of the load conditions that the engine in a heavy-duty vehicle, such as a truck or bus, would see in operation. The cycle includes both city and highway segments. A cold start segment, periods of acceleration and deceleration, idling, and motoring are incorporated into the test. Typical engine loads are 20-25% of the available power at a given speed, and there are few periods of steady operation.

Testing an engine on the FTP heavy-duty transient test cycle requires special equipment and calibration, and is thus expensive to perform. Ricardo has developed a test procedure to allow the evaluation of engine performance over the FTP cycle using more conventional test equipment. The Ricardo 8-Mode FTP simulation is a steady-state, non-motored, test procedure consisting of eight modes designed to represent key regimes of the FTP cycle. Emissions and fuel consumption at each mode are multiplied by a weighting factor, and the results are summed to give an overall outcome. The results of the 8-mode simulation closely correlate with results of the FTP cycle, making the 8-mode simulation a comparatively inexpensive way to gage engine performance over the FTP cycle. The detailed nature of the FTP heavy-duty transient test cycle makes it impractical to model the cycle directly for this study. Instead, simulations have been performed at 8 different load/speed points consistent with the Ricardo 8-Mode test.

The predicted reduction in FMEP at each operating point was then considered together with measured data from an 8-Mode engine test to predict changes in fuel consumption.

3.4 Interfaces Considered

The engine interfaces analyzed for this study were the following:

- Piston skirt to cylinder liner
- Piston rings to cylinder liner
- Cam to cam bearings
- Cam to follower
- Pushrod to rocker arm
- Rocker arm to valve bridge

The interfaces were chosen because of the expected significance of their contributions to engine friction, as well as the existence of appropriate models from previous engine analyses. Figures 2 and 3 show pictures of similar components.

3.5 Friction Reductions

The effects of a surface treatment on real-world friction at an internal engine interface are nearly impossible to predict through laboratory or analytical methods alone. Thus, for this study, varying degrees of friction reduction were considered, to bracket the potential for real-world improvement. Friction reductions for each interface were based on consideration of laboratory friction test results, expected conditions at the interface, and experience with motoring friction measurements of existing engines. Table 1 shows a summary of the typical friction coefficients that Ricardo would apply at each interface, and the reduced coefficients applied at that interface. The ring-to-liner and skirt-to-liner friction coefficients were reduced by 90%, 60%, and 30%, while the valvetrain interface friction coefficients were reduced by 20%, 10%, and 5%. Based on friction reduction results observed at Argonne, real-world friction reductions due to application of the NFC coatings are expected to fall between the 30% and 60% cases for the piston skirt and ring pack, and between 5% and 10% for the valvetrain interfaces.

3.6 PISDYN

The Ricardo software PISDYN was used to simulate the friction behavior of the piston skirt as it travels up and down the cylinder liner, and friction in the wrist pin bearings. PISDYN is a time-domain simulation of piston secondary dynamic motion, which treats, via detailed models, hydrodynamic and boundary lubrication at the skirt-liner interface and the wrist pin bearings. Elasticity of the skirt and/or cylinder liner are considered. An elasto-hydrodynamic lubrication (EHL) code predicts oil film pressure, using a mass-conserving solution of the Reynolds Equation, at each node of a matrix covering the surface of the piston skirt. The remainder of the reaction force at each node is calculated, using a Greenwood-Tripp model, and attributed to asperity contact between the skirt and liner. Hydrodynamic losses are calculated by the EHL code, and asperity friction is calculated using a friction coefficient and contact pressures across the skirt matrix. Wrist pin bearing lubrication and friction are calculated in a manner similar to the aforementioned.

For this study, an existing PISDYN model from a previous analysis of a heavy-duty diesel engine (Engine "A") was used. Engine "A" has an inline 6 cylinder configuration. It incorporates an articulated piston design and has an overall displaced volume of 10 liters. A baseline series of runs, representative of the Ricardo 8-Mode test, was executed, using the friction coefficient that Ricardo would typically apply. The asperity friction coefficient was reduced, in separate cases, by 90%, 60%, and 30%. The actual amount of friction reduction at this interface is expected, based on experimental results, to be in the range of 30% to 60%. The 90% case was run as a best case scenario. Predicted reduction in power loss due to piston skirt friction was tabulated, for simulations at each of the 8-modes of the Ricardo 8-mode FTP cycle simulation, for each of the 4 friction coefficient cases.

3.7 RINGPAK

The Ricardo software RINGPAK was used to calculate the friction between the rings and liner. RINGPAK is a time-domain simulation of ring motion, and addresses both hydrodynamic and boundary lubrication at the ring-liner interfaces. Lubricant pressures at these interfaces are obtained through the implementation of a mass-conserving scheme to solve the Reynolds Equation. When ring-liner clearances are small, a Greenwood-Tripp model is used to evaluate the asperity contact pressures. Hydrodynamic losses are calculated for the lubricant, and ring asperity friction is calculated using contact pressures and a friction coefficient.

For this study, an existing RINGPAK model from a previous analysis of a heavy-duty diesel engine (Engine "A") was used. . Engine "A" has an inline 6 cylinder configuration. It incorporates an articulated piston design and has an overall displaced volume of 10 liters. A baseline series of RINGPAK runs, representative of the Ricardo 8-Mode test, was executed, using the ring-liner asperity friction coefficient that Ricardo typically applies. The asperity friction coefficient was reduced, in separate cases, by 90%, 60%, and 30%. The actual amount of friction reduction at this interface is expected, based on experimental results, to be in the neighborhood of 30% to 60%. . Predicted reduction in power loss due to ring friction were tabulated, for simulations at each of the 8-modes of the Ricardo 8-mode FTP cycle simulation, for each of the 4 friction coefficient cases.

3.8 VALDYN

The Ricardo software VALDYN was used to calculate the friction at key interfaces within the valvetrain. VALDYN is a time-domain simulation of valvetrain dynamics. Friction is addressed in VALDYN via simple friction coefficients and calculated normal forces.

An existing VALDYN model from an analysis of a heavy-duty diesel engine (Engine "B") was used for this study. Engine "B" has an inline 6-cylinder configuration, and has an overall displaced volume of 9 liters. The valvetrain employs a roller-follower, and each cam lobe operates two valves. Exhaust gas pressure was applied at the exhaust valves to increase loading of the valvetrain. Friction coefficients within the valvetrain were reduced, in separate cases, by 20%, 10%, and 5%. The actual amount of friction reduction at this interface is expected, based on experimental results, to be in the range of 5% to 10%. Valvetrain simulations were run, for each of the reduced friction cases, at each of the 8 modes of the Ricardo 8-mode FTP cycle simulation. Predicted cam drive power was tabulated for each simulation.

3.9 Calculation of Fuel Savings

Reductions in FMEP were calculated based on the predicted friction power reductions from the PISDYN, RINGPAK, and VALDYN, simulations. IMEP minus change in FMEP, divided by IMEP, for each of the 8 modes, gives a scaling factor for fuel consumption at each mode. Data from a previous test of a heavy-duty diesel engine (Engine "C") was used as a baseline. Engine "C" has an overall displacement of 11 liters. The valvetrain employs a roller-follower, and each cam lobe operates two valves. The engine incorporates an articulated piston design. The fuel consumption of Engine "C," at each test point, was scaled by the aforementioned scaling factor, to give estimated fuel consumption for the engine with NFC coatings. Application of the weighting factors allows a calculation of estimated change in overall fuel consumption for the FTP cycle.

4.0 RESULTS

The friction power losses predicted for the baseline cases fall in line with experimental measurements of motoring friction. See Figures 4 and 5 for a comparison of predicted friction losses versus previously measured motoring friction data. The high BMEP cases (modes 2, 3, 5, 8) predict higher friction losses at the piston skirt and rings than what was measured in motoring friction experiments, which is to be expected due to increased loading of the interfaces.

The predicted contributions of hydrodynamic and boundary friction to FMEP at the piston skirt are shown in Figures 6-9. Hydrodynamic friction tends to increase with increased piston speed. Asperity friction is dramatically increased in the high-load cases. The particular piston skirt analyzed had a larger-than-desired amount of skirt contact and wear in the high load cases, but the design is in use in production engines, and such contact is not atypical. The reductions in friction coefficient offered the biggest benefit in the high load cases, where the most asperity contact was predicted.

The contributions of hydrodynamic and boundary friction to FMEP at the ring-liner interfaces are shown in Figures 10-13. Hydrodynamic friction increased with increased piston speed, and with increased engine load. Asperity friction was reduced at higher piston speeds due to increased oil film thickness.

While the total friction at the rings was larger than the total friction at the piston skirt, asperity friction at the skirt was larger than that at the rings, and thus the piston skirt showed a greater improvement due to the coatings.

Predicted reductions in skirt friction power for each level of friction reduction, at each operating point, are shown in Table 2. Overall reductions in fuel consumption due to friction reduction at the piston skirt-cylinder liner interface were 0.24%, 0.46%, and 0.68%. Reductions in ring-liner friction power are shown in Table 3. Overall reductions in fuel consumption due to friction reduction at the ring-liner interfaces were 0.13%, 0.23%, and 0.43%. Reductions in cam drive power are shown in Table 4. Overall reductions in fuel consumption due to friction reduction in the valvetrain were 0.06%, 0.12%, and 0.24%.

With the combination of 90% friction reduction at the skirt, 90% at the rings, and 20% at the valvetrain interfaces, a net 1.35% reduction in fuel consumption is predicted. With 60%, 60%, and 10%, the predicted improvement in fuel economy is 0.81%, and with 30%, 30%, and 5%, the improvement is 0.43%. Table 5 summarizes these results.

5.0 CONCLUSIONS AND RECOMMENDATIONS

Application of a friction-reducing surface treatment, like the NFC coatings, at the piston rings and skirt, and at key interfaces within the valvetrain, is expected to result in a reduction in fuel consumption of 0.43% to 0.81% over the FTP heavy duty test cycle. The piston skirt and piston rings are the interfaces where the coating can make the most difference, assuming no changes are made to the engine lubricant.

Hydrodynamic friction represents a very large fraction of friction losses within the interfaces considered, at all operating conditions, indicating that changes to the engine lubricant, such as reduced viscosity, can result in further improvement. Reduced oil viscosity may result in increased metal-to-metal contact and wear, unless a durable, low friction coating can be applied at key interfaces. Ricardo recommends an analytical evaluation of the potential benefits of reduced oil viscosity, which considers potential increases in wear loads, at key interfaces.

If additional funding can be identified, it would be useful to conduct a friction teardown test on a heavy-duty diesel truck engine. This would involve running the engine on a test stand and measuring fuel consumption. The engine would then be motored and total friction losses would be measured. Components would be removed from the engine in a systematic way, and after removal of each component total friction would be measured while the engine is being motored. In this way, the contribution of each part to total friction can be measured by difference.

If funds are available and ANL has the facilities to coat engine parts, the components removed during the teardown test could be coated with appropriate NFC coatings and reassembled into the engine. In doing this, the friction measurements could be made in reverse to quantify the effects of the coatings on specific parts. The rebuilt engine could be run to measure fuel consumption directly and for comparison with the initial measurement from the uncoated engine.

Table 1: Friction Coefficients at each Interface

Component	Runs against	Baseline Coefficient for Analysis	Reduced Friction Coefficients			Friction Model
Follower	Camshaft	0.005	0.00475	0.0045	0.004	Simple
Camshaft	Cam Bearing	0.02	0.019	0.018	0.016	Simple
Rocker Bushing	Rocker Shaft	0.02	0.019	0.018	0.016	Simple
Pushrod Socket	Pushrod	0.05	0.0475	0.045	0.04	Simple
Rocker Tip	Bridge	0.05	0.0475	0.045	0.04	Simple
Piston Skirt	Cylinder Liner	0.08	0.056	0.032	0.008	Detailed Boundary and Hydrodynamic Lubrication
Piston Ring		0.12	0.084	0.048	0.012	Detailed Boundary and Hydrodynamic Lubrication
Piston Pin	Piston Pin Bushing	0.08	0.056	0.032	0.008	Detailed Boundary and Hydrodynamic Lubrication

RICARDO

Table 2: Summary of Piston Skirt Friction Reduction Results

FTP Simulation Mode	1	2	3	4	5	6	7	8	
Speed (rpm)	750	960	1170	1590	1590	1800	1800	1800	
IMEP (kPa)	104	1683	1683	983	1683	325	983	1683	
BMEP (kPa)	0	1565	1565	783	1565	157	783	1565	
Fuel Consumption (kg)	1.766	28.084	48.419	28.793	58.144	10.224	31.094	57.313	
Weighting Factor	0.524	0.03	0.039	0.149	0.075	0.057	0.081	0.045	
Weighted Fuel Consumption (kg)	0.925	0.843	1.888	4.290	4.361	0.583	2.519	2.579	
Reduction in Friction Coefficient	Piston Skirt Friction Power (kW)								
0%	0.100	0.464	0.577	0.371	0.703	0.440	0.458	0.697	
30%	0.096	0.351	0.440	0.364	0.558	0.440	0.454	0.572	
60%	0.092	0.239	0.308	0.359	0.421	0.440	0.451	0.451	
90%	0.088	0.129	0.180	0.355	0.279	0.439	0.448	0.331	
	Piston Skirt FMEP (kPa)								
0%	9.3	33.7	34.4	16.3	30.9	17.1	17.8	27.0	
30%	8.9	25.5	26.3	16.0	24.5	17.1	17.6	22.2	
60%	8.5	17.4	18.3	15.8	18.5	17.1	17.5	17.5	Expected Reduction in Fuel Consumption over FTP Cycle (%)
90%	8.1	9.4	10.7	15.6	12.3	17.0	17.4	12.8	
	Reduction in Fuel Consumption (%)								
0%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%
30%	0.34%	0.49%	0.49%	0.03%	0.38%	0.00%	0.02%	0.29%	0.24%
60%	0.74%	0.97%	0.96%	0.05%	0.74%	0.00%	0.03%	0.57%	0.46%
90%	1.11%	1.45%	1.41%	0.07%	1.10%	0.02%	0.04%	0.84%	0.68%

Table 3: Summary of Ring Pack Friction Reduction Results

FTP Simulation Mode	1	2	3	4	5	6	7	8	
Speed (rpm)	750	960	1170	1590	1590	1800	1800	1800	
IMEP (kPa)	104	1683	1683	983	1683	325	983	1683	
BMEP (kPa)	0	1565	1565	783	1565	157	783	1565	
Fuel Consumption (kg)	1.766	28.084	48.419	28.793	58.144	10.224	31.094	57.313	
Weighting Factor	0.524	0.03	0.039	0.149	0.075	0.057	0.081	0.045	
Weighted Fuel Consumption (kg)	0.925	0.843	1.888	4.290	4.361	0.583	2.519	2.579	
Reduction in Friction Coefficient	Ring Pack Friction Power (kW)								
0%	0.335	0.514	0.648	0.995	1.050	1.130	1.170	1.260	
30%	0.315	0.493	0.655	0.982	1.030	1.120	1.160	1.250	
60%	0.304	0.473	0.635	0.969	1.020	1.110	1.150	1.240	
90%	0.273	0.449	0.615	0.952	1.000	1.100	1.140	1.220	
	Ring Pack FMEP (kPa)								
0%	31.2	37.4	38.7	43.7	46.1	43.8	45.4	48.9	
30%	29.3	35.8	39.1	43.1	45.2	43.4	45.0	48.5	
60%	28.3	34.4	37.9	42.5	44.8	43.0	44.6	48.1	Expected Reduction in Fuel Consumption over FTP Cycle (%)
90%	25.4	32.6	36.7	41.8	43.9	42.7	44.2	47.3	
	Reduction in Fuel Consumption (%)								
0%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%
30%	1.79%	0.09%	-0.02%	0.06%	0.05%	0.12%	0.04%	0.02%	0.13%
60%	2.77%	0.18%	0.05%	0.12%	0.08%	0.24%	0.08%	0.05%	0.23%
90%	5.55%	0.28%	0.12%	0.19%	0.13%	0.36%	0.12%	0.09%	0.43%

Table 4: Summary of Valvetrain Friction Reduction Results

FTP Simulation Mode	1	2	3	4	5	6	7	8	
Speed (rpm)	750	960	1170	1590	1590	1800	1800	1800	
IMEP (kPa)	104	1683	1683	983	1683	325	983	1683	
BMEP (kPa)	0	1565	1565	783	1565	157	783	1565	
Fuel Consumption (kg)	1.766	28.084	48.419	28.793	58.144	10.224	31.094	57.313	
Weighting Factor	0.524	0.03	0.039	0.149	0.075	0.057	0.081	0.045	
Weighted Fuel Consumption (kg)	0.925	0.843	1.888	4.290	4.361	0.583	2.519	2.579	
Reduction in Friction Coefficient	Valvetrain Friction Power (kW)								
0%	0.173	0.233	0.299	0.449	0.449	0.533	0.533	0.533	
5%	0.169	0.228	0.293	0.440	0.440	0.522	0.522	0.522	
10%	0.164	0.223	0.287	0.431	0.431	0.513	0.513	0.513	
20%	0.156	0.212	0.274	0.413	0.413	0.494	0.494	0.494	
	Valvetrain FMEP (kPa)								
0%	19.1	20.1	21.2	23.4	23.4	24.5	24.5	24.5	
5%	18.6	19.7	20.7	22.9	22.9	24.1	24.1	24.1	
10%	18.2	19.2	20.3	22.5	22.5	23.6	23.6	23.6	Expected Reduction in Fuel Consumption over FTP Cycle (%)
20%	17.3	18.3	19.4	21.5	21.5	22.7	22.7	22.7	
	Reduction in Fuel Consumption (%)								
0%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%
5%	0.44%	0.03%	0.03%	0.05%	0.03%	0.14%	0.05%	0.03%	0.06%
10%	0.88%	0.05%	0.05%	0.09%	0.05%	0.29%	0.09%	0.06%	0.12%
20%	1.74%	0.11%	0.10%	0.19%	0.11%	0.55%	0.18%	0.11%	0.24%

Table 5: Summary of Overall Fuel Consumption Results

FTP Simulation Mode	1	2	3	4	5	6	7	8
Time (min)	60	60	60	60	60	60	60	60
Fuel Mass (kg)	1.766	28.084	48.419	28.793	58.144	10.224	31.094	57.313
BMEP (kPa)	0	1638	2213	1033	2045	191	919	1783
Weightings	0.524	0.03	0.039	0.149	0.075	0.057	0.081	0.045
Weighted fuel consumption (kg)	0.93	0.84	1.89	4.29	4.36	0.58	2.52	2.58
							Total fuel for 1 hour:	17.99 kg

With 90-90-20 coatings (Rings - Skirt - Valvetrain % Reduction)

(IMEP + Delta FMEP) / IMEP	0.916	0.982	0.984	0.995	0.987	0.991	0.997	0.990
Scaled Fuel Consumption (kg)	0.848	0.827	1.858	4.271	4.302	0.577	2.510	2.552
							Total fuel for 1 hour:	17.74 kg
Change (kg)	-0.24							
Change	-1.35%							

With 60-60-10 coatings (Rings-Skirt-Valvetrain % Reduction)

(IMEP + Delta FMEP) / IMEP	0.956	0.988	0.989	0.997	0.991	0.995	0.998	0.993
Scaled Fuel Consumption (kg)	0.885	0.832	1.868	4.279	4.323	0.580	2.514	2.562
							Total fuel for 1 hour:	17.84 kg
Change (kg)	-0.15							
Change	-0.81%							

With 30-30-05 coatings (Rings-Skirt-Valvetrain % Reduction)

(IMEP + Delta FMEP) / IMEP	0.974	0.994	0.995	0.999	0.995	0.997	0.999	0.997
Scaled Fuel Consumption (kg)	0.902	0.837	1.879	4.284	4.341	0.581	2.516	2.570
							Total fuel for 1 hour:	17.91 kg
Change (kg)	-0.08							
Change	-0.43%							

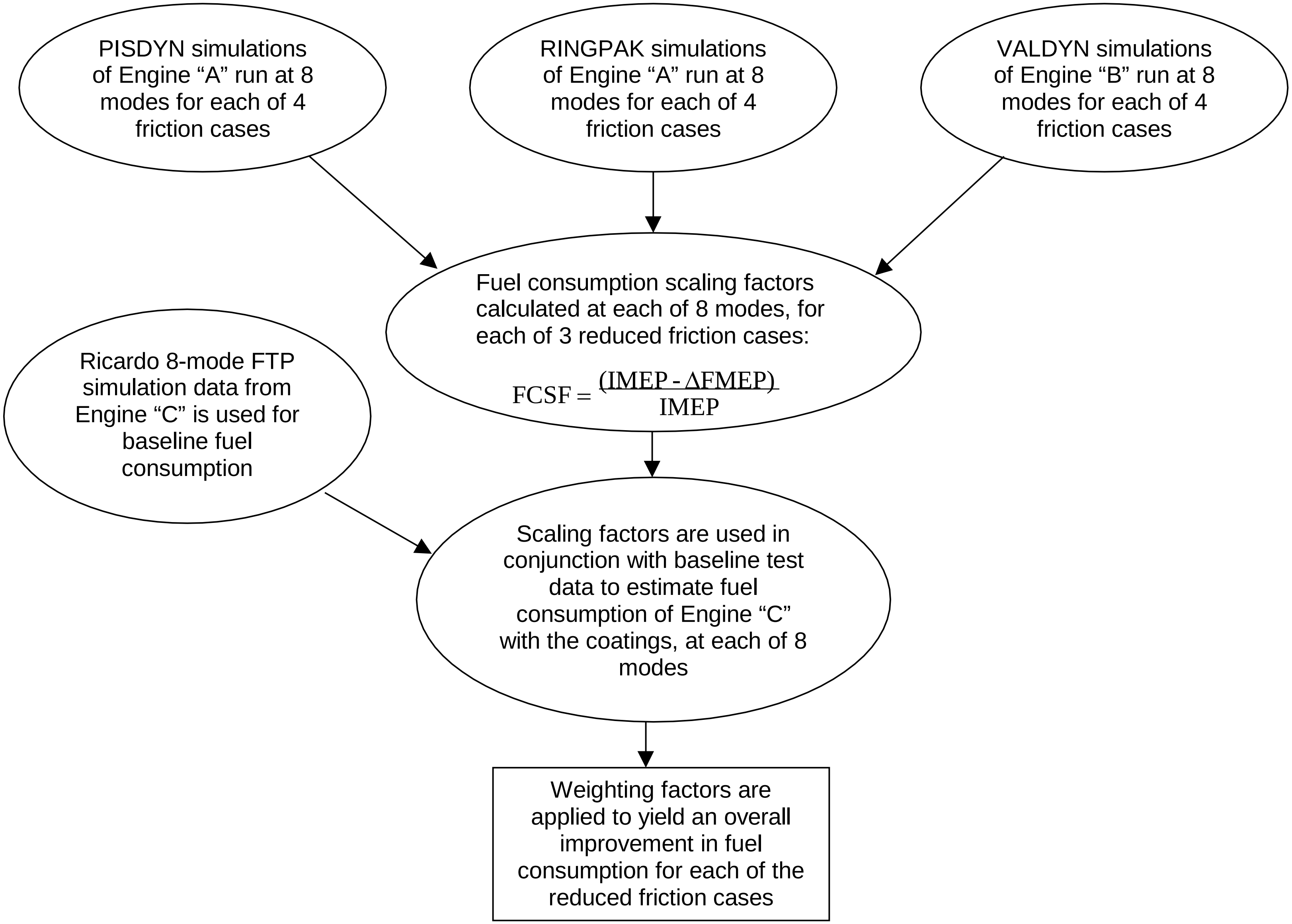

Figure 1: Diagram of Overall Approach

– Phase 1 focused on piston assembly and valvetrain
– Existing piston, ring, and valvetrain models available
– Good potential for asperity friction reduction due to NFC coatings

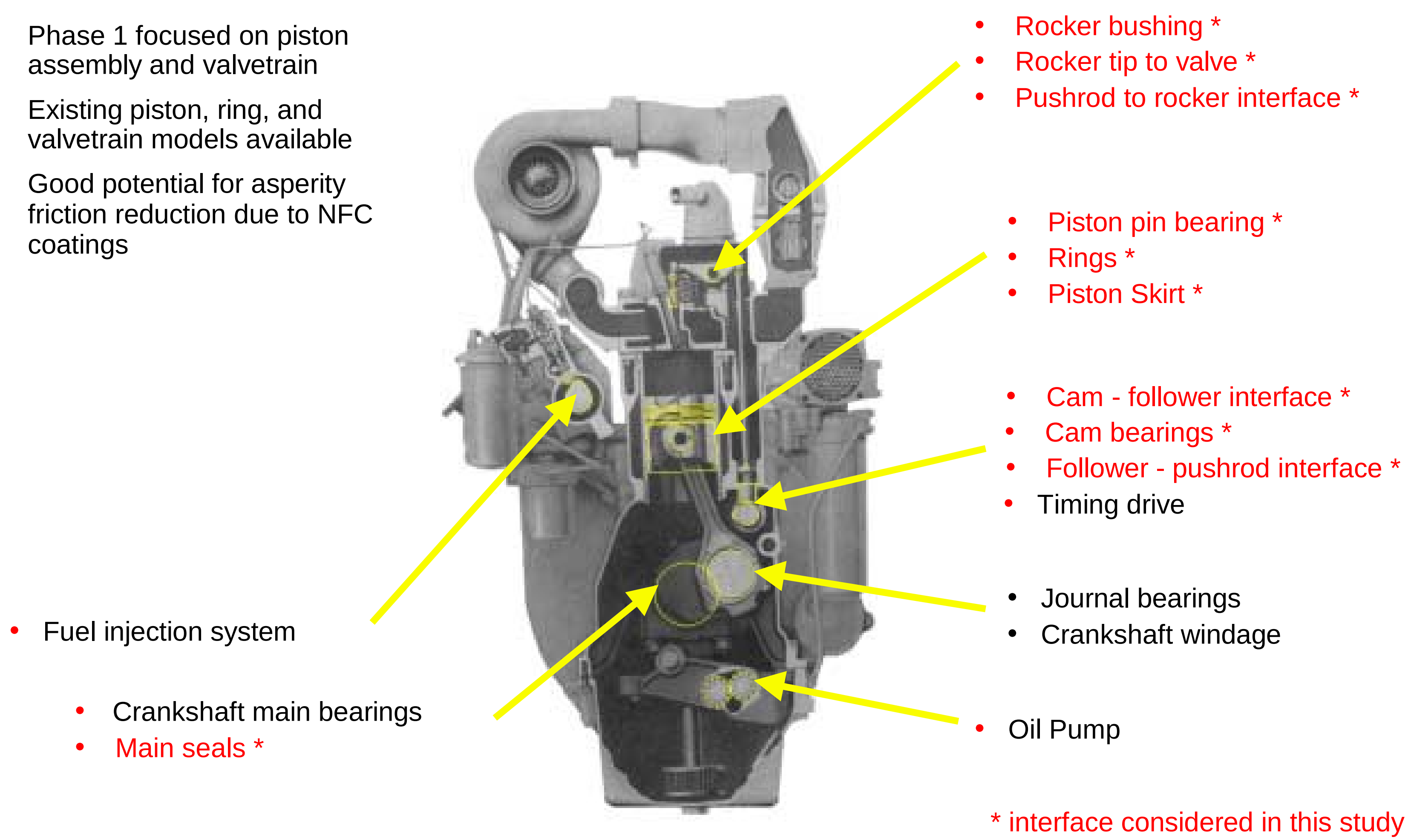

Figure 2: Significant Friction Sources

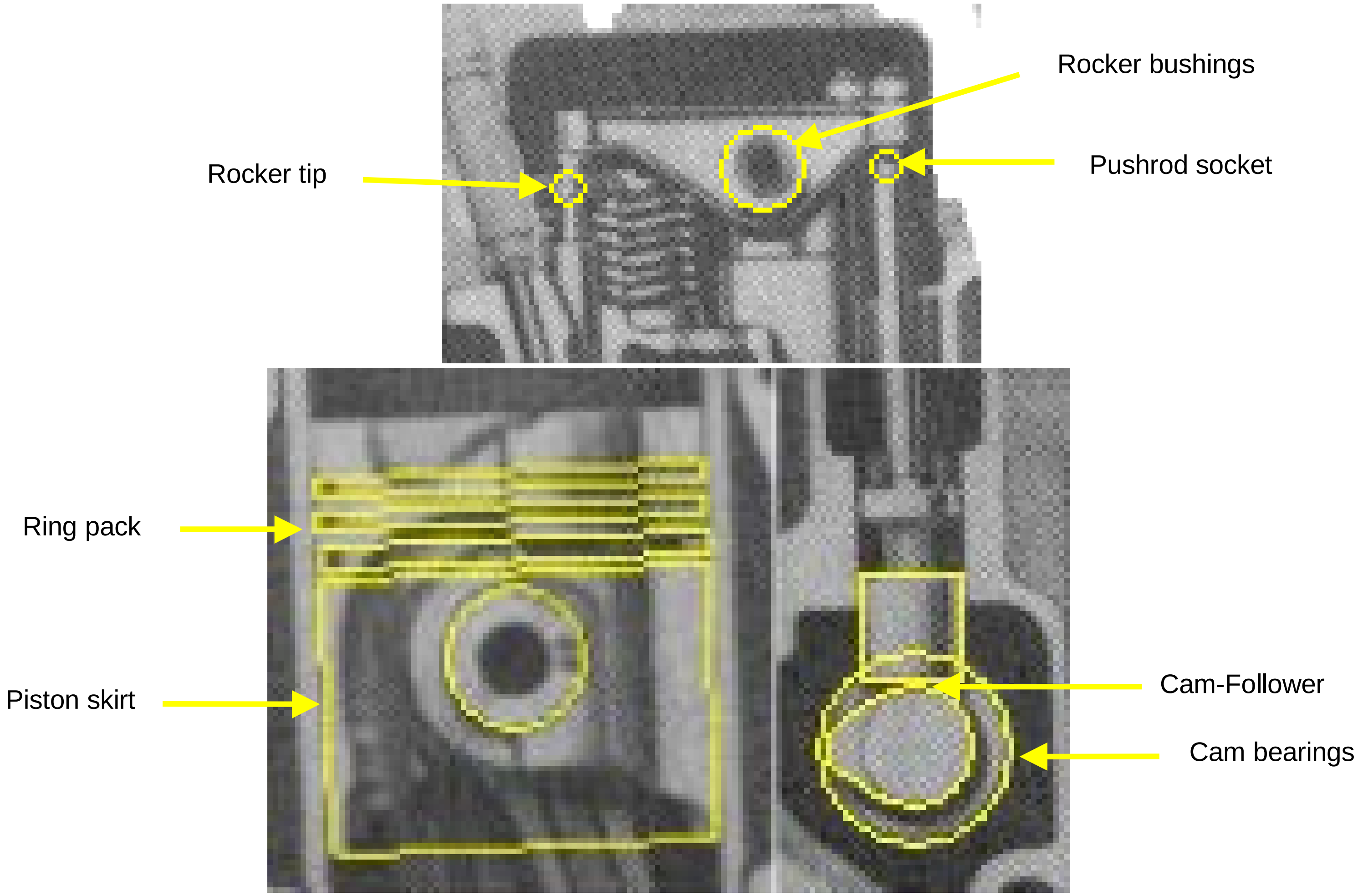

Figure 3: Close-up view of interfaces modeled for this study

Figure 4: Piston Assy Motoring Friction vs. Mean Piston Speed (Diesel Engines)

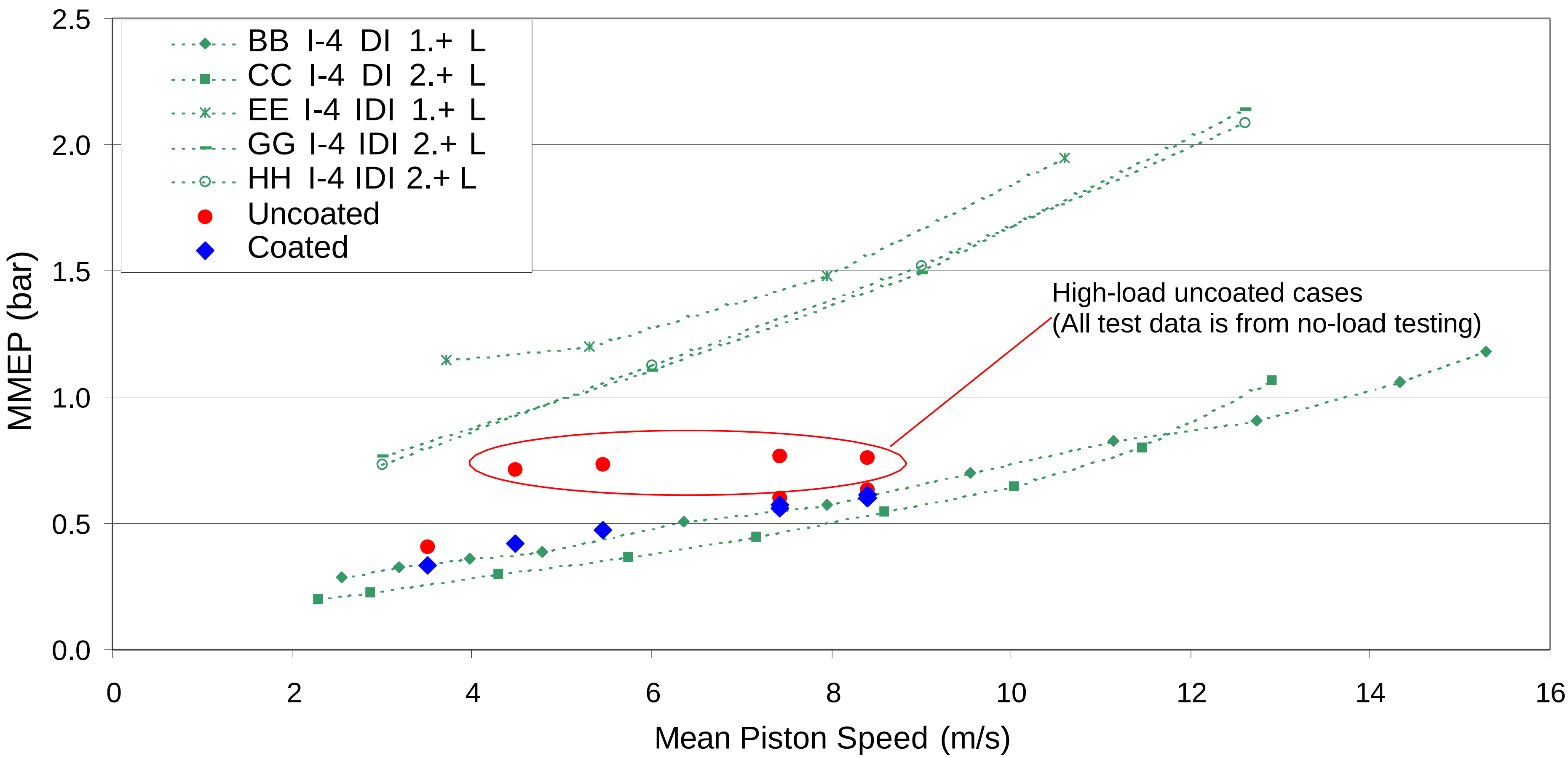

Figure 5: Total Valvetrain Motoring Friction vs. Mean Piston Speed (Diesel Engines)

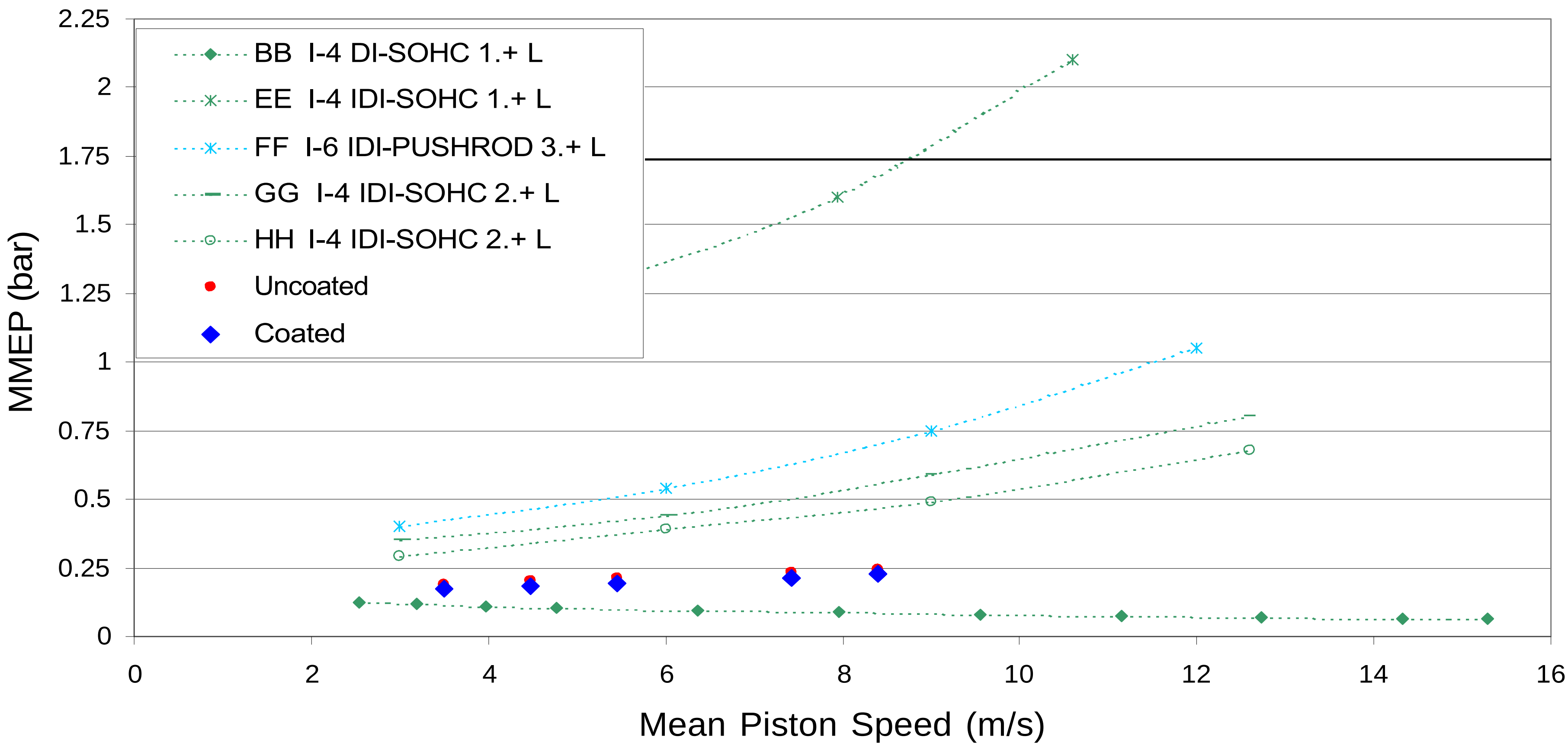

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Figure 6: Piston Skirt Friction Contributions (Uncoated)

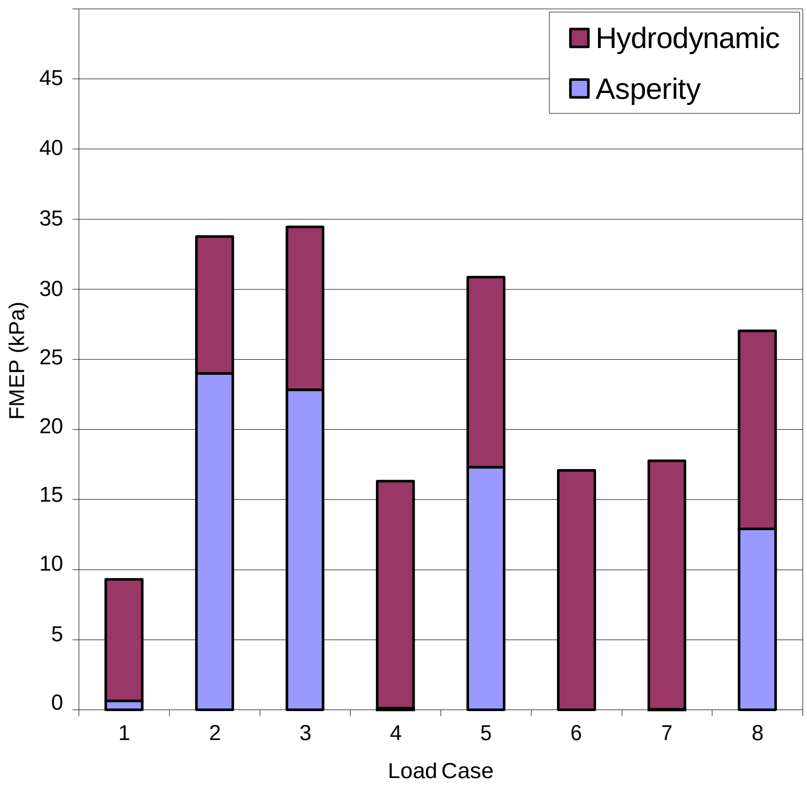


Figure 7: Piston Skirt Friction Contributions (30% Reduction in Asperity Friction Coefficient)

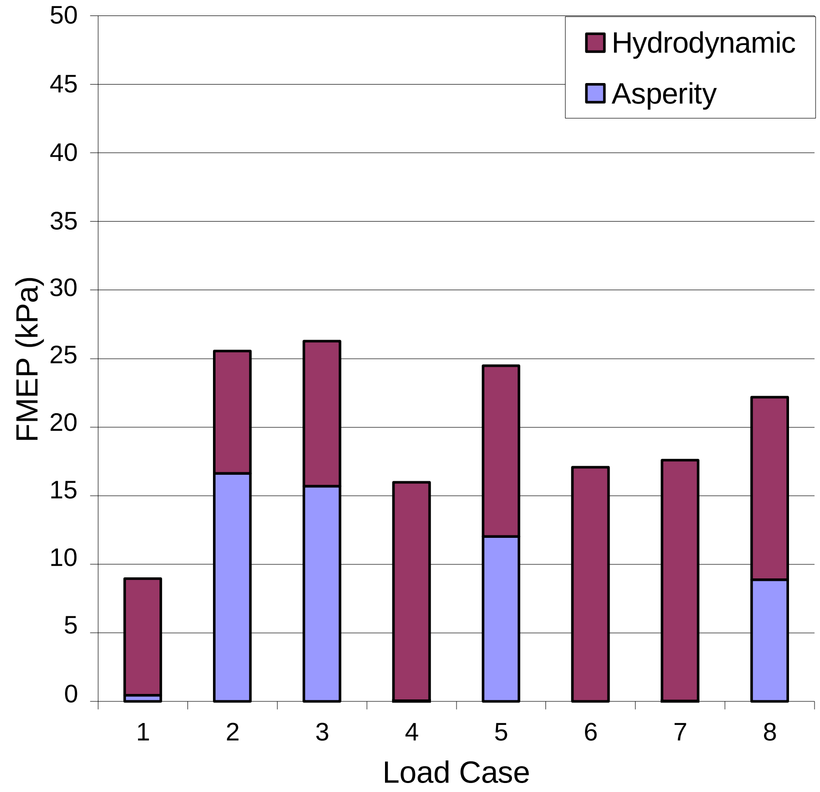


Figure 8: Piston Skirt Friction Contributions
(60% Reduction in Asperity Friction Coefficient)

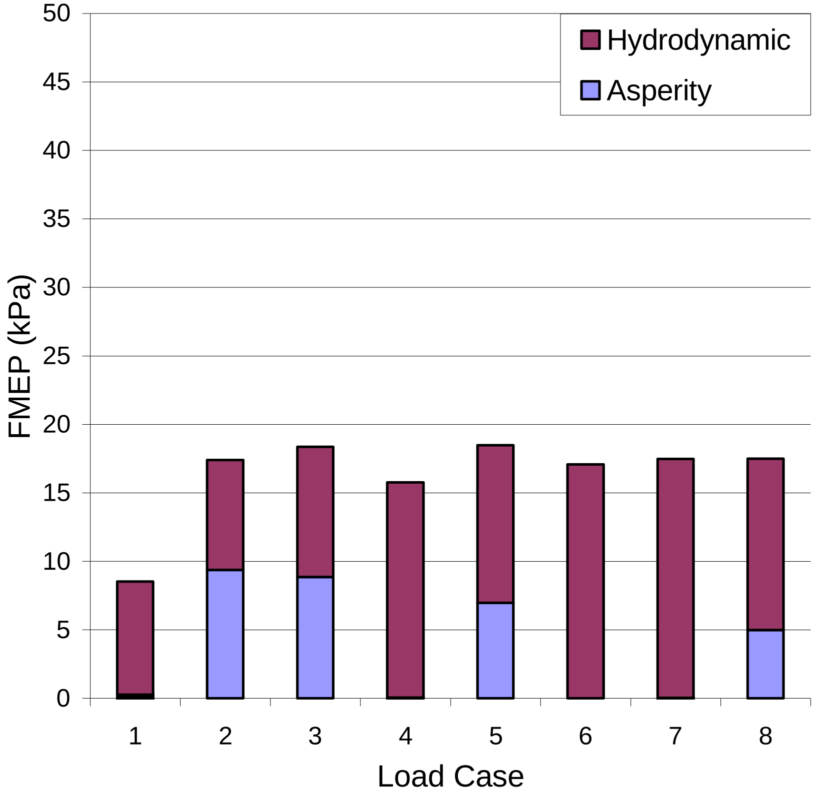


Figure 9: Piston Skirt Friction Contributions
(90% Reduction in Asperity Friction Coefficient)

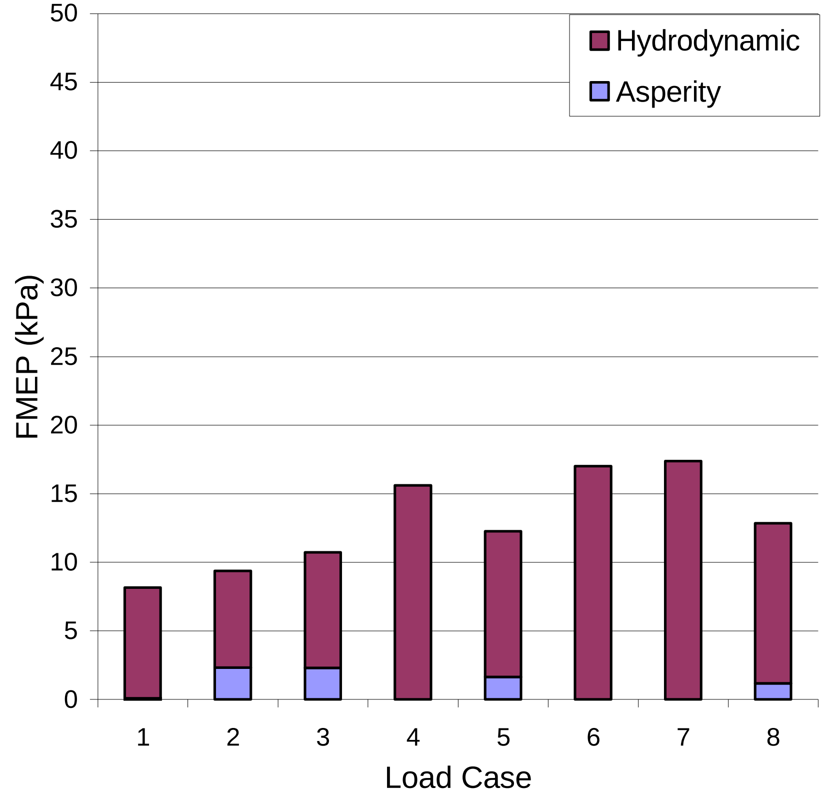


Figure 10: Ring Pack Friction Contributions (Uncoated)

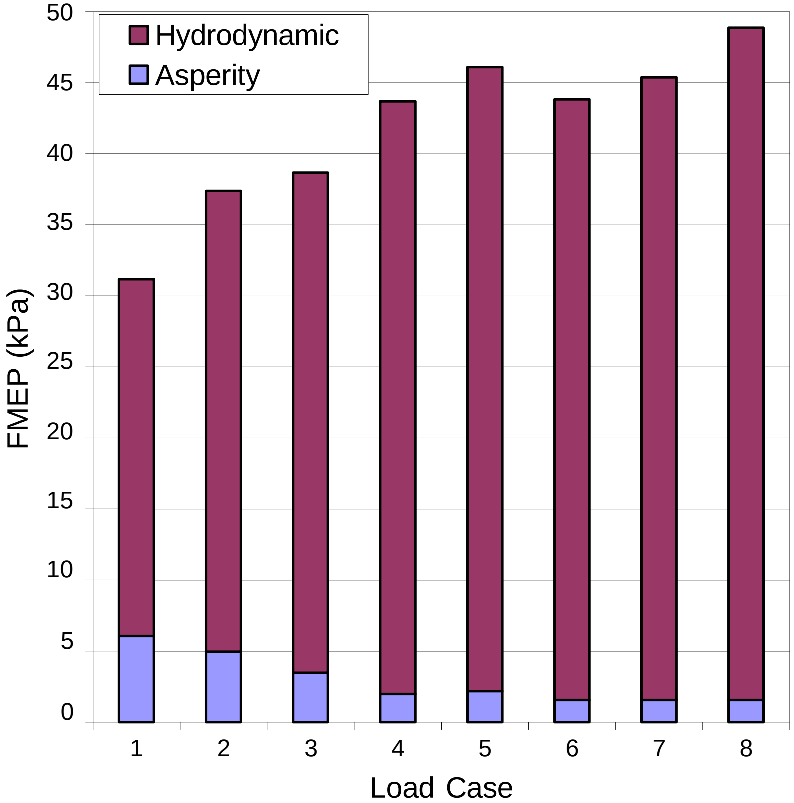


Figure 11: Ring Pack Friction Contributions (30% Reduction in Asperity Friction Coefficient)

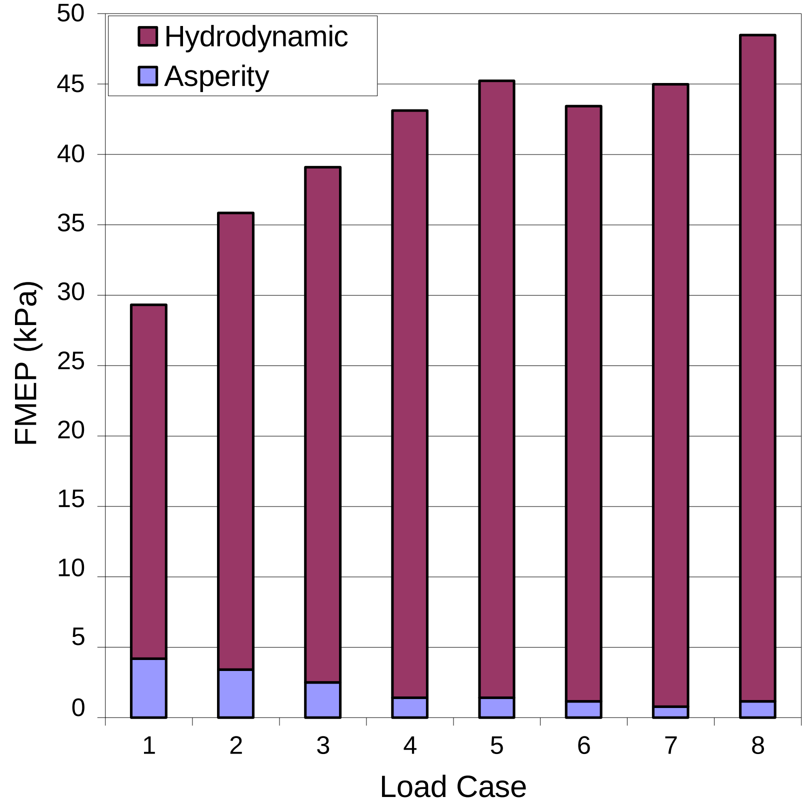


Figure 12: Ring Pack Friction Contributions (60% Reduction in Asperity Friction Coefficient)

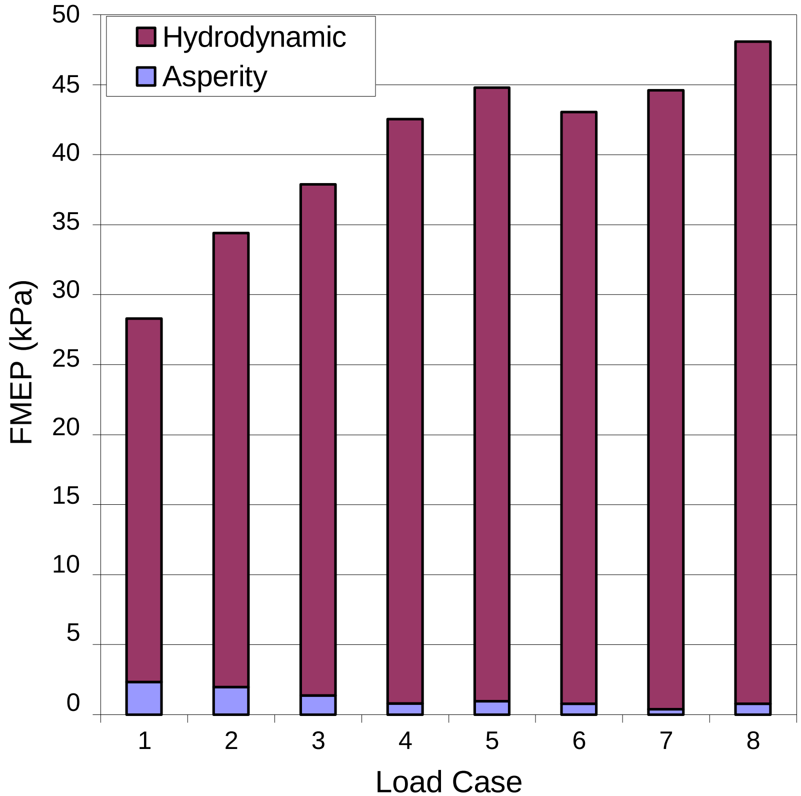
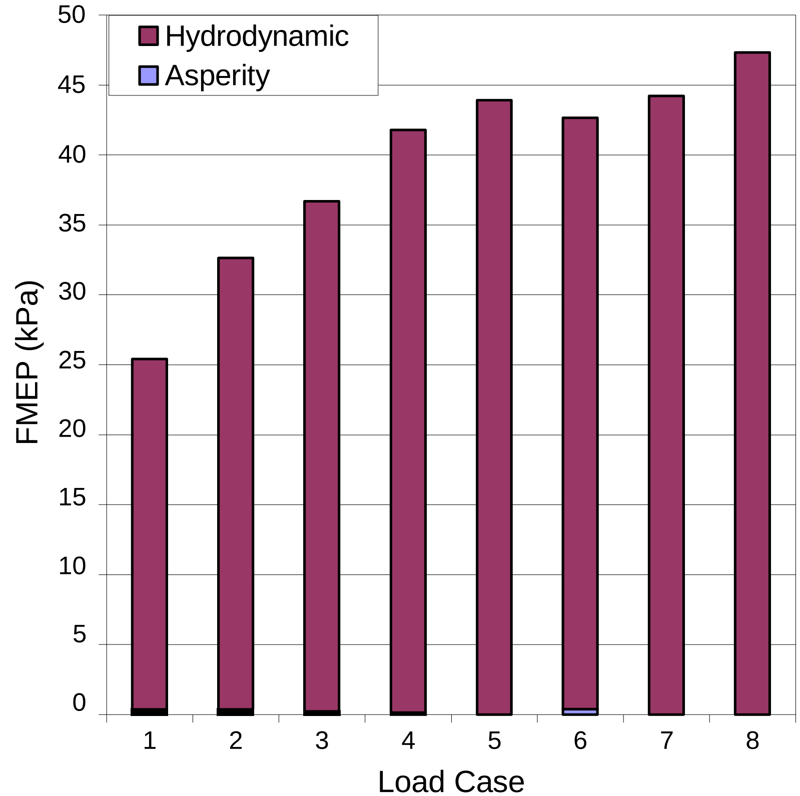


Figure 13: Ring Pack Friction Contributions (90% Reduction in Asperity Friction Coefficient)



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