

ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE
ENGINEERING AND TECHNOLOGY

**AN ENGINE TELEMETRY SYSTEM TO ASSESS PISTON ASSEMBLY
FRICTION AND SECOND LAND PRESSURES**

M.Sc. THESIS

Kaan ÖZDEMİR

Faculty of Mechanical Engineering

Automotive MSc Programme

JUNE 2013

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Thesis Advisor: Assoc. Prof. Dr. Özgen AKALIN

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

**PISTON GRUBU SÜRTÜNME KAYIPLARI VE SEGMANLAR ARASI
BASINÇLARIN DEĞERLENDİRİLMESİ İÇİN BİR MOTOR TELEMETRİ
SİSTEMİ**

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FOREWORD

This thesis is written as a completion for the Automotive MSc programme in ITU Faculty of Mechanical Engineering. This thesis is a part of the project “Power Cylinder Design for Optimized Lube Oil Consumption” supported by Ford Otosan A.S. This thesis covers an engine telemetry system study to perform measurements in the piston system. The project is done in accordance with Ford Otosan Product Development department. The objective is to improve the understanding of the mechanisms involved in piston system behaviour and provide the capability of performing measurements in the piston system during engine operation, in order to further optimize the piston system designs in internal combustion engines to reflect the requests from customers and government regulations.

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TABLE OF CONTENTS

	<u>Page</u>
FOREWORD	Error! Bookmark not defined.
TABLE OF CONTENTS	ix
ABBREVIATIONS	xi
LIST OF TABLES	xiii
LIST OF FIGURES	xv
SUMMARY	xvii
ÖZET	xxi
1. INTRODUCTION	1
1.1 Motivation	1
1.1.1 Engine friction	1
1.1.2 Piston sealing	2
1.1.3 Oil consumption	2
1.1.4 Blow-by	2
1.2 Literature Review	4
1.2.1 Simple methods for piston system friction measurement	4
1.2.2 Floating liner method	5
1.2.3 Instantaneous IMEP method	6
1.2.3.1 Cylinder pressure measurement	8
1.2.3.2 Pegging method	8
1.2.3.3 Grasshopper linkage for strain measurement	10
1.3 Objective	11
2. THEORY	13
2.1 Piston Telemetry	13
2.1.1 Near field data transmission	13
2.1.2 Far field data transmission	15
2.1.2.1 Signal modulation methods	17
2.1.2.2 Signal multiplexing	18
2.2 Friction Force Calculation	19
2.2.1 Piston system inertia force calculation	22
2.2.2 Connecting rod force calculation	24
2.2.3 Connecting rod inertia calculation	26
2.3 Wheatstone Bridge Configuration	26
3. EXPERIMENTAL METHOD	31
3.1 Telemetry Equipment	31
3.1.1 Package checks	38
3.2 Piston Instrumentation	38
3.3 Connecting Rod Instrumentation	42
3.3.1 Strain-gage placement	42
3.3.2 Instrumentations	44
3.4 Connection Between Piston and Rod	46

3.5 Engine Block Modifications and Instrumentations	53
3.5.1 Modifications for pegging sensor.....	53
3.5.2 Engine antenna placement.....	57
3.5.3 Block and ladderframe modifications for stator coil.....	59
3.6 Calibration	61
3.6.1 Strain-gage calibration	62
3.6.2 Pressure sensors calibration	66
3.6.2.1 Telemetry system lag	71
3.6.3 Thermocouple calibration	72
4. SIMULATED RESULTS	73
4.1 Inputs	74
4.1.1 Constants	74
4.1.2 Variables.....	75
4.2 Results	77
5. CONCLUSIONS AND RECOMMENDATIONS	85
REFERENCES	89
APPENDICES	91
CURRICULUM VITAE	99

ABBREVIATIONS

AM	: Amplitude Modulation
BDC	: Bottom Dead Centre
BMEP	: Brake Mean Effective Pressure
CAE	: Computer Aided Engineering
DOE	: Design of Experiments
FEA	: Finite Element Analysis
FFM	: Frequency Flicker Modulation
FM	: Frequency Modulation
FMEP	: Friction Mean Effective Pressure
IMEP	: Indicated Mean Effective Pressure
MEP	: Mean Effective Pressure
NVH	: Noise, Vibration and Harshness
PFP	: Peak Firing Pressure
PM	: Particulate Matter
TDC	: Top Dead Centre

LIST OF TABLES

	<u>Page</u>
Table 2.1: Sensitivity of Wheatstone bridge configurations to different scenarios..	29
Table 3.1: Accuracy of the NI DAQ device	67

LIST OF FIGURES

	<u>Page</u>
Figure 1.1 : Scheme for floating liner method [7].	6
Figure 1.2 : Free body diagram of the piston [11].	7
Figure 1.3 : Pegging sensor location and adaptor design [14].	9
Figure 1.4 : Grasshopper linkage mechanism [14].	10
Figure 2.1 : Circuit for near field data transmission.	14
Figure 2.2 : Scheme for piston telemetry system.	16
Figure 2.3 : Diagram for AM and FM.	17
Figure 2.4 : Diagram for FFM.	18
Figure 2.5 : Detailed scheme for piston telemetry system.	20
Figure 2.6 : Location of the sensors used for friction calculation.	21
Figure 2.7 : Simplified free body diagram of the piston.	22
Figure 2.8 : Dimensions involved in piston equation of motion.	23
Figure 2.9 : Simplified free body diagram of the connecting rod upper region.	25
Figure 2.10 : Wheatstone bridge configuration.	27
Figure 2.11 : Options for Wheatstone bridge configuration.	28
Figure 3.1 : Connecting rod with transmitter and rotor coil units.	33
Figure 3.2 : Stator coil unit mounted on the ladderframe.	33
Figure 3.3 : Receiver and inductive power generator units.	34
Figure 3.4 : Adapter box.	34
Figure 3.5 : Scheme of the telemetry equipment.	35
Figure 3.6 : CAD representation of the telemetry equipment.	36
Figure 3.7 : Location of the Adjustment Knobs on the Receiver Unit.	37
Figure 3.8 : CAD packaging check.	38
Figure 3.9 : Packaging check with endoscope.	38
Figure 3.10 : CAD representation of the sensor locations inside the piston.	39
Figure 3.11 : Kulite pressure sensor, the bridge is highlighted.	40
Figure 3.12 : Size of the Kulite pressure sensor.	40
Figure 3.13 : Placement of sensors inside the piston.	41
Figure 3.14 : Smoothing the surface inside the piston with sandpaper.	41
Figure 3.15 : Placement of sensor cables and terminal for cable connections.	41
Figure 3.16 : Covering up the cables using JB Weld.	42
Figure 3.17 : Possible strain-gage locations on the rod.	42
Figure 3.18 : Result of rod FEA and selected gage location.	43
Figure 3.19 : Instrumentation of strain gages and connections to the transmitter.	45
Figure 3.20 : Using adhesives to cover the cables.	45
Figure 3.21 : Placement of pressure sensor bridges.	46
Figure 3.22 : The condition of Bowden cable design after testing.	47
Figure 3.23 : Teflon tube routing study done at CAD.	47

Figure 3.24 : Setup for design #2.	48
Figure 3.25 : Design #2 CAD pictures with cross-section of the cylindrical holder.	48
Figure 3.26 : Design #3: Cross section of the holder and piston drilling.....	49
Figure 3.27 : CAD pictures and packaging checks for design #3.	50
Figure 3.28 : Design #3: Result of dynamometer testing.....	51
Figure 3.29 : Design #4: CAD Pictures.	52
Figure 3.30 : Design #4: Pictures after the dynamometer test.	53
Figure 3.31 : CAD picture of pegging adaptor system.	54
Figure 3.32 : CAD pictures of pegging adaptor installations in cylinder block.	55
Figure 3.33 : Picture of the adaptor after testing and the drill in the cylinder block.	55
Figure 3.34 : Pictures showing block drilling and the adaptor inside the block.	56
Figure 3.35 : CAD representation showing the block antenna requirements.	58
Figure 3.36 : Engine block surface after smoothing and antenna brackets.....	59
Figure 3.37 : Pictures showing antenna and resistor installation.	59
Figure 3.38 : Ladderframe modifications.....	60
Figure 3.39 : Small machining at block underside to take out stator coil cables.....	60
Figure 3.40 : Engine block modifications.	61
Figure 3.41 : CAD screens showing the adaptor design.	63
Figure 3.42 : CAD checks for gripping area in the piston.	64
Figure 3.43 : Pictures showing the setup placed in the rig.....	65
Figure 3.44 : Calibration curves for strain-gage.	66
Figure 3.45 : The circuit shown in Datatel manual.	68
Figure 3.46 : The circuit constructed in PSpice and resulting input voltage.	69
Figure 3.47 : Picture of the electrical circuit.	69
Figure 3.48 : Screenshot of LabView control panel.	70
Figure 3.49 : Screenshot of LabView block diagrams.	70
Figure 3.50 : Calibration curves for pressure sensors.	71
Figure 3.51 : Figure for telemetry system lag.	72
Figure 3.52 : Calibration curve for transmitter thermocouple.	72
Figure 4.1 : Connecting Rod Regions.	75
Figure 4.2 : Friction curve used for this study.	76
Figure 4.3 : Cylinder pressure curve used for this study.....	76
Figure 4.4 : Graphs for Piston Motion and Connecting Rod Angle.....	77
Figure 4.5 : Graph for Piston Inertia.	78
Figure 4.6 : Graph for Connecting Rod Inertia.	79
Figure 4.7 : Graph for comparing Piston Force and Strain-Gage Force.	80
Figure 4.8 : Regenerated Piston Assembly Friction Force.	80
Figure 4.9 : Percentage of loss of gas force.	81
Figure 4.10 : Comparison of losses due to inertia and friction.	82

AN ENGINE TELEMETRY SYSTEM TO ASSESS PISTON ASSEMBLY FRICTION AND SECOND LAND PRESSURES

SUMMARY

Among the various studies in automotive industry, improving fuel economy and emissions of vehicles lead the pact. The emission regulations are dictating tighter and tighter standards, whereas the customers are asking for better and better fuel economy. One of the biggest contributors to these parameters lies within the heart of the engine, the piston system.

Various fields of study are available for improving the piston system to get better engine parameters. This thesis focuses on piston frictional and sealing losses.

As with all frictional losses, by improving piston system friction, it's possible to improve engine features like fuel economy, performance and emissions. Reducing friction has other benefits like improving engine lifetime and reducing cooling requirement.

A responsibility of the piston system is to seal the combustion gasses and oil as much as possible. Most of the oil that escapes from the crankcase to the cylinder, passing through the piston, is burned in the cylinder, causing oil consumption. An engine with high oil consumption requires frequent servicing. Especially for heavy duty truck segment, the customers desire as longer service intervals as possible. Another effect of oil consumption is on the combustion itself, where burning oil causes a lower combustion quality and thus higher emissions.

Following the intake stroke, the air is compressed and prepared for combustion. The gas that escapes around the piston during the compression stroke leads to power loss and increases emissions. During combustion stroke, the gas loss causes less force to be acted on the piston, further reducing engine performance. These gasses that escape to the crankcase mix with the engine oil. The oil inside the gas is mostly separated, the remaining portion further increases oil consumption. Following the separation, the gas is mixed with the intake air and sent back to the engine; reducing combustion efficiency.

Because of the reasons listed above, it plays an important role to further understand the reasoning of these effects. In this context, it was decided to measure piston system parameters in an actual running engine. Ford 9L Diesel 380PS Euro5 engine was selected, mainly because of ease of access to the moving components and lower engine speeds.

The sealing property of the piston is closely linked with ring dynamics. It was decided to measure piston 2nd land pressure, the outer region on the piston, between top and 2nd rings. The gas pressure inside this region greatly affects ring dynamics and thus sealing ability of the piston.

For measuring piston system friction, it was planned to use Instantaneous IMEP method. This method uses force balance to calculate piston system friction. Cylinder pressure is measured to compute the net force acting on the piston. Some of this force is used to overcome friction and inertia of the components. The remaining portion is transferred to the connecting rod, which is also measured via strain-gages placed on the part. Inertia forces are calculated using engine speed and mass of the components. Using force balance, the only unknown in the equation, which is piston system friction, is calculated.

In order to obtain good accuracy for cylinder pressure measurement, pegging method was used. An absolute pressure sensor was placed near BDC to correct the reading of the relative cylinder pressure sensor.

For obtaining sensor data from the piston and connecting rod, it was planned to use a telemetry system. The system uses radio waves to transmit signal from the connecting rod to the block. Although being complicated, the system requires minimum modification on the engine, thus allowing for the measurements to be representative of the original engine. A transmitter is placed on the connecting rod, which collects sensor data and converts it into radio signal. This signal is picked up by the antennas in the block and sent to the receiver and data acquisition unit, where it is converted back into voltage. In order to supply electricity to the transmitter on the connecting rod, magnetic coil system was used, where the stator coil is placed inside cylinder block and the rotor coil is placed on the connecting rod big end.

Two strain-gages were placed on the connecting rod. 90-degree rosettes were used so it was possible to connect the sensors in Full Poisson Bridge formation, allowing for good temperature and bending compensation. The sensors were placed as close to the connecting rod small end as possible, to reduce the inertia effect of the region in between the sensor and piston pin. Instrumentation work to connect the strain-gages to the transmitter was done carefully using different types of epoxy resins.

The pressure sensors in the piston also needed to be connected to the transmitter on the connecting rod. For the cabling between the piston and connecting rod, a Teflon tube and metal holder parts were used. Care was taken to make sure that the Teflon tube was not stressed and under rapid movement. The design was tested in a trial engine and test was passed.

For pegging sensor, an adaptor was designed. Engine block was drilled accordingly. The drilling has to pass through the block water jacket, thus care was taken on the sealing of coolant. This design was also tested in a trial engine and test was passed.

The pressure and strain-gage sensors were calibrated using the complete telemetry equipment. For strain-gage calibration, connecting rod is placed in a test rig to apply various loads. According to the applied force, the voltage in the receiver was recorded to obtain the relation between receiver voltage and force on the connecting rod. For calibration of pressure sensors, a calibration circuit was prepared to send controlled voltage to the transmitter. The ratio between the input and output voltages were obtained. Using the calibration curves of the pressure sensors, the input voltage was converted to pressure, thereby obtaining the relation between pressure and output voltage.

For the calculation of friction force, a MATLAB program was written. The program is given 3 pressure sensor inputs to compute net force on the piston. This is subtracted by inertia forces using piston and connecting rod equations of motion,

component weights and engine speed. For better accuracy, the program was written to take the engine speed variations into account. In order to check that the program is working correctly, simulated friction results were used.

Last step in the project is to prepare the brackets for block antenna. The antenna must be close to the connecting rod transmitting antennas at all times. The bracket designs have been prepared.

Many other possibilities present themselves after these measurements are completed. The measurements can be used to validate CAE models to optimize design of piston system. Moreover the telemetry equipment can be reused for more measurements. For example Eddy-Current gap sensors can be used to measure oil film thickness, piston secondary motion or ring movements. Obtaining this capability is highly beneficial for the vehicle manufacturer.

PISTON GRUBU SÜRTÜNME KAYIPLARI VE SEGMANLAR ARASI BASINÇLARIN DEĞERLENDİRİLMESİ İÇİN BİR MOTOR TELEMETRİ SİSTEMİ

ÖZET

Taşıt üreticilerinin günümüzde en çok önem verdiği konular arasında araçların tükettiği yakıt miktarı ve salınan egzoz gazlarının düşürülmesi ilk sıralarda yer almaktadır. Bunun sebeplerini zamanla sıkılaştırılan egzoz emisyon regülasyonları ve artan yakıt maliyetleri karşısında müşteriden gelen isteklere dayandırmak mümkündür. Bir taşıtta bu parametreleri etkileyen en önemli parçalardan biri motor ve motorun kalbi olarak adlandırabileceğimiz piston sistemidir. Dolayısıyla piston sisteminde yapılacak iyileştirmeler taşıt üreticileri için büyük önem taşımaktadır.

Piston sisteminde yakıt tüketimi ve emisyonların düşürülmesi için yapılabilecek çalışmalar birbirinden farklı konularda olabilmektedir. Bu tezin kapsamında yapılan çalışma piston sisteminin yarattığı sürtünme ve özellikle segman sistemi ile sağladığı sızdırmazlık üzerinedir.

Sürtünme kayıpları her zaman en düşük seviyeye getirilmeye çalışılır. Bu sayede birçok parametre iyileştirilebilir. Enerji kaybı daha az olacağından yakıt tüketimi iyileştirilecek, aynı miktar yakıt ile daha fazla güç alınabilecektir. Benzer şekilde sürtünmeyi düşürmenin emisyonların da düşürülmesine katkısı vardır. Daha az yakıt kullanılacağı için karbon dioksit düşecektir. Ayrıca yanmayı daha verimli hale getirerek karbon monoksit, parçacık emisyonu gibi değerleri de bir miktar düşürmek mümkündür. Son olarak sürtünme azaltılarak sürtünmenin parçalara olan aşınma etkisini de azaltıp motor ömrünü artırmak mümkündür.

Piston sisteminin önemli görevlerinden birisi de sızdırmazlık sağlamaktır. Piston sistemi, silindir içerisindeki havanın karter bölgesine geçmesini olabildiğince engellemenin yanında, karter bölgesinden gelen yağın da olabildiğince silindir içerisine çıkmasını engellemekle yükümlüdür.

Karter bölgesindeki yağın silindir içerisine kaçması sonucunda yağ tüketilmektedir. Fazla yağ tüketen bir motorun sık sık servis edilmesi gerekmektedir, bu da müşteriye maliyet olarak yansımaktadır. Özellikle kamyon segmentinde müşterinin beklentisi olabildiğince uzun servis aralığıdır. Ayrıca yağın silindir içerisinde yanması, yanma verimini olumsuz etkileyip emisyonları artırmaktadır. Bu şekilde yanma ortamında çalışan piston ve segmanlar olumsuz etkilenebilmekte, parçalar üzerinde çok fazla karbon birikmesinden ötürü motor arızaları oluşabilmektedir.

Emme döngüsü sırasında içeriye alınan hava, sıkıştırma döngüsü sırasında yüksek basınçlara ulaşmaktadır. Bu sırada piston sistemi bu havayı olabildiğince silindir içerisinde tutmakla yükümlüdür. Silindirden kaçan gazlar içerideki hava miktarını azaltacak, yanma verimini olumsuz etkileyerek güç düşüşüne ve emisyonların artmasına sebep olacaktır. Sıkıştırma döngüsünü takiben yanma döngüsü sırasında maksimum silindir basınçlarına ulaşmaktadır. Bu sırada silindirden kaçan gazlar ise

piston üzerine etkileyen kuvvetin azalmasına yol açmakta, motorun gücünün düşmesine sebep olmaktadır.

Silindirden kaçan bu gazlar, motorun başka bölgelerinden kaçan gazlar ile birlikte krank boşluğunda birirmektedir. Bu bölgede şiddetli yağ hareketleri mevcuttur. Krank yataklarından sızan yağlar, karterdeki yağın çalkalanması, piston soğutma jetinden püskürtülen yağ ve bu yağların krank ve biyel gibi hareketli parçalar ile çarpışması sonucunda bu bölgede biriken gazlar yağ ile karışmaktadır. Öncelikle bu karışım yağdan arındırılmalıdır. Süzme işlemi sonucunda her zaman bir miktar yağ kalmakta ve yağ tüketimine yol açmaktadır. Yağdan olabildiğince arınmış bile olsa, içerisinde yanma birleşenleri olduğundan dolayı bu gazlar atmosfere salınamazlar; emisyon standartları gereği egzoz sisteminden geçirilmek zorundadır, fakat egzoz sistemindeki yüksek basınç dolayısıyla bu gazlar egzoz gazları ile birleştirilemezler. Bu sebepten ötürü gazlar motorun emmesine verilerek geri dönüştürülmektedir. Bu da motordaki parçaları etkilediği gibi yanma üzerine de bilinmeyen bir etki getirerek verim düşüşüne yol açmaktadır.

Görülebildiği gibi piston sisteminin motor üzerinde yol açtığı etkiler büyüktür. Bu yüzden piston sisteminin yarattığı sürtünme ve sızdırmazlık özelliğinin daha iyi anlaşılabilmesi fayda sağlamaktadır. Bunun gerçekleşebilmesi için bu tezin kapsamında gerçek bir motor üzerinde, çalışma koşullarında ölçüm alınması düşünülmüştür. Ford 9L Dizel 380PS Euro5 kamyon motoru bu çalışma için seçilmiştir. Büyük bir motor kullanıldığı için parçalara kolay erişilebilmiş, gerekli modifikasyonlar rahatça yapılabilmektedir. Ayrıca motor maksimum gücü ve torku düşük devirlerde verdiği için sistemin uzun süre çalışması ve gerekirse başka motorlarda kullanılabilmesi sağlanmıştır.

Piston sisteminin sızdırmazlığı, piston segmanları dinamiği ile yakından bağlantılıdır. Segmanların yuva içerisindeki hareketleri ve bu hareketlerin sebeplerini anlayarak sızdırmazlığı iyileştirmek mümkündür. Bunun için piston üzerinde, birinci ve ikinci segman arasındaki bölgeden basınç ölçülmesi düşünülmüştür. Birinci segmandan sızan gazlar bu bölgede birikip basınç artışına yol açmakta, ve her iki segmanın dinamiğini etkileyerek sızdırmazlıkta önemli rol oynamaktadır. Bu etki analizler ile öngörülmeye çalışılsa da tasarımlar üzerindeki en büyük etki geçmiş tecrübeler olmaktadır. Segman arasından yapılacak bu ölçüm bilgisayar destekli analizlerin doğrulanmasında kullanılıp tasarımın iyileştirilmesine olanak sağlayacaktır.

Piston sürtünmesinin ölçülebilmesi için literatür taraması yapılmış, “Instantaneous IMEP” isimli metot seçilmiştir. Bu metot kuvvet dengesi kullanılarak piston sisteminin sürtünmesinin ölçülmesi üzerinedir. Silindir içerisinde oluşan gaz basıncı piston üzerine kuvvet olarak etki etmektedir. Bu kuvvetin bir kısmı sürtünme kayıplarına ve eylemsizlik kuvvetlerine harcanmaktadır. Geriye kalan kuvvet piston biyel koluna aktarılmaktadır. “Instantaneous IMEP” yöntemi ile, piston üzerine etki eden kuvvet, silindir basıncı ölçülerek hesaplanmaktadır. Biyel koluna uygulanan kuvvet, parçanın üzerine yerleştirilen gerinim pulları ile ölçülmektedir. Eylemsizlik kuvvetleri, motorun dönüş hızı ve parçaların kütleleri kullanılarak hesaplanmaktadır. Geriye tek bilinmeyen olan sürtünme kuvveti hesaplanabilmektedir.

Silindir basıncının hassas olarak ölçülebilmesi için “pegging” isimli yöntem kullanılmıştır. Silindir içerisinden basınç ölçen sensörler mutlak basınç ölçememekte, göreceli basınç ölçebilmektedir. Bunun yanında ölçtükleri değer zaman içerisinde kayma göstermektedir. Ölçülen değer doğrulanması için

kullanılan en hassas yöntem “pegging” yöntemidir. Bu yöntemde, silindir içerisine, alt ölü noktaya yakın bir yerden ikinci bir basınç sensörü ile basınç okunmaktadır. Alt ölü noktaya yakınlık sebebiyle bu sensörün göreceği basınçlar çok yüksek değildir, bu sayede hassas ve mutlak basınç ölçen bir sensör kullanılabilir. Piston sensör üzerinden geçtikten sonra, iki sensörün aynı değer okuduğuna emin olunup, silindir basınç sensörünün doğrulaması yapılabilmektedir.

Biyel kolundaki gerinim ve pistondaki basınç bilgilerinin alınabilmesi için telemetri sisteminin kullanılması düşünülmüştür. Bu sistem, piston ve biyel gibi hareketli parçalardan veri toplamak için kullanılan en iyi sistemdir. Biyel kolundaki elektronik aparat ile sensörlerden okunan veri radyo dalgasına çevrilip motor bloğu içerisinde yayınlamaktadır. Bu sinyal bloktaki antenler aracılığı ile toplanıp işlenmek üzere motor dışında bulunan ekipmanlara gönderilmektedir. Motor içerisinde ayrıca biyel kolundaki elektronikleri beslemek için manyetik alan üreten bobinler bulunmaktadır.

Bu sistem karmaşık olmasının karşılığında motor üzerinde en az değişiklik gerektiren sistemdir. Kullanılan parçaların motor birleşenlerinin çalışmasına etkisi çok azdır, böylece motorun orjinal halini temsil etmesi ve alınan verilerin gerçekçi olması sağlanmıştır.

Telemetri sistemi Datatel firmasından temin edilmiştir. Firma biyel kolu üzerine verici elektroniğini ve manyetik indükleme için bobinleri yerleştirmiş, blok üzerine yerleştirilecek olan karşı bobinleri braketleri ile birlikte temin etmiş, blok sonrasında verinin toplanıp işlenmesi için gerekli aparatları göndermiştir. Sistem bu şekilde alındıktan sonra piston ve biyel üzerine sensörler yerleştirilmiş, verici devreye bağlantıları için enstrümantasyonu yapılmıştır.

Biyel üzerine gerinim pullarının monte edileceği bölge için öncelikle gerinimin olabildiğince sabit kaldığı düz bir yüzey aranmıştır. Bu sayede, iki taraflı yerleştirilecek olan pullar arasındaki ufak mesafe farklılığının okunan değerde büyük hatalara sebep olmaması sağlanmıştır. Biyel üzerindeki gerinim dağılımı sonlu elemanlar analizi ile bulunup uygun yer seçilmiştir.

Pullar olabildiğince biyel küçük ucuna, pistonu pim ile bağlantı yapılan bölgeye yakın değildir; çünkü kuvvet hesabı sırasında biyelde gerinim pulları ile küçük uç arasında kalan bölge de eylemsizlik kuvveti yaratacak, gerinim pullarının okuduğu değeri etkileyecektir. Seçilen bölge küçük uç ile biyel gövdesinin radyusunun bittiği yerdendir. Biyelin küçük uç ile büyük uç arasındaki gövde kısmı “I” profil şeklindedir. Bu profilin alt ve üst yüzeylerine gerinim pulları yerleştirilmiştir. 90 derecelik pullar kullanılmış, böylece toplam pul sayısı 4’e tamamlanarak tam köprü bağlantısı sağlanabilmektedir. Bu bağlantı ile sıcaklığın ve biyelin bükülmesinin etkisi minimuma indirilmiştir.

Gerinim pulu köprüsünün oluşturulması ve biyel üzerindeki vericiye bağlanması için enstrümantasyon yapılmıştır. Kullanılan kabloların motor çalışma şartlarında biyel üzerinde kalması için farklı epoksi reçineler kullanılmıştır.

Piston segmanları arası basınç ölçümü için piston alt yüzeyinden yuva açılmıştır. Sensörler piston dış yüzeyine bakacak şekilde monte edilip kablolar yuva içerisinden pistonun altına getirilmiştir. Gerinim pullarında olduğu gibi bu sensörlerin de biyeldeki verici ile bağlantısının sağlanması gerekmektedir. Bu ara bağlantı için teflon kablo, metal tutucu parçalar ve setskurlar kullanılarak bir kablo taşıyıcı tasarımı oluşturulmuştur. Teflon kablo veri taşıyan kablolar için taşıyıcı görevi görmektedir. İki taraftan piston ve biyeye metal tutucu parçalar ve setskurlar ile

tutturulmuştur. Setskurlar ayrıca teflon t p n metal aparattan kaymasını da engellemektedir. Teflon t p n piston ile biyel arasındaki rotası se ilirken t p n hareketinin minimuma indirilmesi d   n lm   t r. T p biyel k   k ucunun yan tarafından  ıkmakta, b t n k   k ucu dola ıp piston p m ba lı ına ba lanmaktadır. Bu tasarım deneme motorunda denenip ba arılı oldu u g r lm   t r. Basın  sens rlerinin kendileri ile birlikte gelen k pr  devreleri biyel  zerine monte edilmi tir. Biyeldeki vericiyle ba lantı sa lamak i in kullanılan kablolar, gerinim pullarında oldu u gibi epoksi re ineler ile tutturulmu tur.

“Pegging” i lemi i in, motor blo una yukarıdan 13 cm mesafede yandan delik delinmesi uygun g r lm   t r. Aparat tasarlanmış ve blok  zerinde uygun  ekilde yuva a ılmıştır. Aparat iki par a olarak d   n lm   t r. Dı  par a sens r ile birlikte blok i erisine yerle tirildikten sonra i erideki par a ile sens r n konumu sabitlenmi tir. Aparat blok su ceket i erisinden ge ip silindire ula tı ından dolayı suyun sızdırmazlı ı  nem ta ımaktadır. Aparat  zerine uygun  ekilde o-ring monte edilip sızdırmazlık macunu ile sızdırmazlık sa lanmışır. Tasarım deneme motorunda test edilmi  ve sorun  ıkmadı ı g r lm   t r.

Telemetri sisteminde kullanılan par aların tasarımı sırasında motorun di er par alarına  arpmaması g z  n nde bulundurulmu tur. Biyel  zerindeki verici ve enstr mantasyonun piston so utma jeti, piston alt y zeyi, krank kolları ve kar ı a ırlıkları ile temas etmedi i do rulanmışır. Aynı  ekilde pistondaki enstr mantasyonun biyel ve krank ile temas etmedi i do rulanmışır. Blok  zerindeki manyetik ind kleyicinin de krank veya ya   ekme borusu ile temas etmedi i kontrol edilmi tir.

Telemetri sisteminin do ası gere i sens r kalibrasyonunun hassas  ekilde yapılması  nem ta ımaktadır. Sens rden okunan basın  veya gerinim sinyali rasyo dalgasına d n  t r lmek gibi bir ok d n   mden ge irildi i i in, sistemin sonundaki  niteden okunan voltaj sinyali ile sens r sinyalleri arasındaki ba lantıyı  nceden tahmin etmek m mk n de ildir. Okunan sens r de eri ile  ıkı  voltajı arasındaki ili kinin do ru belirlenmesi i in kalibrasyonun hassas  ekilde ger ekle tirilmesi gerekmektedir.

Gerinim pullarının kalibrasyonu i in piston biyel kolu komple olarak,  zerinde verici gibi t m ekipmanlar montajlı halde iken kalibre edilmi tir. Par a, kontroll  kuvvet uygulayan bir test ekipmanına ba lanmışır. Bu ba lantı i in gerekli aparatlar tasarlanıp  retilmi tir. Ama  biyel  zerine do rudan kuvvet vererek, bu kuvvetin yarattı ı  ıkı  voltajını okumaktır. B ylece aradaki b t n a amalar, gerinim pullarının aktif hale gelmesi, vericiye sinyal g nderilmesi, vericinin sinyali radyo dalgasına  evirip alıcıya g ndermesi ve alıcının sinyali i leyip voltaja  evirmesinin do ru  alı tı ı kontrol edilmi tir. Bu aradaki i lemlerde ne kadar sinyal  retildi i  nemsenmemi tir.

Basın  sens rlerinin kalibrasyonu i in ise telemetri ekipmanının kitap ı ında belirtildi i gibi kalibrasyon devresi olu turulmu tur. Devre verici ile ba lanmış ve  ıkı  voltajı okunmu tur. Devreye bir sinyal i leme kartı ile temsili basın  sens r  sinyali g nderilmi tir. Giri  sinyali ile  ıkı  sinyali arasındaki ili ki olu turulmu tur. Her basın  sens r n n kendi kalibrasyon e risi verildi inden, sisteme verilen sinyallerin hangi basın  de erlerine denk geldi i bilinmektedir. B ylece basın  de erine kar ılık  ıkı  voltajının de eri bulunmu tur.

S rt nme de erinin hesaplanması i in 5 tane sens r bilgisi kullanılacaktır. Silindir i indeki basın , karterdeki basın tan  ıkartılarak net basın  bulunacaktır. Bu sırada

pegging sensörü kullanılarak silindir içi basınç sensörün her döngüde doğrulaması yapılacaktır. Bu 3 sensör sonucunda pistona gelen net kuvvet hesaplanacaktır. Biyeldeki gerinim pullarının okuduğu değere bakılarak, pistona gelen net kuvvetten biyeldeki gerinim kuvveti çıkartılacaktır. Son okunacak olan değer ise motor hızı olacaktır. Volan dişlisi üzerindeki sensör vasıtasıyla anlık olarak motor hızı değeri okunacaktır. Bu değer ile, piston sisteminin kütlesi ve piston hareket denklemleri kullanılarak piston sisteminin eylemsizliği hesaplanacaktır. Benzer şekilde biyelin gerinim pulu üzerinde kalan bölgesi için de eylemsizlik hesabı yapılacaktır. Bu hesaplar için MATLAB programı yazılmıştır. Piston ve biyelin hareket denklemleri çıkartılırken motordaki ufak hız değişimleri olacağı göz önünde bulundurulmuştur.

Motor üzerindeki işlemler henüz bitmediğinden MATLAB programının çalıştığının kontrol edilebilmesi için bilgisayar analizlerinden elde edilen sürtünme değeri programa girilmiştir.

Projede son aşama olarak motor bloğuna antenler tutturulacaktır. Sensör verisinin doğru alınabilmesi için, bloktaki antenler biyeldeki verici kablolarla her daim yakın mesafede olmalıdır. Bunun için gerekli braket tasarımları hazırlanmıştır.

Yapılacak bu çalışma ile birçok amaç öngörülmüştür. Alınan piston sürtünme ve segman arası basınç verileri ile bilgisayar modelleri doğrulanabilecek, tasarımlar iyileştirilebilecektir. Ayrıca aynı telemetri sistemi başka ölçümler için de kullanılabilir. Örneğin “Eddy-current” mesafe sensörleri kullanılarak yağ filmi kalınlığı, piston ikincil hareketi veya segman konumları belirlenebilecektir. Bu tür bir kabiliyet kazanmak taşıt üreticilerine avantaj sağlamaktadır.

1. INTRODUCTION

1.1 Motivation

Situated in the heart of the engine, the piston is one of the most important parts in an internal combustion engine. Together with being highly stressed and vulnerable, it is also one of the biggest contributors to important engine features such as performance, fuel economy and emissions.

The ever growing demand of improving these engine parameters provides the fuel for the studies on the piston. The customer continues to demand improved fuel economy and performance, whereas the regulations continue to dictate tighter emission standards. Various fields of research are available for such studies on the piston; this thesis focuses on the losses caused by the piston through friction and sealing.

1.1.1 Engine friction

Piston system is known to be the largest contributor to friction losses. Early studies done in 1970s suggest that piston system is responsible of up to 75% of total engine friction [1]. As the engine development progressed, in 2000s this value has been dropped to 55% [2, 3]; whereas recent studies suggest that a modern piston system contributes to around 45% of total engine friction [4].

This friction is caused by the relative motion of different parts. Together with the rings, the piston moves inside the cylinder. The running faces of the rings and the outer surface of the piston (especially through piston skirt) are in contact with the cylinder, which is the main source of friction in the piston system. The remaining portion occurs due to the relative movement of the connecting rod with respect to the piston. These two components are held together by the piston pin, which is in contact with the piston pin bores and connecting rod small end, causing friction.

The energy loss caused by friction leads to deterioration of many engine parameters. Reducing friction has affects like increasing engine performance and fuel economy;

together with improving emissions. CO₂ emissions would be reduced through burning less fuel, together with other emissions like CO and PM through a more efficient combustion. Moreover, friction has negative effects on the components themselves where high friction generally leads to high wear on piston and cylinder. Final effect is the temperature increase on the components and engine, which imposes increasing demand on the cooling system.

1.1.2 Piston sealing

Through the rings positioned around the piston inside the ring grooves, the piston system aims to seal, as much as possible, both the combustion gases and engine oil. Combustion gasses located on top of the piston must be restricted to reach the crankcase region below the cylinders; whereas the engine oil must be restricted to get in contact with combustion. The gasses that leak from the cylinder cause increase in “blow-by”, whereas some of the oil that reach the cylinders burn and cause oil consumption.

1.1.3 Oil consumption

The main effect of oil consumption is on service intervals. Under worst case conditions, at the end of this interval, there still needs to be a minimum amount of oil left in the oil sump for the engine to operate normally. Especially in heavy duty truck segment, the vehicle is required to operate as much as possible without having the need to service. Decreasing oil consumption is one way to increase the service interval.

Another effect of oil consumption is on the combustion itself. Under certain conditions, oil consumed inside the cylinder may increase engine performance, but this is not a desired effect since it cannot be controlled. On the other hand, this consumed oil decreases combustion quality, therefore negatively effecting emissions. The resulting by-products of oil combustion are known to negatively affect engine components, for example through carbon build-up on piston and rings, which may lead to catastrophic engine failure through ring sticking and cylinder scuffing.

1.1.4 Blow-by

As the intake air is compressed during compression stroke, the pressure inside the cylinder starts to increase. This increase in pressure is essential for formation of

combustion and combustion efficiency. The gases that escape around the piston during this cycle decrease this pressure. Moreover, since some of the intake air is lost, the volumetric efficiency is reduced. These badly affect engine performance and fuel economy.

During combustion and expansion, the gasses continue to leak from the cylinder, causing decrease on the amount of force applied to the piston, reducing engine torque and power.

The gasses that leak from the cylinder join with other gasses sent from other components in the engine, forming the blow-by gasses. These gasses join in the crankcase, the region around the crankshaft and above the oil sump. This region is dominated by violent oil movements; where the oil that is sprayed around crankshaft bearings and squirted from piston cooling jets, gets splashed randomly by the movement of crankshaft, connecting rods and pistons. The blow-by gasses mix with the engine oil. This mixture cannot simply be released to the atmosphere; firstly due to high oil content which would cause high oil consumption. Oil must be separated from the mixture. Even after this separation, the mixture, now containing mostly combustion gasses, must pass through exhaust after-treatment system to meet with emission regulations. This gas cannot be joined with the exhaust system because of its low pressure, so it must be recycled back into the engine, joined with the intake and entered into the cylinders. The crankcase ventilation system is responsible for these operations. The overall effects of blow-by are increase in oil consumption, because oil can never be separated completely; and decrease in combustion efficiency, because of uncontrolled gas content and amount.

The many negative effects described previously give reason to study the mechanisms, increase understanding and the find out possibilities to reduce these effects. This thesis focuses on performing a measurement in an actual internal combustion engine, during normal engine operation. It was decided to perform two measurements: The friction caused by the piston system and the pressure between the first two rings of the piston. The friction measurement will increase the understanding of the mechanisms of friction, whereas the pressure measurement will increase understanding of piston sealing properties through the piston ring dynamics. These measurements can be used to optimize piston system design and reduce the negative effects like blow-by and oil consumption. The measurements can be used as

a comparative basis, where different designs can be tested in the same engine and their effects are studied. In addition, the measurements can be used to increase accuracy of CAE analysis where a CAE model can be validated and applied on other engines and other designs.

After obtaining the capability of making such measurements on the piston system, the range of measurements can be extended to cover various fields. Gas pressure can be measured from different locations; like back of ring grooves or 3rd land, the region between 2nd and 3rd piston rings; aiding in understanding ring dynamics. Eddy current gap sensors can be used to measure distance of the piston to the cylinder bore, computing oil film thickness and piston secondary motion; which can help improve piston friction and piston NVH. Gap sensors can also be used to measure ring locations inside the groove, aiding further to understand ring dynamics. Stain-gages can be placed on the piston to measure strain at critical locations, for example the piston skirt in order to optimize skirt design and reduce friction.

1.2 Literature Review

1.2.1 Simple methods for piston system friction measurement

Some of the methods for piston system friction measurement consist of measurements done without engine firing operation. One of the simplest methods consists of engine dynamometer measurements, where the engine is run at motoring condition and the torque required to keep the engine at a certain speed is measured. Since there's no combustion, this measurement can be done with only the piston and cranktrain system installed in the engine. In this condition, the torque required to turn the engine equals the piston and cranktrain system friction [5].

The biggest disadvantage of such methods is that measurement cannot be performed under firing condition where piston system friction is increased greatly due to the combustion forces that act on the piston.

Another simple method to measure friction is realized using Mean Effective Pressure. The engine is placed in a dynamometer and run normally. Cylinder pressure transducers are used to calculate IMEP. Brake power is measured from the dynamometer and is used to calculate BMEP. The difference between IMEP and BMEP is equal to FMEP, which is an indication of total engine losses and contains

piston system friction [5]. With this method, it's not possible to isolate the piston system friction so this measurement can only be used comparatively; for example in a DOE study where different piston system designs are tested and their effect on FMEP is observed. While doing so, the engine needs to be stripped down and rebuilt with new parts; such actions are known to cause change in engine parameters. Moreover, effects like wear are continuously causing changes on the measurements. Finally isolating the differences in piston system friction using the much bigger FMEP values require very precise measurement. Because of these reasons, this is not an accurate method to measure piston system friction.

1.2.2 Floating liner method

The earliest studies to measure piston system friction in firing condition use a method called floating liner [6]. In this method, cylinder liner is suspended in the cylinder block, allowing for small movement along the direction of piston travel. This movement is caused by the force applied on the liner by the piston, which is equivalent to the friction force itself. Continuous measurement of this force, via the position of the cylinder liner or via load sensors, is possible; thereby making this method the first example for instantaneous piston system friction measurement. The scheme for this method is given in Figure 1.1.

Later studies improved this technique to get better accuracy [7, 8]. Liner stiffness has been improved to increase the natural frequency of the liner [7], which would otherwise cause interaction with the friction measurement. Additional radial supports have been used to reduce bore deformation as much as possible [7], which would otherwise create unrealistic working conditions for the piston assembly. This technique was later used in studies such as to increase the understanding about the break-in phenomenon [9] and to isolate the role of the oil control ring in piston assembly friction [10].

There are two main disadvantages of this method. Firstly, extensive modifications are required on the engine, mainly due to making the cylinder liner moveable. Secondly, despite the extensive efforts, it's not possible to simulate the exact engine operating conditions. The movement of the cylinder liner causes the relative motion between the piston and cylinder to change, especially at TDC and BDC. This causes friction to change. In addition, cylinder bore deformation does not represent the

original engine, and care must be taken to eliminate the forces caused by the movement and vibration of the cylinder liner itself. These effects cause additional noise on the measurements and thus make this method inaccurate for measuring piston system friction [11].

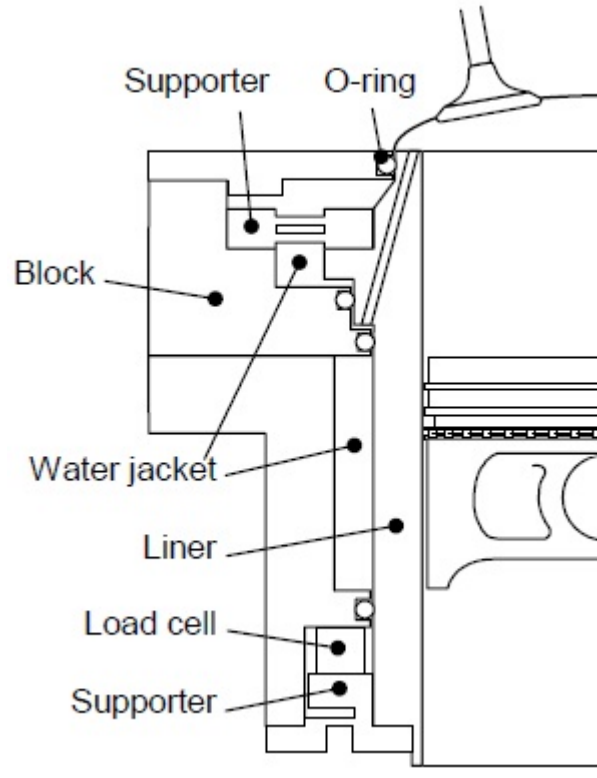


Figure 1.1 : Scheme for floating liner method [7].

1.2.3 Instantaneous IMEP method

One of the first examples of accurate friction measurement in an internal combustion engine was performed by Uras and Patterson [11]. The friction measurement method is referred to as Instantaneous IMEP method. This method uses force balance to calculate piston system friction.

The gas pressure inside the cylinder applies force onto the piston. Some of this force is used to overcome friction and inertia of the piston system. The remaining force is applied on the connecting rod. Figure 1.2 shows the free body diagram of the piston [11].

The force applied on the piston is measured via accurate in-cylinder pressure sensors. The force applied on the connecting rod is measured via the strain-gages placed on the shank region. The inertia forces are calculated from engine speed, which is

continuously measured, and the weight of the components. Using these values, the only remaining unknown in the equation, which is the piston assembly friction, is calculated.

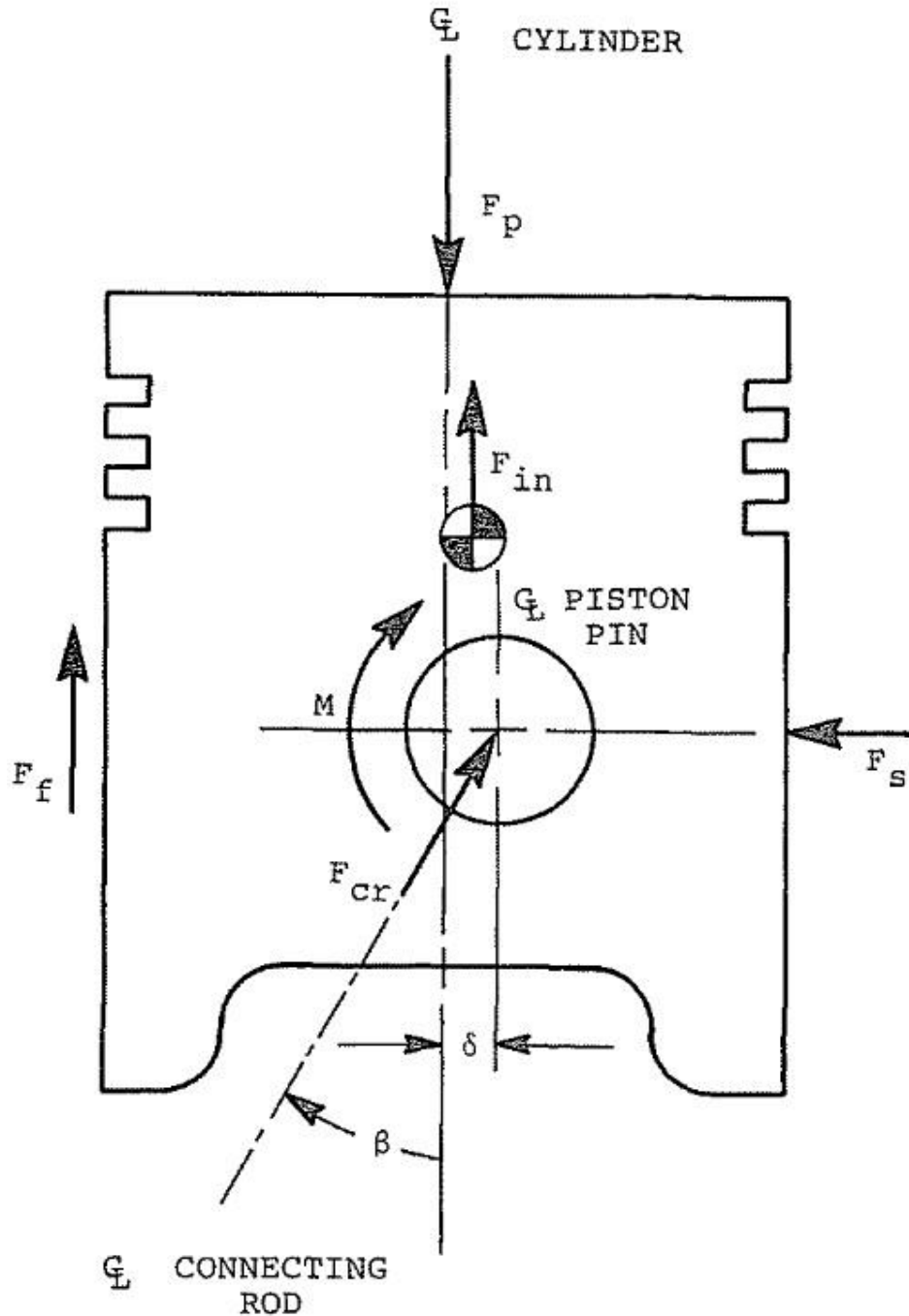


Figure 1.2 : Free body diagram of the piston [11].

Acquiring the ability to measure piston assembly friction allows for future studies to be conducted in order to increase the understanding of the mechanisms that lead to frictional losses. Some of the later studies used this approach to investigate the

effects of different lubricants, piston ring and engine variables on instantaneous piston and ring assembly friction [12, 13].

1.2.3.1 Cylinder pressure measurement

It is important to perform these measurements as accurately as possible. Due to the nature of this method, big numbers are subtracted from each other to find a relatively small value, friction. Small errors in the measurement of cylinder pressure or connecting rod strain translate to a much bigger error in friction. In order to prevent such big errors, water cooled cylinder pressure transducers were used [11]. Such sensors aim to reduce the effects of combustion heat and pressure on the measurement via water cooling. The pressure of the cooling water must be kept constant to prevent interaction with the sensor reading. Despite the efforts, the sensor continues to show non-linear tendencies and the values measured tend to drift over time.

Cylinder pressure measurement is widely applied in the industry during the development of an engine. The goal is usually to control cylinder peak firing pressure (PFP), therefore the measurement accuracy is not as big a concern as in friction measurement. The cylinder transducers used are relatively simpler and does not contain water cooling. These sensors can only measure relative pressure, which constantly needs to be converted to absolute pressure. This is usually done via an additional absolute pressure transducer placed in the exhaust system, near the exhaust valve. It's assumed that during the exhaust cycle, the two locations have the same pressure. Using this info, the value read by the cylinder pressure sensor is corrected by the absolute pressure sensor. Such absolute pressure sensors have lower pressure capability, therefore they cannot be placed directly in the cylinder. The advantage of this system is that it's easy to apply. Most of the time, pressure transducers already exist in the exhaust system to check the pressures, therefore no additional work is required. The main error in this approach is the assumption of same pressure in the cylinder and in the exhaust. Because of gas flow from cylinder to the exhaust system, the two pressures are never exactly the same.

1.2.3.2 Pegging method

Later studies done on Instantaneous IMEP method used cylinder pegging to further improve the pressure sensor reading [14]. They have moved the absolute pressure

sensor from the exhaust system into the cylinder to a lower position in the cylinder block. This location is usually selected towards BDC, to make sure that sensor's reading range is not exceeded. Once the piston passes the point where the absolute sensor is placed, the correction for the cylinder pressure transducer can be made. This is the most accurate method used to date and thus was selected for the system in this study. Figure 1.3 shows the location of the sensor in the cylinder block and the adaptors for the sensor [14].

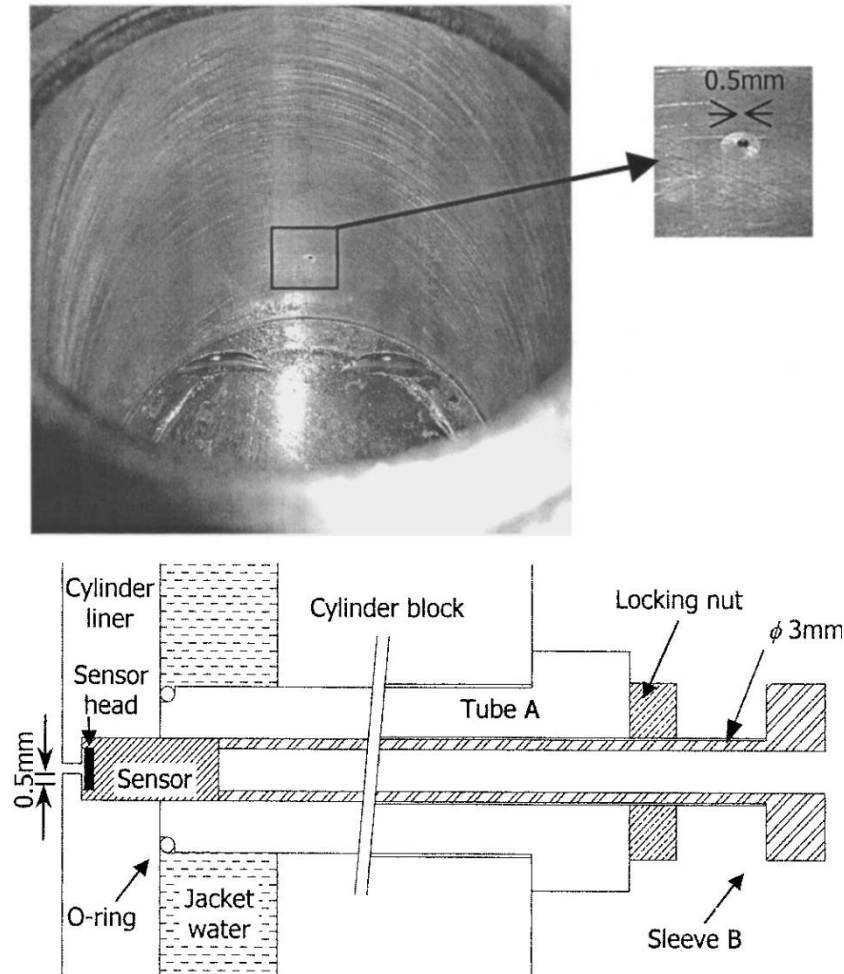


Figure 1.3 : Pegging sensor location and adaptor design [14].

Other studies disagree with this approach, indicating that drilling a hole through the side of the block is difficult from machining and sealing perspective [15]. The alternative solution was to perform pegging at the inlet port, very close to the intake valve seat. Such a placement allows for the pegging operation to be performed during the intake stroke. With this method, same measurement accuracy with the pegging operation from the cylinder was able to be obtained [15]. However, the engine used in this study was an atmospheric petrol racing engine. Such engines have

aggressive valve timing where the valve is kept open for a small portion of the compression stroke (after the piston passes BDC). This is called retarded valve timing. The principle is to use the inertia of the intake air to allow more air to fill the cylinder. Due to this valve timing strategy, such engines show higher pressures inside the intake. On the other hand, in a heavy duty diesel engine, intake valve is closed almost immediately after BDC, therefore the pressure in the intake is kept small. It was decided that pegging operation with such small pressures will not yield accurate results, therefore it was decided to use cylinder pegging technique.

1.2.3.3 Grasshopper linkage for strain measurement

Another important measurement is the strain measurement from the connecting rod. Since it's a moving part, it's difficult to obtain the sensor information. The studies explained earlier use a method called grasshopper linkage [11, 14]. This method consists of a two bar linkage between the connecting rod and the block. This linkage is used to carry the cables required for sensor measurement. Figure 1.4 shows the picture from the linkage.

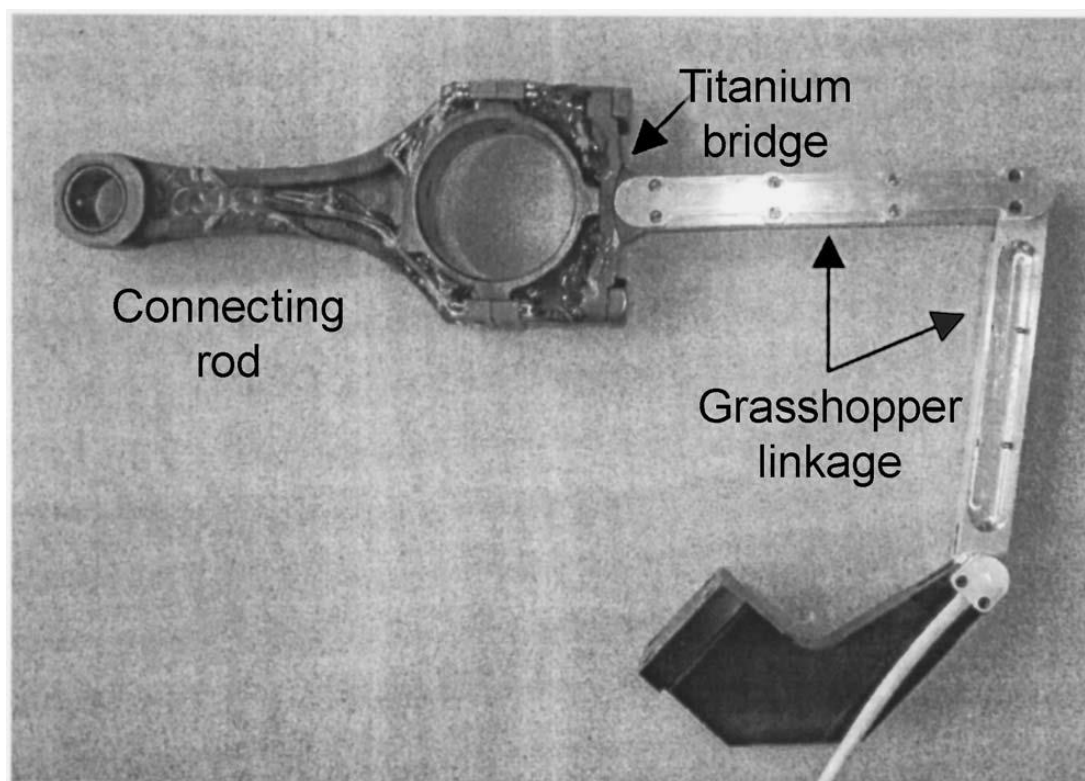


Figure 1.4 : Grasshopper linkage mechanism [14].

This linkage provides direct connection between the strain-gages and data acquisition equipment. Despite its relatively simple construction, it requires substantial

modifications inside the cylinder block. Moreover, the linkage interacts with the movement of the connecting rod, applying additional forces to the system, so the modified engine does not fully represent the original engine. Lastly, the linkage is not robust enough for high speed / long duration operations.

1.3 Objective

The objective of this study is to measure piston system friction and piston 2nd land pressure under firing condition using a telemetry system. Unlike the grasshopper system, telemetry system does not provide direct linkage between the sensors and measuring equipment. The sensor signal is converted into a radio signal and transferred inside the cylinder block wirelessly. This system does not contain the disadvantages of the grasshopper system; however requires complicated electronic equipment for the conversion of sensor signal into radio signal, transmitting this signal, receiving and processing the signal, and providing electrical power to the electronics on the moving components. It's possible to obtain such devices from certain manufacturers.

Friction measurement will be performed in the same way as used by [11]. Connecting rod is instrumented by strain-gages. Instantaneous IMEP method is used to calculate piston system friction. Cylinder pegging method is used for maximum accuracy of the cylinder pressure measurement.

In addition, there'll be two absolute pressure transducers inside the piston, located in 2nd land. This is the intermediate region outside the piston, between the top and 2nd rings. The pressure in this region is determined by the combustion gasses that escape from the top ring. This pressure plays an important role for the stability of the top and 2nd rings, and thus affects the sealing of the system.

For this purpose, Ford 9L Diesel 380PS Euro5 Truck Engine was selected. The advantage of using such a big engine is the ease of access to the piston system components and relatively easy engine modification process. In addition, the engine provides maximum torque and maximum power at low engine speeds, which increases the lifetime of telemetry equipment and instrumentation, allowing for longer measurement periods and the possibility to reuse some of the equipment.

2. THEORY

2.1 Piston Telemetry

The best method to obtain sensor data from moving components, such as the internal components of the engine like piston and connecting rod, is the telemetry method. This method is used widely in the industry, by many vehicle manufacturers and suppliers. The main principle is to perform data transfer from moving components to stationary components, most of the time the engine block, without any physical contact. There are two methods for this data transfer, the simpler near field transmission and the more complicated far field transmission [16].

2.1.1 Near field data transmission

This transmission method uses electromagnetic induction to transfer data. The principle is the same as in a transformer. Two magnetic coils are brought up close to one another to transfer energy. The goal in a transformer is to change the voltage of the source; either to increase it for long distance transmission, or decrease it to power up electronic devices that need less voltage. In near field data transmission, the goal is not to change the voltage, but to provide the system the ability to physically remove the electrical connection [16. 17].

The two magnetic coils are called stator and rotor coils. These terms are driven from electric motors and used for piston telemetry as well, even though usually the rotor coil only makes a reciprocating motion.

Figure 2.1 represents the circuit for such a telemetry system.

The left hand side represents the circuit for the stator coil side. This coil is usually placed in the engine block. The wires are connected to the data acquisition equipment. The equipment provides alternating current to the coil, which generates alternating magnetic flux in order to induce alternating current on the rotor coil. The right hand side represents the circuit for the rotor coil side. The sensors or electronic equipment is represented by the resistance R , which is fed by the current induced on

the rotor coil. A simple example for this resistance is a thermistor, which is used to measure temperatures.

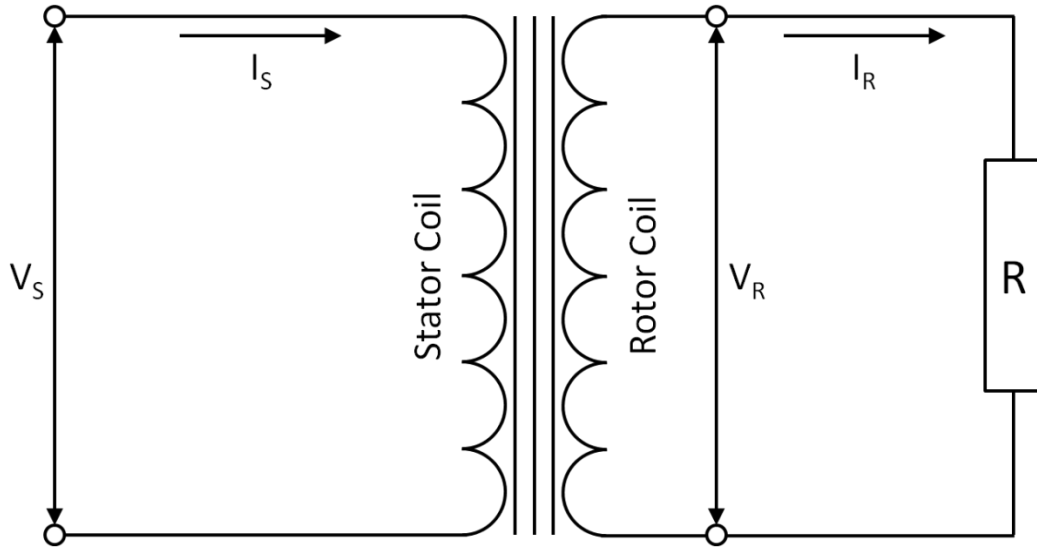


Figure 2.1 : Circuit for near field data transmission.

The formula for the relation between the voltages and currents of the stator and rotor side under ideal conditions is given in (2.1).

$$\frac{V_S}{V_R} = \frac{I_R}{I_S} = \frac{N_S}{N_R} = a \quad (2.1)$$

$$V_R = I_R \times R$$

The terms N_S and N_R represent the windings on the stator and rotor coils respectively. Assuming no difference between the windings of the stator and rotor coils, the system would produce the same voltage and current at each side of the circuit.

Depending on the voltage given from the source, the resistance will pull some amount of current or vice versa. In the case of a thermistor, the resistance will change according to the temperature in the location of the sensor, which can be read on the source. For example by using a constant current source on the stator side, the temperature can be read using the voltage across the stator coil. It is usually preferred to use a constant current source rather than constant voltage source, since the voltage generated by the resistance can be much more accurately read than the current.

This data transmission can only occur when the two coils are in close proximity of one another. For piston telemetry, since the rotor coil is moving together with the

piston or connecting rod, this can occur only during a short period of time, generally around BDC. Therefore the data transmission can only occur around BDC and not the entire cycle. This type of data transmission is called spot-mode transmission and it is more suitable for measurements such as temperature, where one data point per cycle is sufficient. For the measurements in the context of this thesis, friction and pressure, data has to be sent throughout the cycle in order to meet the requirements, therefore this transmission method is not suitable [16, 17].

2.1.2 Far field data transmission

This more advanced technique uses the same principle as in radio telecommunication. The signal that is to be transmitted, for the case of piston telemetry the sensor data, is converted into RF signal via modulation. The signal is broadcasted by the transmitter and picked up by the receiver antennas. The receiver then demodulates the signal back to its original state and thus the information is transferred. This method allows the data to be continuously sent, therefore was selected for the telemetry system used in this thesis.

This method requires electronics to be placed on the moving components, piston or connecting rod, in order to perform the signal modulation operations. One way to power the electronics is via using a battery. For piston telemetry, this battery has to be placed on the piston or connecting rod alongside the electronics. The weight and working principle of the battery does not allow for good power supply, due to the high acceleration and high temperatures reached during engine operation; so this is not a suitable method for piston telemetry.

In piston telemetry, these electronics need to be powered via magnetic induction. This system uses the same principle with near field data transmission but it only contains power transfer, like in a transformer. Due to the distance restriction of magnetic induction, the electronics can only be supplied with electricity for a limited time around BDC. Since continuous measurement is required, electricity has to be stored in capacitors and used throughout the cycle. Figure 2.2 shows the scheme of the piston telemetry system.

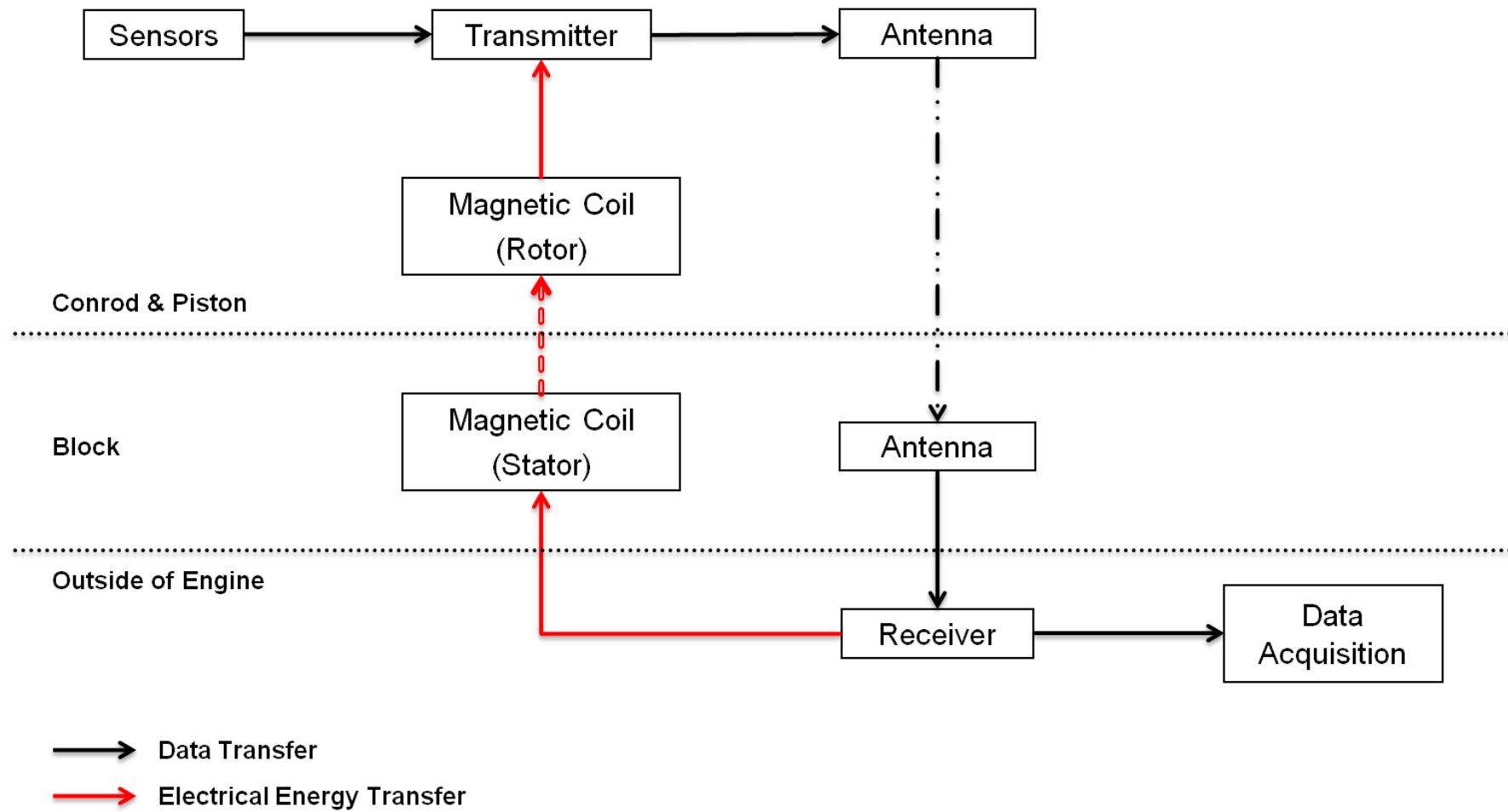


Figure 2.2 : Scheme for piston telemetry system

2.1.2.1 Signal modulation methods

In far field data transmission, most commonly used modulation methods are amplitude and frequency modulation, which are shown in Figure 2.3. In AM, the signal is converted into an electromagnetic wave of changing amplitude with a predetermined fixed frequency; whereas in FM, the signal is converted to a wave of fixed amplitude but changing frequency within predefined limits.

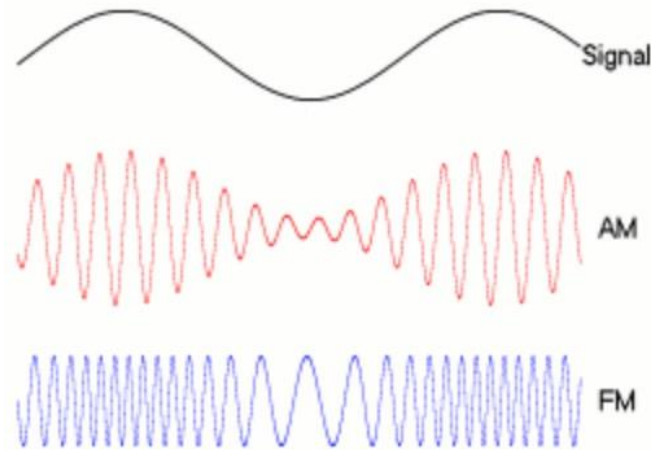


Figure 2.3 : Diagram for AM and FM.

AM signals are being used for very long distance communication. The signal frequencies are within around 150 kHz to 25 MHz. The biggest disadvantage of this type of communication is its weakness against noise, which causes minor differences in the amplitude of the wave, thereby disrupting the demodulated signal.

FM signal is the most widely used type. The mostly used frequency band is from 87.5 MHz to 108 MHz. The range is shorter than AM signals, but FM signals are much more robust against noise, since change in the amplitude of the wave does not have any effect on the demodulated signal. The frequency of the signal can be further increased to the range of 300 MHz for better noise cancellation, but at the cost of communication range.

Some of the piston telemetry system uses another type of modulation called frequency flicker modulation, FFM. In this method, similar to FM, the signal is transferred in fixed amplitude. The frequency of the signal switches between two predetermined frequencies; i.e. between 250 MHz and 300 MHz. The rate of this change is determined by the magnitude of the input signal. For example for a

pressure reading, the higher the pressure gets, the faster the signal switches between the two frequencies. For the example in Figure 2.4, the signal is read as “5”.

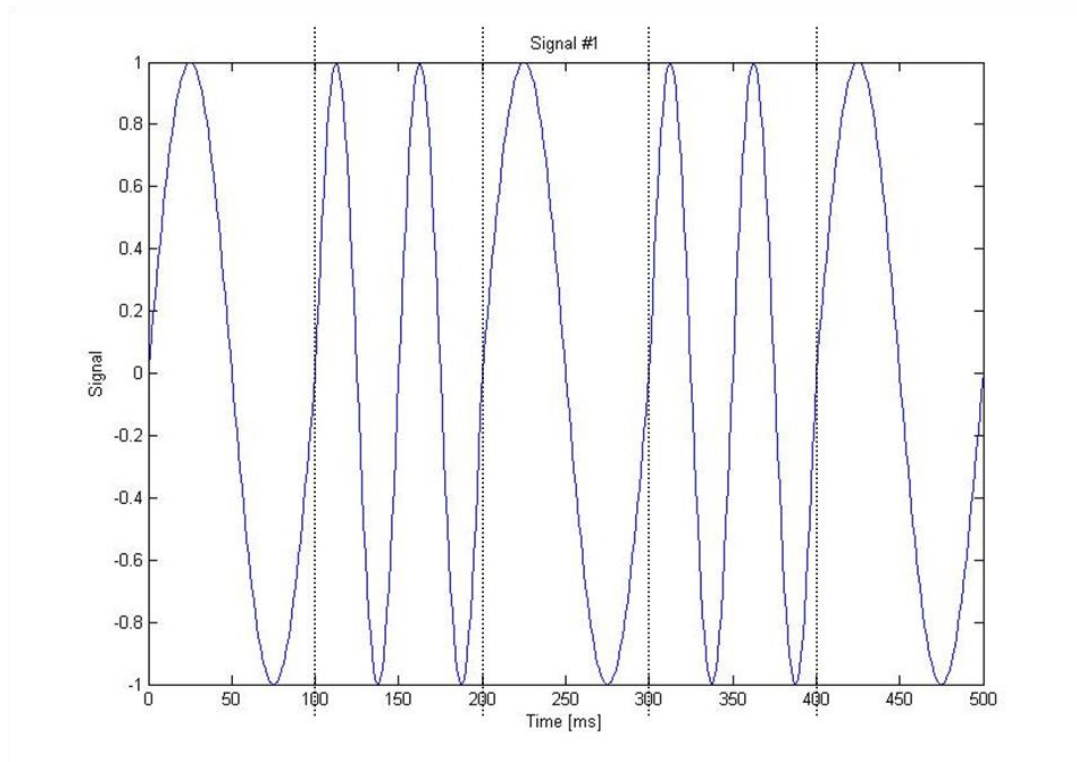


Figure 2.4 : Diagram for FFM.

With this method, the input signal has to be digitized and each data point has to be sent within one sampling period. Due to this limitation, the signal is generally converted into microwaves with frequencies in the order of a few GHz. This technique requires more expensive and precise equipment, however the signal virtually shows no distress against noise.

For the telemetry system used in this thesis, FM method was used because of the relatively simple design and the ability to multiplex signals, as explained in the next section.

2.1.2.2 Signal multiplexing

As is usually the case for telemetry systems, different signals from different measurements may have to be joined into one signal and transferred. This action is called multiplexing.

A simple method for multiplexing is to switch between the different signals, sending one after the other. This method makes it difficult to send continuous measurements.

One option is to make this switching slow; send one signal for several cycles or a few seconds, and then switch to the next signal. This is not preferred for piston telemetry due to small changes in the engine conditions and cycle-to-cycle variation. If FFM method is used for modulation, it is possible to squeeze all of the signals into one sampling period; however this would impose a bigger demand on the electronics and would reduce signal resolution.

Another method is to use subcarrier signals. In this method, each signal is modulated individually using a different property; for example if FM is used, each signal is modulated within a different frequency range but same amplitude. These modulated signals are called subcarrier signals. These signals are then added up on top of one another. The resulting signal contains both of the channels, but now varies in amplitude; so to reduce the effect of noise, the new signal is treated as if it's the measured signal and it's re-modulated to a higher frequency, again using FM.

After the resulting signal is picked up by the receiver, it's demodulated using FM. The resulting signal contains two different data sets, so band-pass filters are used to separate the signals. These filters allow signals within certain frequency range to pass, killing the other frequency components. After the filtering operation, subcarrier signals have been regenerated, which are placed in another demodulation to obtain the initial sensor data.

There are other modulation methods available; however the subcarrier signals method is one of the simplest and fulfils the requirements, therefore was used for the piston telemetry system.

Figure 2.5 is taken from the Datatel manuals; it shows the detailed scheme for the signal processing in the telemetry system.

2.2 Friction Force Calculation

Instantaneous IMEP method is used to calculate piston system friction. The method uses force balance to calculate piston friction. Figure 2.6 shows the locations of the sensors used for this method.

The two pressure sensors in the cylinder and crankcase are used to calculate net pressure applied onto the piston. The third sensor near BDC is used to correct the measurements of the cylinder pressure sensor using the pegging technique.

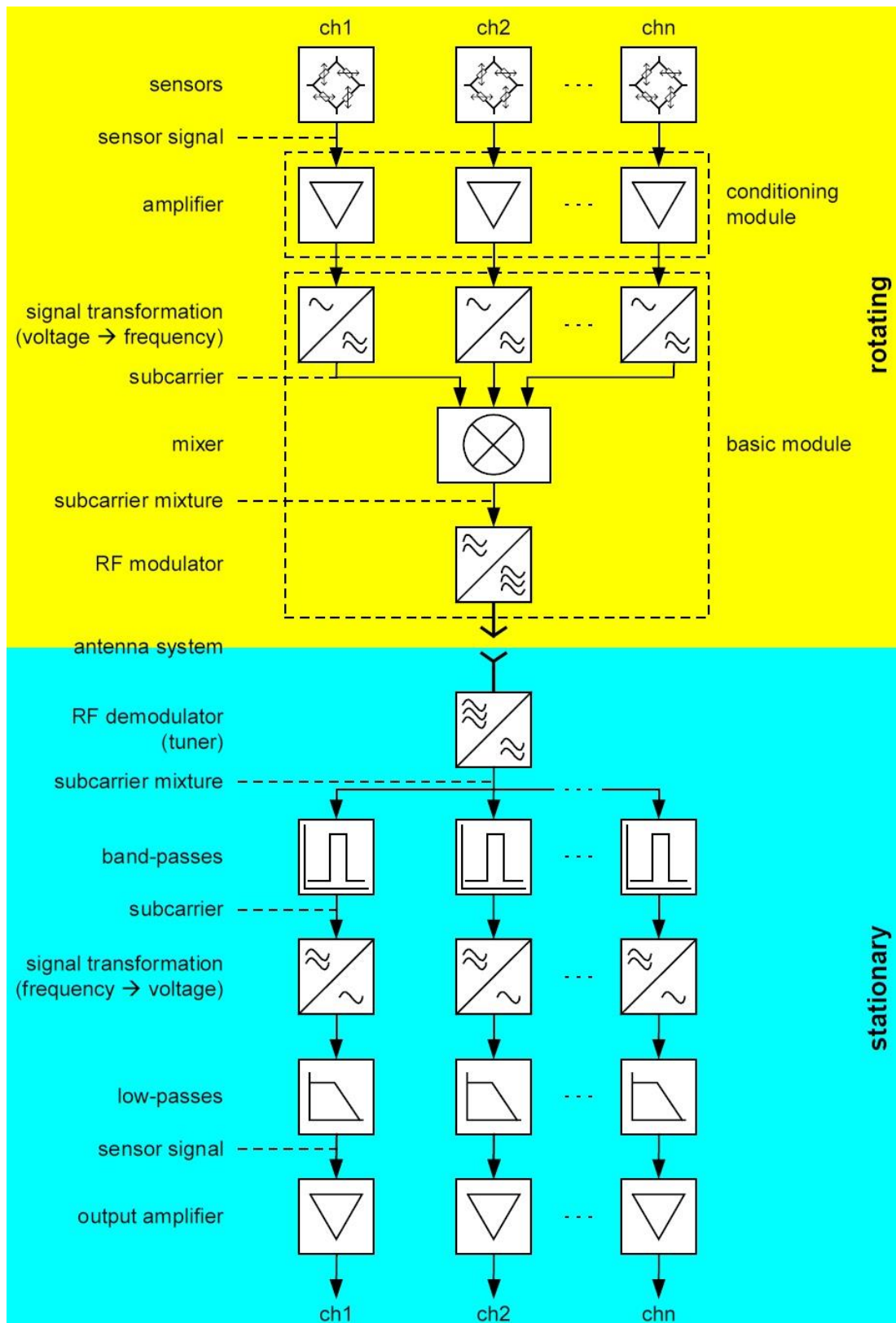


Figure 2.5 : Detailed scheme for piston telemetry system.

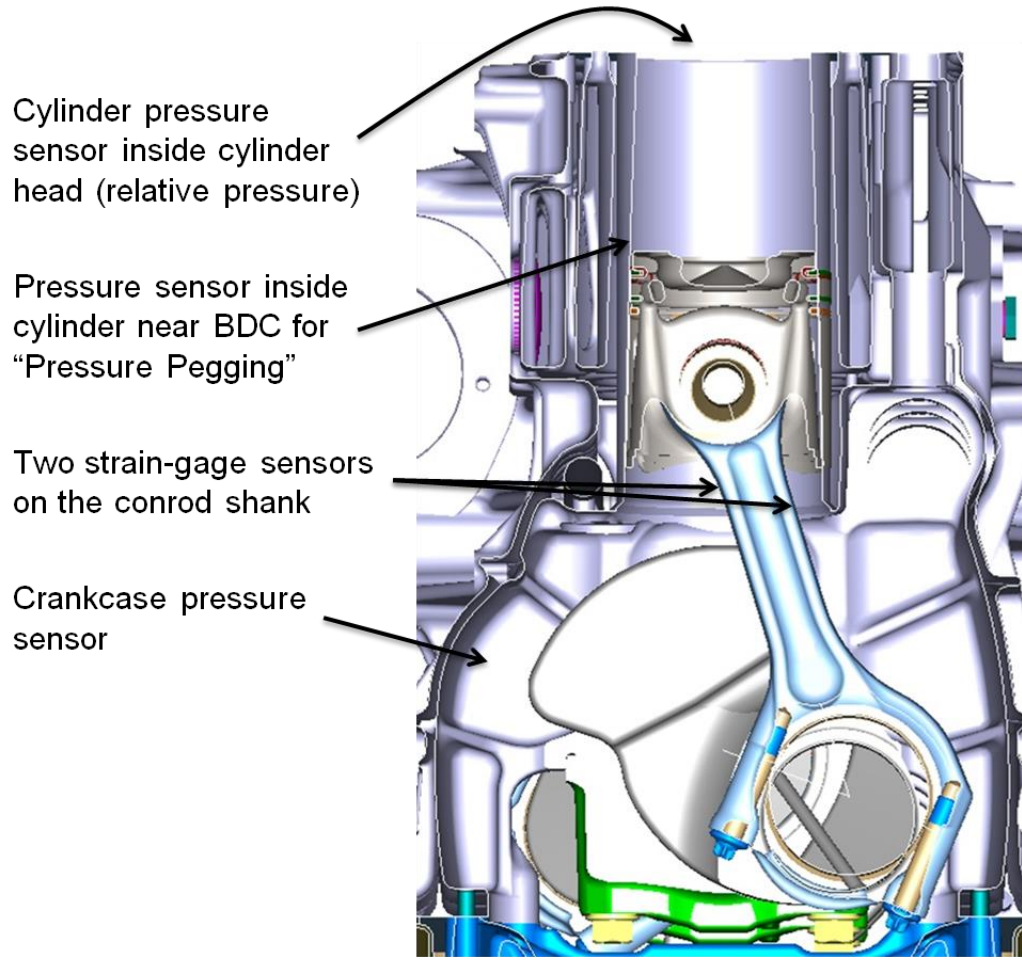


Figure 2.6 : Location of the sensors used for friction calculation.

A simplified version of the free body diagram of the piston is given in Figure 2.7. This figure only shows the forces of interest and does not include side forces or moments involved.

F_G is the net gas force, F_W is the gravitational force due to the weight of the piston system, F_I is the piston system inertia force, F_F is the frictional force, F_{Rod} is the force applied to the connecting rod and β is the angle of the connecting rod with respect to cylinder axis.

Using the force balance, the frictional force can be calculated as in (2.2).

$$F_F = F_G + F_W - F_I - F_{Rod} \cos \beta \quad (2.2)$$

Gas force is calculated using the relation in (2.3).

$$F_G = A_{cyl}(P_{cyl} - P_{cc}) \quad (2.3)$$

A_{Cyl} is the cylinder area, P_{Cyl} is the absolute gas pressure inside the cylinder and P_{CC} is the absolute pressure inside the crankcase.

F_W is calculated using the weight of all the components in the piston system; including bare piston, 3 rings, pin and 2 circlips (small rings that hold the pin inside the piston).

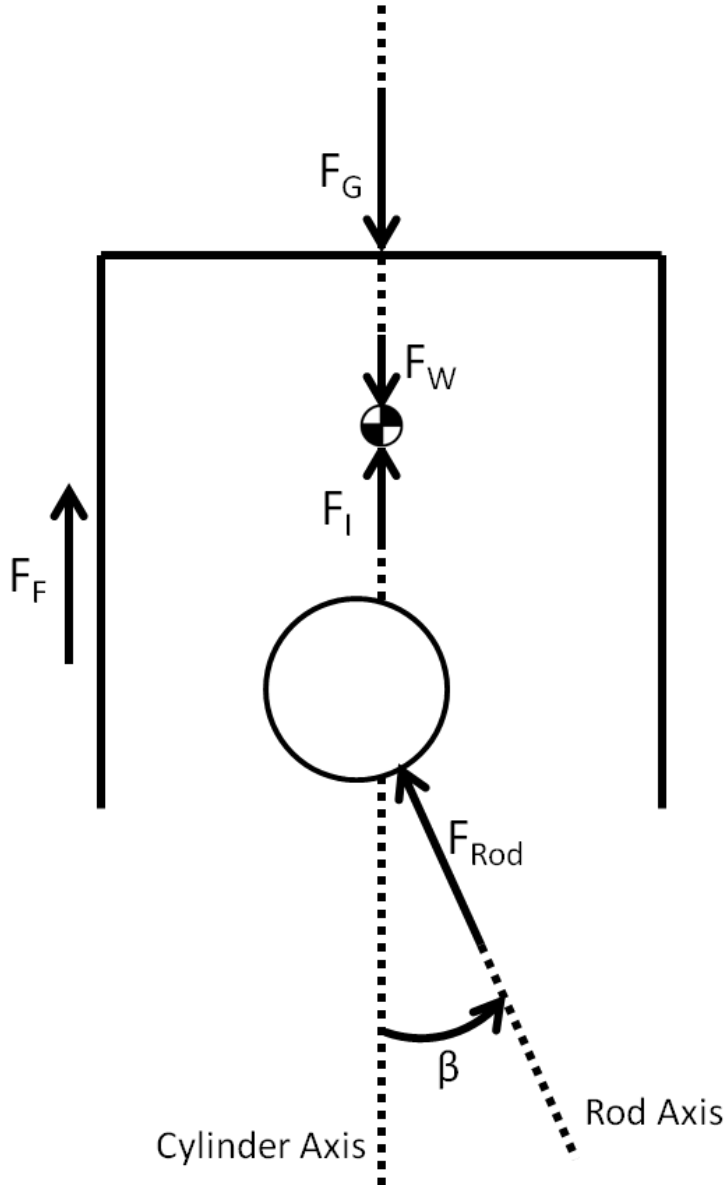


Figure 2.7 : Simplified free body diagram of the piston.

2.2.1 Piston system inertia force calculation

Inertia of the piston system is calculated using the 2nd derivative of the equation of motion of the piston, $\ddot{x}$ and piston system mass, m_p , as shown in (2.4).

$$F_I = m_p \ddot{x} \quad (2.4)$$

The dimensions involved for this motion are shown in Figure 2.8.

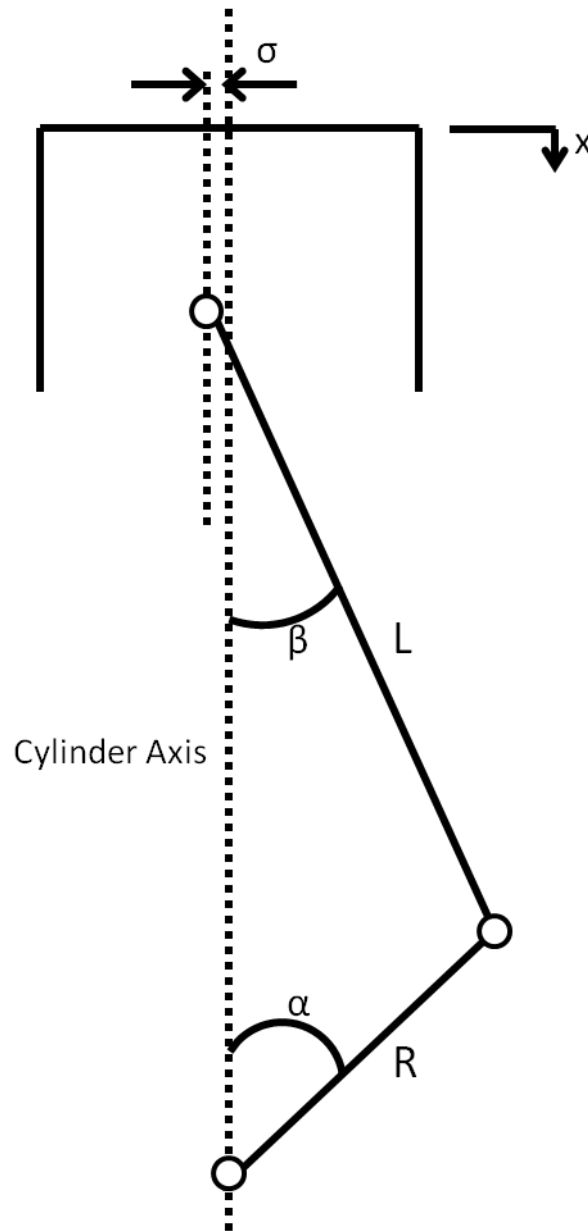


Figure 2.8 : Dimensions involved in piston equation of motion.

R is the crank throw, L is the length of the connecting rod, α is the crank angle where 0° corresponds to TDC and σ is the pin offset, the distance between cylinder axis and pin location inside the piston (the reason for this distance is to create additional moments that helps in balancing the side forces on the piston skirt). $\ddot{x}$ denotes the start and direction of the motion. TDC is assumed to be the starting position where x

equals to 0. As the piston moves down, x increases. At BDC, x equals to the stroke value.

Using geometry, the piston equation of motion can be calculated as in (2.5).

$$x = R + L - R \cos \alpha - \sqrt{L^2 - (R \sin \alpha + \sigma)^2} \quad (2.5)$$

In this equation; R , L and σ are constants. Crank angle, α , is measured while the engine is running, using the crankshaft encoder.

The engine speed is never constant; even at a constant rpm, the engine makes small speed fluctuations. This occurs due to the nature of the combustion process. Each cylinder fires rapidly and provides a strong but short burst of force. The engine speeds up slightly as each cylinder combusts and slows down in between the combustions. (The flywheel aims to reduce these speed fluctuations by causing a significant increase in engine rotating inertia.)

While taking the derivative of piston motion with respect to time, engine speed fluctuations must be taken into account in order to minimize the error for friction calculation.

Using same geometric principles, it's also possible to express connecting rod angle β in terms of crank angle, as in (2.6).

$$\beta = \sin^{-1}((R \sin \alpha + \sigma)/L) \quad (2.6)$$

2.2.2 Connecting rod force calculation

The strain gages on the connecting rod will be used to measure the force in the connecting rod with respect to its principle axis. However, this force does not equal to the force applied to the rod by the piston. The strain-gages are placed in between the big end (the region that connects to the crankshaft via bearings) and small end (the region that connects to the piston via the pin), which is called rod shank. The section above the strain gages will yield an inertia force. This force is supplied from the piston force.

For simplicity, connecting rod is mathematically modelled as a cylinder for the small end and a rod with constant cross section for the shank. The small end section

performs the same movement as with the pin, so the weight and inertia of this section was added to the piston assembly.

The free body diagram of this model is given in Figure 2.9. The bending moments and forces perpendicular to the principle axis of the rod are not taken into account, since strain-gages have excellent bending compensation due to the usage of the Wheatstone bridge.

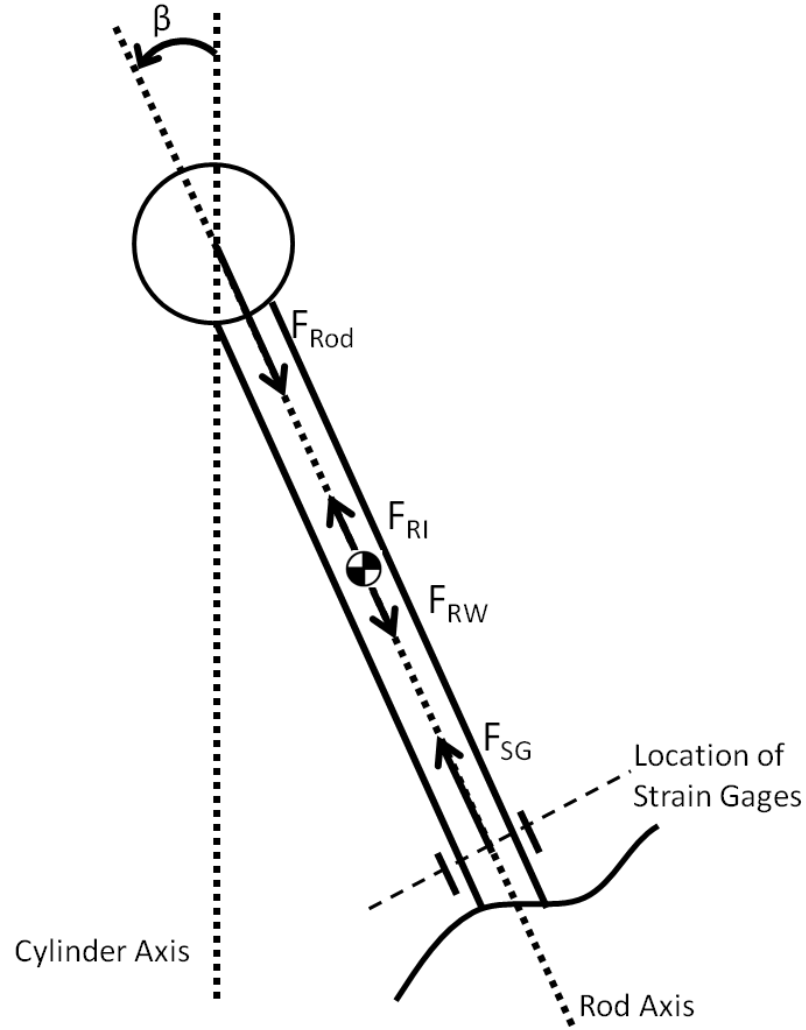


Figure 2.9 : Simplified free body diagram of the connecting rod upper region.

F_{SG} is the force read by the strain gages, F_{RI} and F_{RW} are the inertia and weight forces of the upper section of the rod shank respectively. The relation is simply written in (2.7).

$$F_{Rod} = F_{RI} + F_{SG} - F_{RW} \quad (2.7)$$

2.2.3 Connecting rod inertia calculation

The shank region of the connecting rod performs a complicated motion containing both reciprocating and circular motion. The reciprocating motion is similar to the motion of the piston. The centre for the circular motion is the small end, where the rod rotates around this centre with the angle β .

For an arbitrary distance of “r” from the small end centre, the acceleration is given in (2.8).

$$a_r = \ddot{x} \cos \beta - r \dot{\beta}^2 \quad (2.8)$$

The first component denotes the reciprocating motion and the second component denotes circular motion.

This value is integrated over the complete region to find the inertia force. It's assumed that this region follows the same cross-section, therefore it's possible to show the mass of this region outside of the integral as m_R .

Two distances are defined for the limits of this region; r_1 is the starting distance, right under the small end and r_2 is the location of strain gages. The integration is performed in (2.9).

$$F_{RI} = m_R \int_{r_1}^{r_2} a_r dr \quad (2.9)$$
$$F_{RI} = m_R \left(\ddot{x} \cos \beta - \frac{r_2^2 - r_1^2}{2} \dot{\beta}^2 \right)$$

Using the formulas from (2.2) to (2.9), piston system friction force can be calculated. MATLAB codes were generated for this purpose, which can be found in the appendix section.

2.3 Wheatstone Bridge Configuration

This is a widely used standard for sensory measurements due to various advantages like high sensitivity and good compensation capabilities. The configuration contains 4 resistances connected as shown in Figure 2.10.

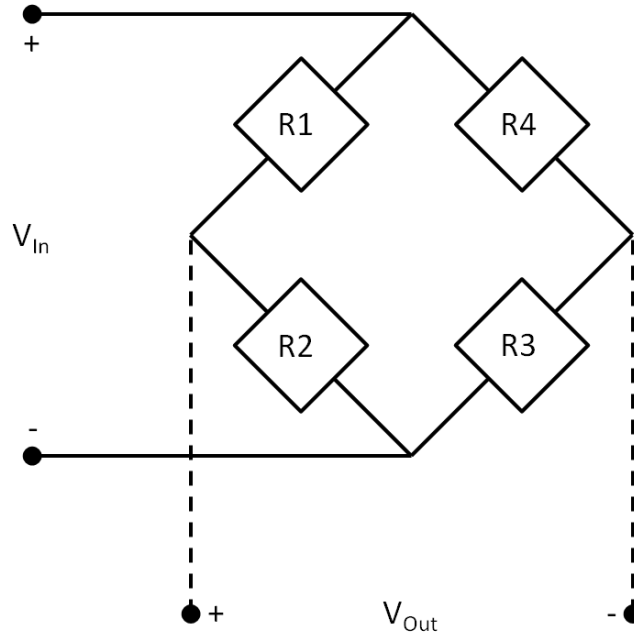


Figure 2.10 : Wheatstone bridge configuration.

The circuit is supplied with a predetermined fixed voltage from V_{In} . If all the resistances are the same, output voltage will be zero. Depending on the application, some or all of these resistances can be sensors, where others would be dummy resistors. The sensors and dummy resistors are selected such that they have the same resistance at a certain condition; i.e. for strain gages when there's no strain. A small change in the resistance of the sensor causes bigger changes on the output voltage; therefore the output voltage can be measured to read sensory input.

The output voltage is calculated using the formula in (2.10).

$$V_{Out} = V_{In} \times \left(\frac{R3}{R3 + R4} - \frac{R2}{R1 + R2} \right) \quad (2.10)$$

Strain-gages are small stamp like sensors that are placed on a flat surface. The gage has a certain resistance once placed on the part. As the part strains due to stress, the length and/or shape of the gage changes. If the gage is expanded, its resistance increases.

Various configurations were considered for connecting strain-gages, which are shown in Figure 2.11.

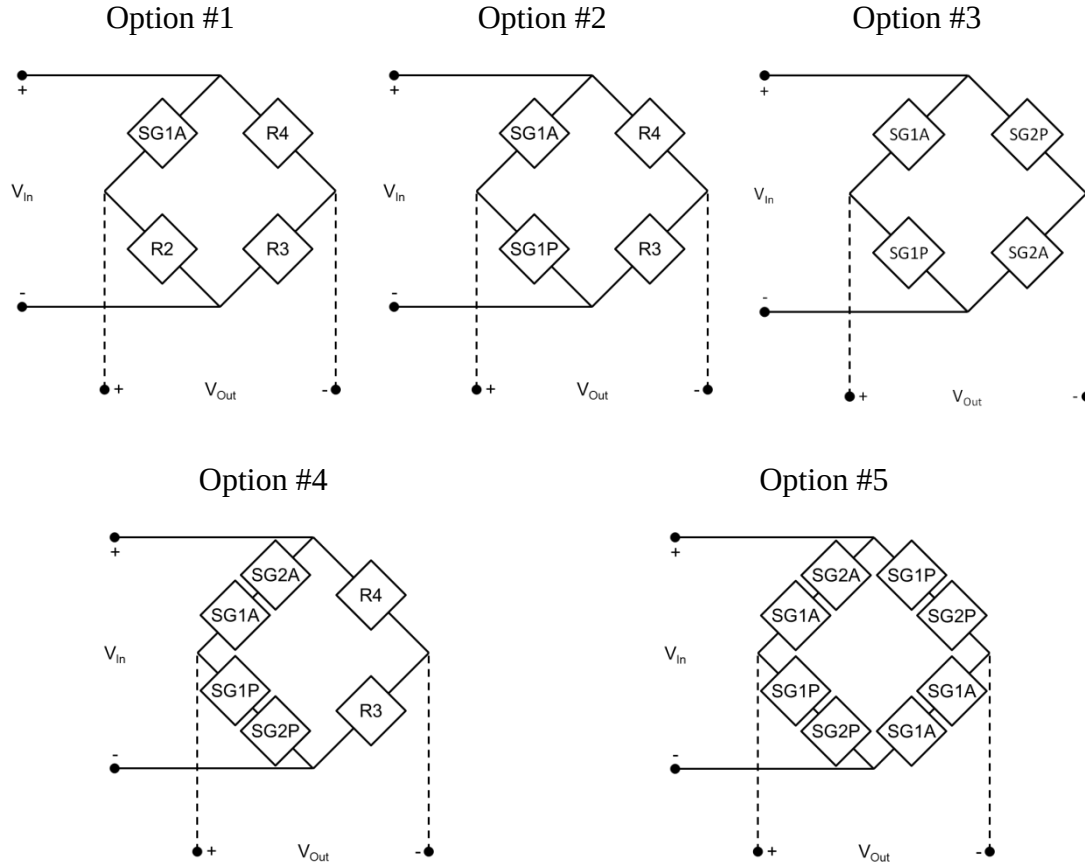


Figure 2.11 : Options for Wheatstone bridge configuration.

SG1A and SG2A indicate active strain gages that are placed in the direction of the principle axis of the connecting rod. SG1P and SG2P indicate passive gages (Poisson gages) placed in the vertical direction. This is achieved using 90 degree tee-rosette type strain gages. R2, R3 and R4 indicate the dummy resistors.

Option #1 is the reference, containing only one active gage. Option #2 contains one active and one passive gage. The rest of the options contain two active and two passive gages connected in different configurations. Option #3 is the most widely used configuration.

A calculation was performed to understand the advantages and disadvantages of each configuration. Three scenarios were considered:

- Axial elongation of the rod: Resistance of active gages are increased, rest of the resistances are unchanged.

- Bending of the rod: Resistance of one of the active gages are increased, whereas the other gage is decreased by the same amount. Rest of the resistances are unchanged.
- Temperature increase: Resistance of all of the gages are increased, including the passive gages. The resistance of dummy resistors are unchanged.

The objective is to provide bending and temperature compensation, where the output voltage does not change for the scenarios #2 and #3. In addition, the output voltage is required to be as much sensitive to axial elongation as possible.

Table 2.1 is constructed for the above scenarios. In each case, sensor resistance is changed by 10% and its effect on output voltage is written.

Table 2.1 : Sensitivity of Wheatstone bridge configurations to different scenarios.

	#1	#2	#3	#4	#5
Axial	2.38%	2.38%	4.76%	2.38%	4.76%
Bending	2.38%	2.38%	0.25%	0%	0%
Temperature	2.38%	0%	0%	0%	0%

Option #1 and #2 are sensitive against bending and/or temperature, therefore are not suitable. Option #4 has excellent bending and temperature compensation but shows less sensitivity to axial elongation, therefore it is not suitable. Option #5 has the best abilities, however requires a very complicated setup which imposes difficulty for instrumentation on the rod. Option #3 was chosen as the optimum solution due to high axial sensitivity, acceptable bending compensation, excellent temperature compensation and ease of instrumentation.

3. EXPERIMENTAL METHOD

3.1 Telemetry Equipment

The electronic equipment required to setup the telemetry system has been acquired Datatel Telemetry. The following units have been prepared by Datatel:

- Transmitter Unit: 2 units with 2 channels each
- Magnetic Coils: Stator and rotor units
- Brackets for mounting the stator unit on the ladderframe
- Receiver and data acquisition unit
- Inductive power generator unit
- Adapter box

Datatel performed the following modifications and instrumentations on the connecting rod:

- Mounting transmitter units
- Mounting rotor coil unit onto the big end
- Instrumentations of magnetic coil and antenna connections to the transmitters

Furthermore, Datatel provided the necessary information and technical drawings to modify the block and ladderframe (the part that is placed underneath the block which aims to improve block strength) in order to mount the stator coil unit and antennas.

The two transmitters on the connecting rod have been placed on each side of the shank region of the part. The units work separately in different frequency bands. Each unit is capable of sending data from two different channels. One unit is responsible for sending the pressure data from the two pressure sensors inside the piston. The other unit sends the strain-gage data and the temperature of the connecting rod. This temperature data will only be used to make sure that

transmitters don't exceed the recommended temperature limit. These four measurements will be taken from four different channels.

The receiver and inductive generator units are placed outside of the engine. They work in tandem to provide the necessary inputs and outputs of the system. The receiver unit is connected to the electric grid power supply. Through the "power out", it sends most of this electricity to inductive power generator in order to power the stator coil. This operation is controlled using the "control signal" output in the receiver. The resulting electrical power is transmitted to the stator coil using the "primary coil" output in the inductive power generator.

The inductive power is passed through the adapter box where no signal processing takes place. The role of this box is to collect sensor data from the three antennas: Two are placed inside the block in each side, and one is placed above the stator coil. The data is collected into one antenna wire, and sent back to the inductive power generator using the same wire bundle with the electrical power (this wire bundle in itself contains different wires for these different operations).

The antenna information is sent back to the receiver unit using the "RF Out" output in the inductive power generator. The receiver processes and separates the four sensor signals and outputs them via the four BNC outlets.

Following figures show the equipment sent by Datatel Telemetry. Figures 3.1 shows the instrumented connecting rod, Figure 3.2 shows the stator coil, Figure 3.3 shows the receiver and inductive power generator units, Figure 3.4 shows the adapter box unit, Figure 3.5 shows the scheme of the complete telemetry system and Figure 3.6 shows a CAD representation for the placement of the equipment in the engine.



Figure 3.1 : Connecting rod with transmitter and rotor coil units.

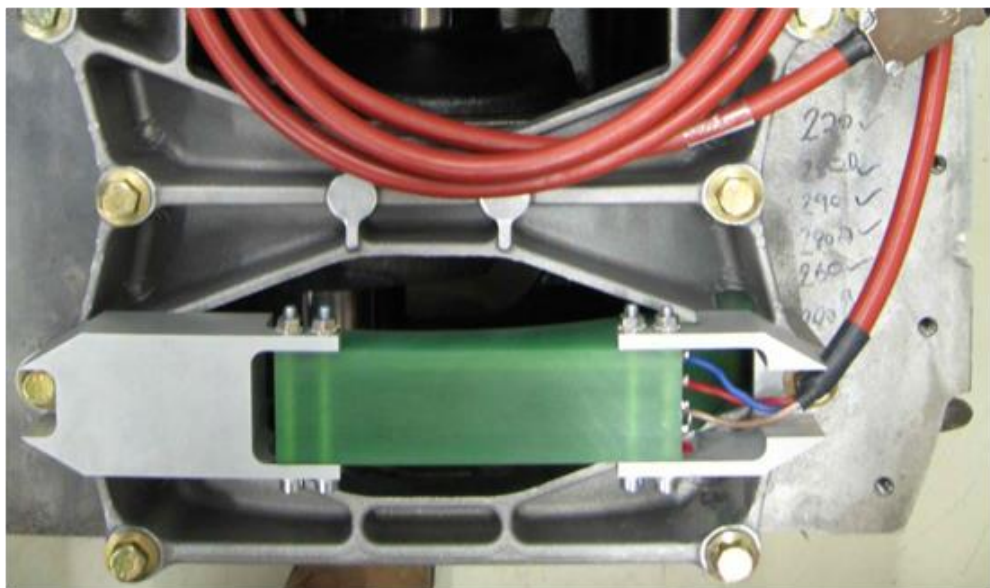


Figure 3.2 : Stator coil unit mounted on the ladderframe.



Figure 3.3 : Receiver and inductive power generator units.



Figure 3.4 : Adapter box.

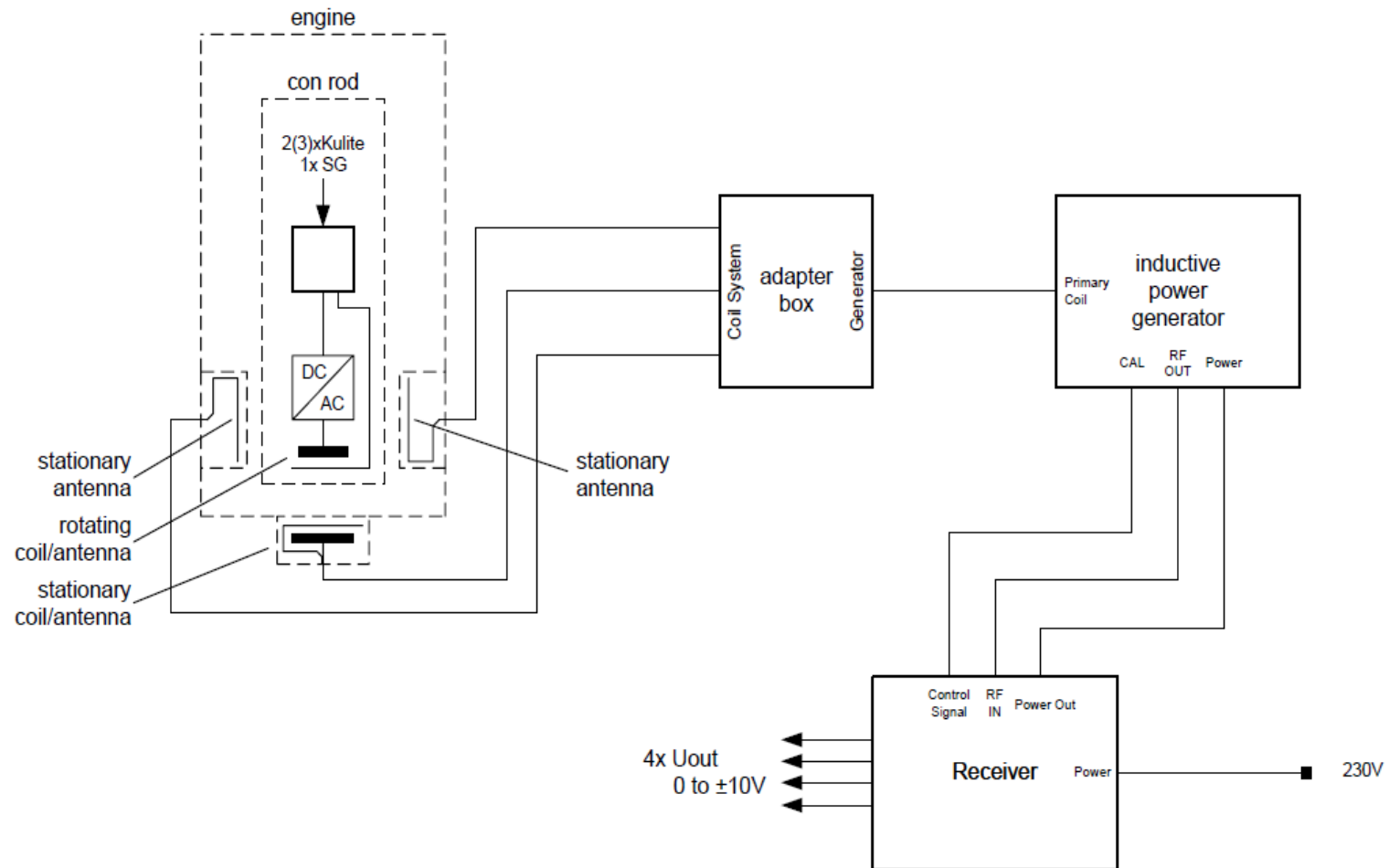


Figure 3.5 : Scheme of the telemetry equipment.

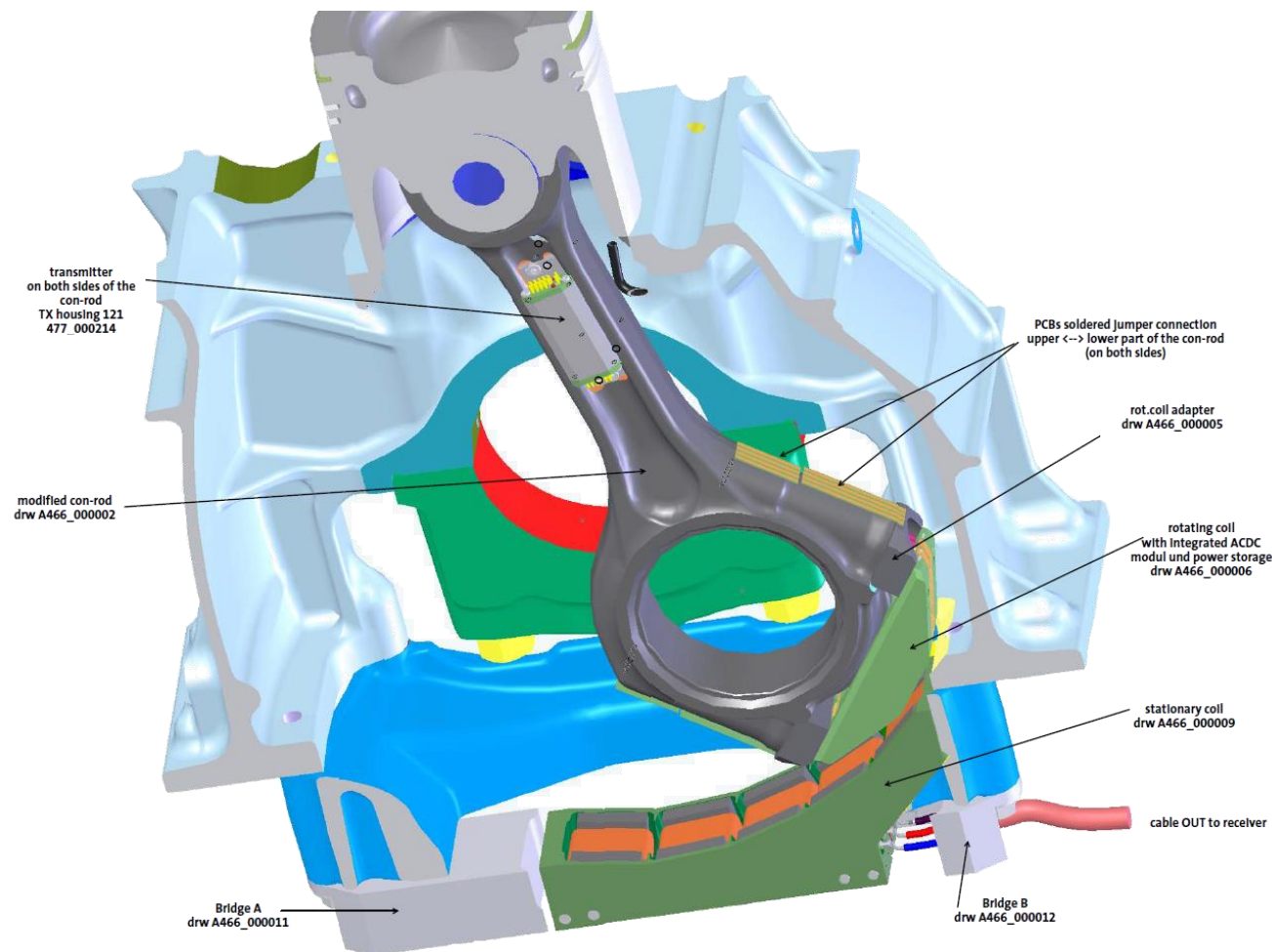


Figure 3.6 : CAD representation of the telemetry equipment.

The receiver unit contains multiple knobs for adjusting operations. These are the “Gain” and “Zero” adjustment knobs, and inductive power adjustment knobs.

The gain and zero adjustment knobs perform fine tuning operation on the output signal. The gain knob changes the amplitude of the signal, whereas the zero knob shifts the signal up or down. This operation can be performed separately for all of the four channels. Note that only fine adjustments can be performed with these knobs, it’s not possible to cause a significant effect on the output signal. These knobs are blocked after calibration is completed to avoid any change in calibration curves.

The inductive power adjustment knobs change the magnitude of the inductive power being sent to the stator coil. There are two knobs, the upper knob is used when the engine is in operation, whereas the lower knob is used when the engine is at standstill (for standstill operation, the coils must be close by; the piston must be around BDC).

When the inductive power is not enough for the electronics on the connecting rod to work properly, the signal quality drops. This can be counteracted by the inductive power adjustment knob. However, care must be taken to prevent exceeding the allowable voltage limit on the transmitter, which must be checked from the “Level Contr. Out” output at the back of the receiver.

Figure 3.7 shows the locations of these adjustment knobs.

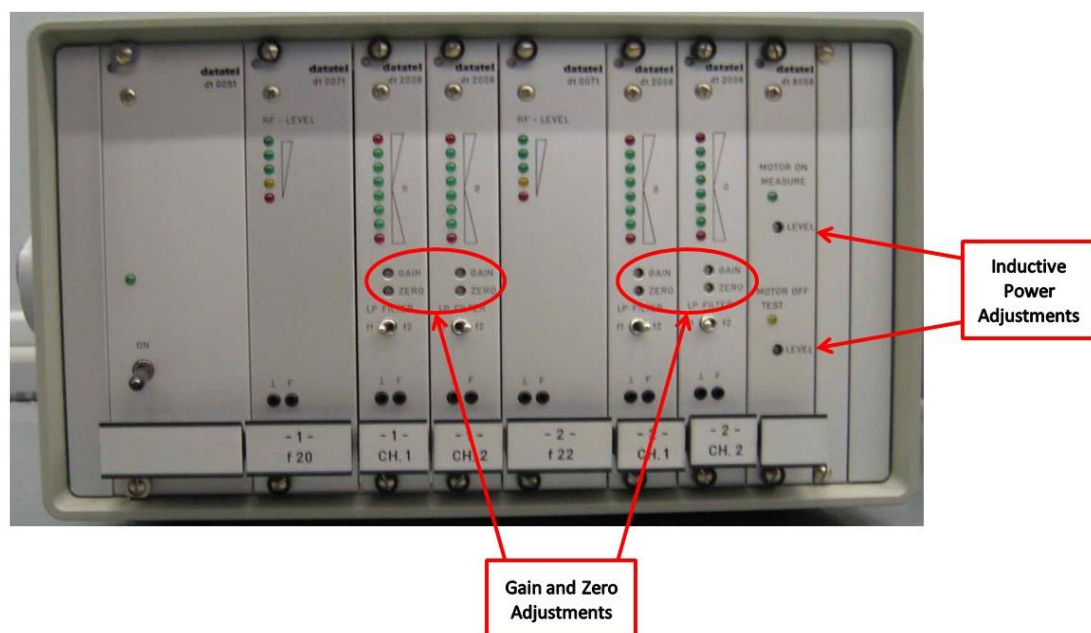


Figure 3.7 : Location of the Adjustment Knobs on the Receiver Unit.

3.1.1 Package checks

During the design of the telemetry system, engine CAD data had been used to avoid any contact with engine components. After the system was obtained from Datatel, packaging checks have been performed via endoscope to make sure that the telemetry equipment does not clash with the neighbouring components. No issues were found. Figure 3.8 shows the CAD checks and Figure 3.9 shows the endoscope check.

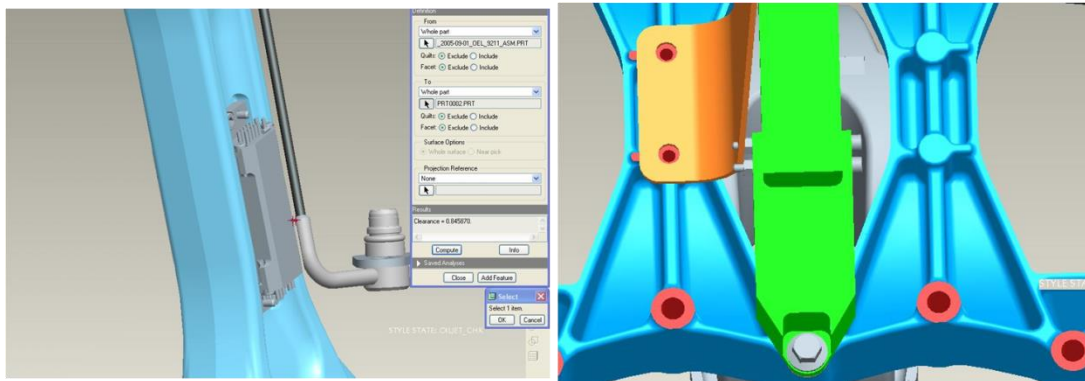


Figure 3.8 : CAD packaging check.

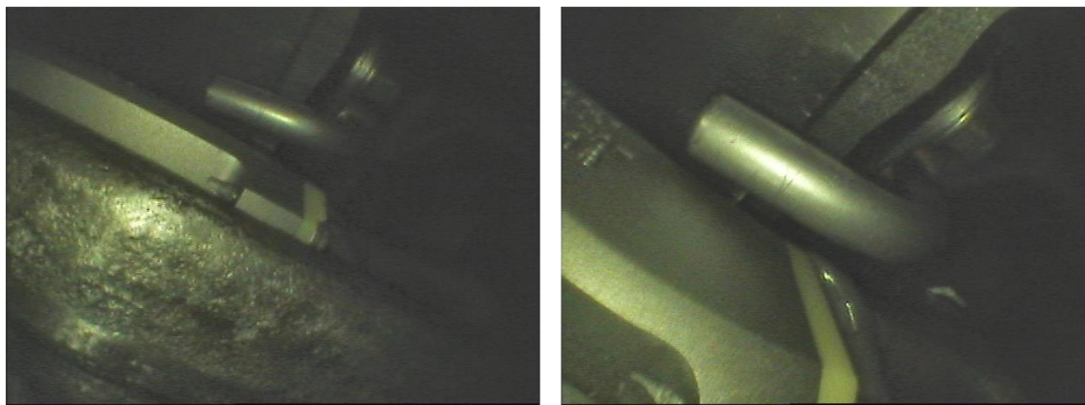


Figure 3.9 : Packaging check with endoscope.

3.2 Piston Instrumentation

In order to measure the pressure of the 2nd land region of the piston, two pressure sensors were used; one on the thrust side, the other on the anti-thrust side.

Apart from the motion along the cylinder axis, the piston is pushed side to side inside the cylinder, resulting in piston secondary motion. This motion is due to the angle between the cylinder axis and connecting rod axis. Side forces are applied onto the piston due to this angle, causing the secondary motion. These forces are highest

when combustion reaches the peak pressure. The side that the piston leans on during combustion is called thrust side.

Most engines turn in clockwise direction when looked from the front of the engine. Since combustion occurs after TDC when the connecting rod is tilted to the right, thrust side of the piston is on the left when looked from the front of the engine.

The 2nd land region comes very close to the cylinder bore during this tilting motion; sometimes contacting the bore. Because of the changing volume of the region between 2nd land and cylinder bore, it's useful to measure the pressure from two different locations.

Figure 3.10 shows the location of these sensors in the piston.

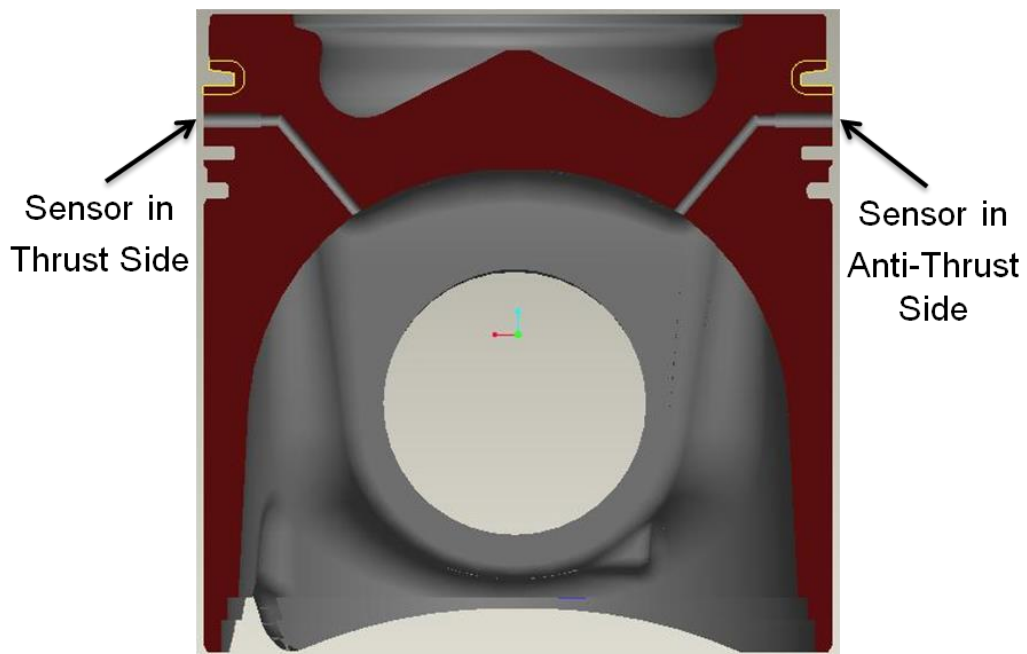


Figure 3.10 : CAD representation of the sensor locations inside the piston.

The pressure sensors selected are called Kulite high temperature miniature pressure transducers. These sensors are able to measure up to 70 bars. Each sensor is factory calibrated and provided with a datasheet. Each sensor contains a separate bridge unit (which were mounted on the connecting rod). This bridge unit provides temperature compensation for up to 235°C. The sensors are very small and thus are well suited for placement inside the piston.

This sensor was also selected to be used in pegging operation.

Figure 3.11 and Figure 3.12 show the pictures of the sensors.

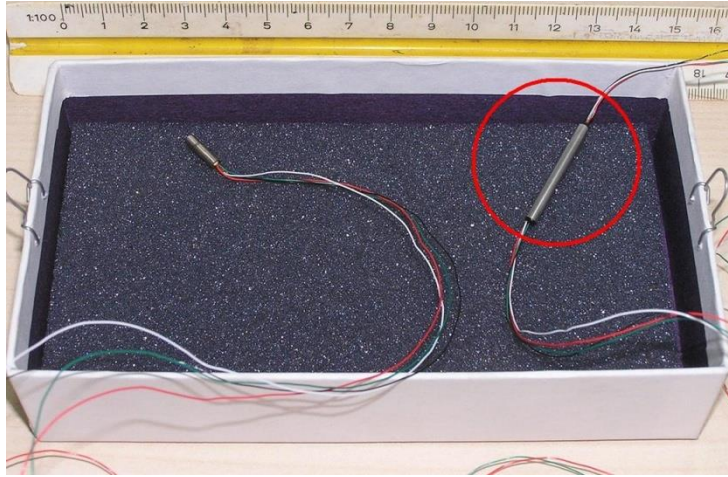


Figure 3.11 : Kulite pressure sensor, the bridge is highlighted.

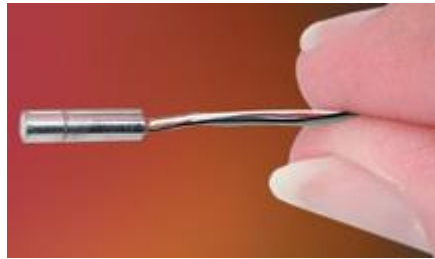


Figure 3.12 : Size of the Kulite pressure sensor.

The stages of piston instrumentation are as follows:

- Placement of the two pressure sensors inside the drilled holes: A delicate operation that requires needle and tweezers.
- Rubbing the piston with sandpaper along the route of the cables: The goal is to provide maximum sticking of the cables onto the piston.
- Sticking terminals to the pin boss for cable connections.
- Cabling from the sensor to the terminals and soldering of connections using solder with silver content for high temperature resistance.
- Covering up the cables with adhesives: JB Weld was used for piston instrumentation. It is a very strong two stage epoxy which can even be drilled once solidified. The aim is to hold the cables in place under the heavy acceleration loads reached during engine operation.

Figure 3.13, Figure 3.14, Figure 3.15 and Figure 3.16 show the stages of instrumentation.



Figure 3.13 : Placement of sensors inside the piston.



Figure 3.14 : Smoothing the surface inside the piston with sandpaper.



Figure 3.15 : Placement of sensor cables and terminal for cable connections.



Figure 3.16 : Covering up the cables using JB Weld.

3.3 Connecting Rod Instrumentation

3.3.1 Strain-gage placement

Strain-gage measurement must only show the axial strain along the principle axis of the connecting rod, the bending strains must not be measured. For the Wheatstone bridge to compensate the bending strain, two gages must be placed symmetrically on each side of the connecting rod.

Two locations were considered for strain-gage placement, which are shown in Figure 3.17.

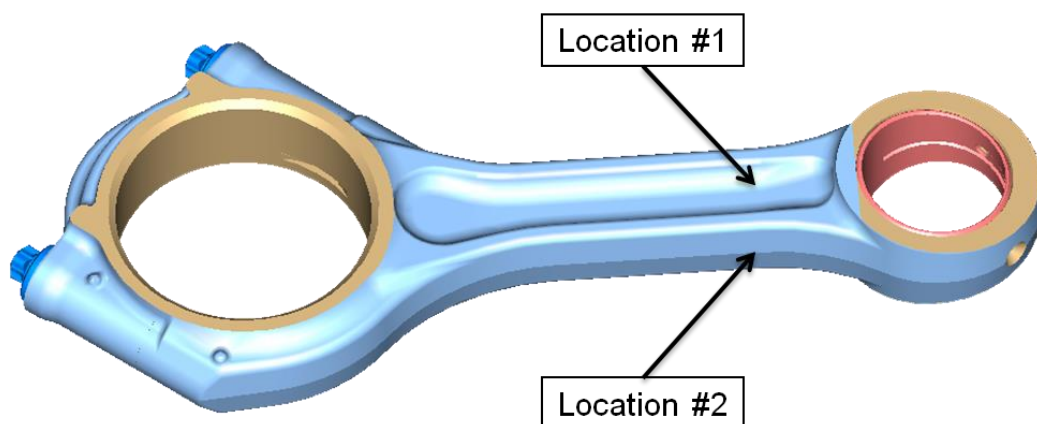


Figure 3.17 : Possible strain-gage locations on the rod.

Location #1 shows very little bending strain, therefore is the preferred option for measuring purely axial strain. However, application on location #1 is difficult because of the absence of flat surfaces and the crowdedness caused by the cables going to the transmitters.

Location #2 doesn't have these disadvantages, however it exhibits much more bending due to side forces applied by the piston. Placement of gages on location #2 will cause one gage to expand and the other to shrink as the rod bends. This will allow the Wheatstone bridge to minimize the effect of bending thanks to excellent bending compensation. Because of these reasons, location #2 was selected.

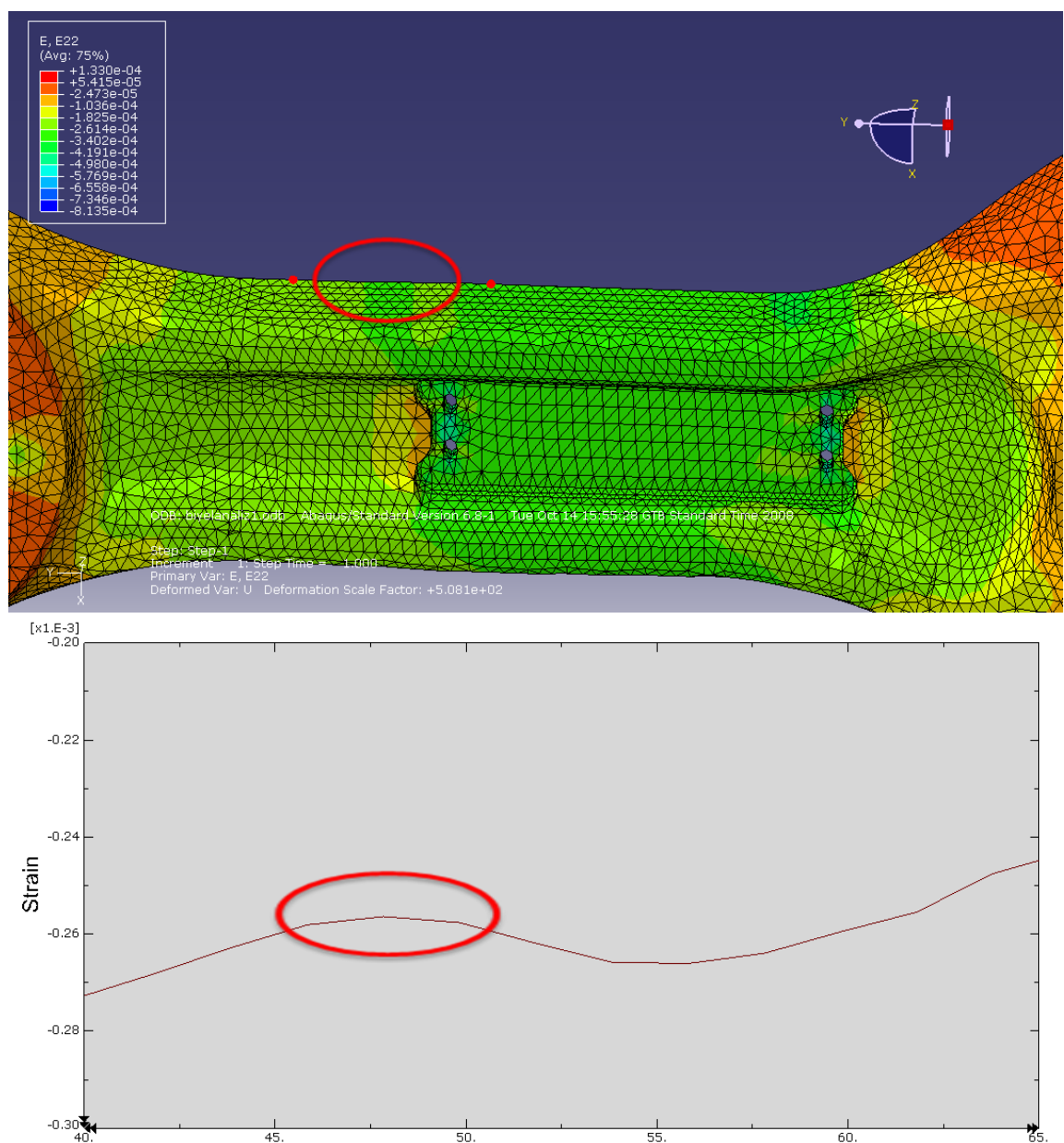


Figure 3.18 : Result of rod FEA and selected gage location.

It's not possible to place the two strain gages at exactly the same distance from the small end centre; one gage is bound to be placed higher than the other, albeit by a few tenths of a millimetre. The gages must be carefully placed on the spot with as constant strain as possible. This will minimize the strain difference between the two gages and hence the measurement error. A FEA was run on the connecting rod to find such a spot.

The final input to the strain-gage placement was the distance to the small end. This was aimed to be minimized in order to minimize the inertia effect.

Figure 3.18 shows the result of the FEA and the selected region.

It was found that there are locations further away from the small end with almost constant strain. However the highlighted region was selected due to closeness to the small end and acceptable strain stableness.

The FEA was also used to check that the drillings for the transmitter housings will not cause durability issues. It was found that the factor of safety is above recommended limits despite the modifications.

3.3.2 Instrumentations

For instrumentation, the following adhesives were used:

- Strain-gage M-Bond curing agent: The gage needs to be fully bonded with the connecting rod for successful strain measurement. After the gage is placed and the cure is applied, the rod is placed in oven for heat treatment. Temperature of the oven was carefully monitored to make sure that the temperature limit of the transmitter is not exceeded.
- JB Weld (Same as the one used in the piston)
- Vishay Micro-Instruments Dow Corning: Forms a softer layer that can be removed. This was selected to allow for cable or soldering replacements in the case of a problem and disconnection.

These three adhesives are tested in a trial engine in an engine dynamometer. No problems were observed during the test.

Connecting rod instrumentations include strain gage placement, pressure sensor bridges placement (for the sensors in the piston), placement of terminals for the

connection to the piston and the necessary cabling. Figure 3.19, Figure 3.20 and Figure 3.21 show the instrumentations performed.

Note the flat surfaces machined on the strain-gage locations. This is necessary for successfully strain-gage operation; it also helps to further reduce the strain differences between the two gages.



Figure 3.19 : Instrumentation of strain gages and connections to the transmitter.



Figure 3.20 : Using adhesives to cover the cables.

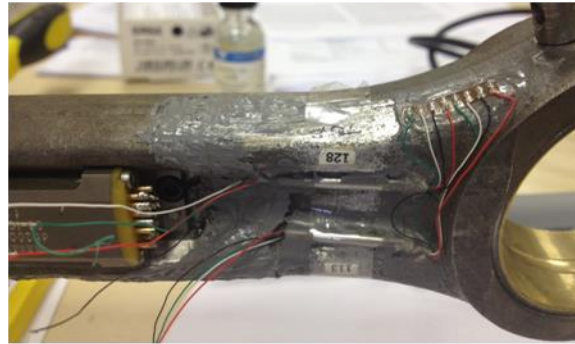


Figure 3.21 : Placement of pressure sensor bridges.

3.4 Connection Between Piston and Rod

A robust setup is required to carry the pressure sensor cables to the connecting rod. This setup needs to be durable under engine operation for several hours, and needs to be easily replaced in the case of damage and cable rupture. It needs to compensate the relative motion between the piston and the rod, together with withstanding the inertia forces due to piston movement.

Some applications pass the cables through the pin. In these applications, the pin has to be modified from “floating” to “fixed” type. This option was eliminated since it’ll jeopardise friction measurement.

Total of 4 designs were utilized and tested in dynamometer. The test consisted of 5 hours running at max torque and overspeed condition.

Design #1 is “Bowden Cable” design and was applied by Coventry University, UK. The design contains a heat threaded spring placed between the piston and rod. Brackets are used to hold the spring in place. This design failed during engine testing where the spring was taken apart from the fixing points. Figure 3.22 shows the condition of the system after the test.



Figure 3.22 : The condition of Bowden cable design after testing.

It was decided to use “Teflon” tubes for carrying the cables; due to its flexibility, strength and temperature resistance. Small holder parts were to be used to fix the tube onto the piston and rod. Several routing options were considered and checked in CAD. The aim is to provide the required flexibility (due to relative motion between piston and rod) with minimum cable motion. Figure 3.23 shows some of these routings.

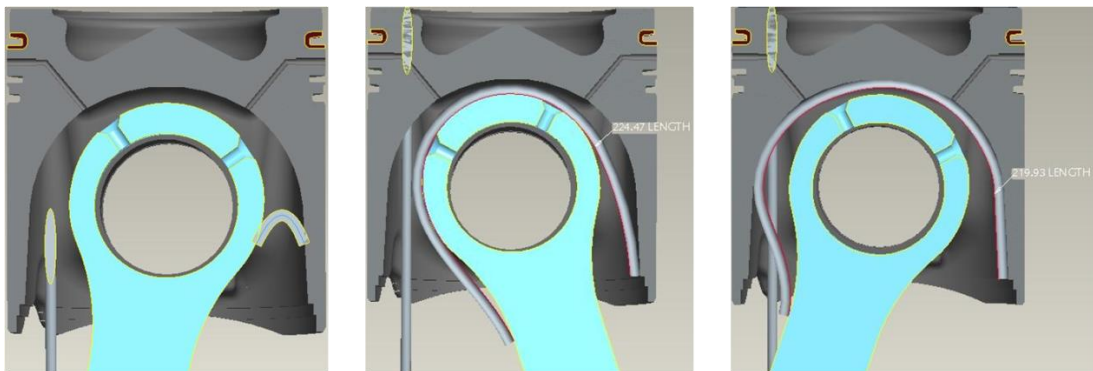


Figure 3.23 : Teflon tube routing study done at CAD.

The routing was further studied in laboratory using actual engine components. It was decided to fix the tube on the side of the connecting rod small end, make it travel around the small end to reach to the reference pad below the piston pin bore. With this routing, the tube is given enough flexibility with the least amount of movement and bending stress. Care was taken to avoid contact with the piston cooling jet part and the oil jet that comes out of this part (which is used to cool the piston).

The following designs use this Teflon cable and this routing with different fixing options. Design #2 contains a cylindrical holder with single screw to hold the Teflon

tube. The holders are glued to piston and rod using JB Weld. Figure 3.24 and Figure 3.25 show the pictures of this system (the blue tube is not Teflon but a plastic tube for representation).

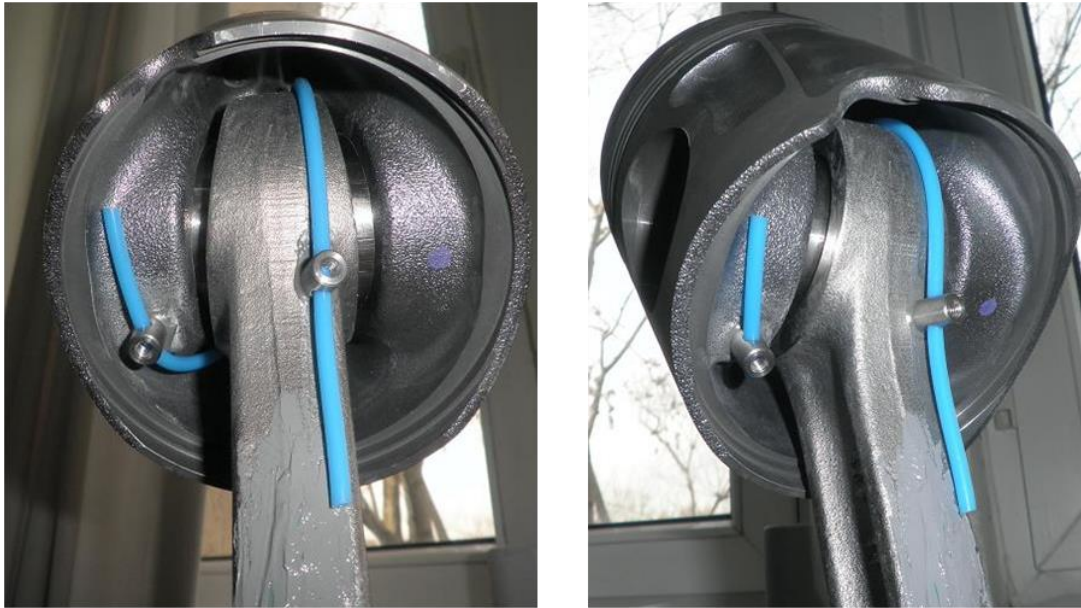


Figure 3.24 : Setup for design #2.

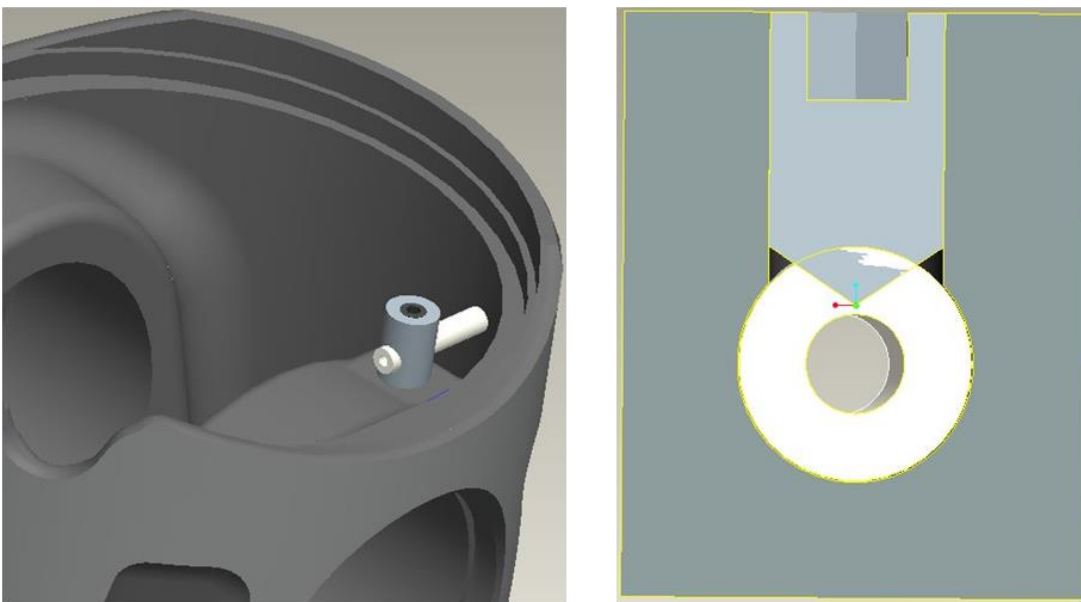


Figure 3.25 : Design #2 CAD pictures with cross-section of the cylindrical holder.

This design got completely destroyed during engine testing. The problem was found to be due to the fixation of the cylindrical holders on the piston and rod. The differences between the expansion coefficients of the engine parts, JB Weld and holder parts are thought to cause the holder to snap off from the piston and rod. After this trial, it was decided to switch to bolted systems.

Design #3 contains the same cylindrical holder, but with an additional screw to hold the Teflon tube fixed to the piston and rod. Instead of gluing, the holder is screwed onto piston & connecting rod. This screw size is carefully determined to not to cause damage to piston or connecting rod.

For the drilling operation, it is relatively easy to find a suitable location on the piston due to the extra material on the reference pad; but since the connecting rod is already heavily modified with strain-gages and instrumentations, the only place to fix this is the small-end. Using the FEA analysis, the best location with the least stress is found to be the region just above the radius joining the small end with the shank. It's still a highly stressed region, hence appropriate dynamometer testing with a dummy engine was performed to make sure that the rod will survive during the measurement. The test was passed successfully.

For the piston, the only consideration was to make sure that the bolt did not reach below the minimum thickness of the pin bore.

CAD checks were performed to make sure there's no clash with neighbouring components. Holder part was getting too close to the crankshaft counterweight, so it was decided to machine the counterweight by 4 mm to prevent any clash.

Figure 3.26 and Figure 3.27 show the pictures of the system.

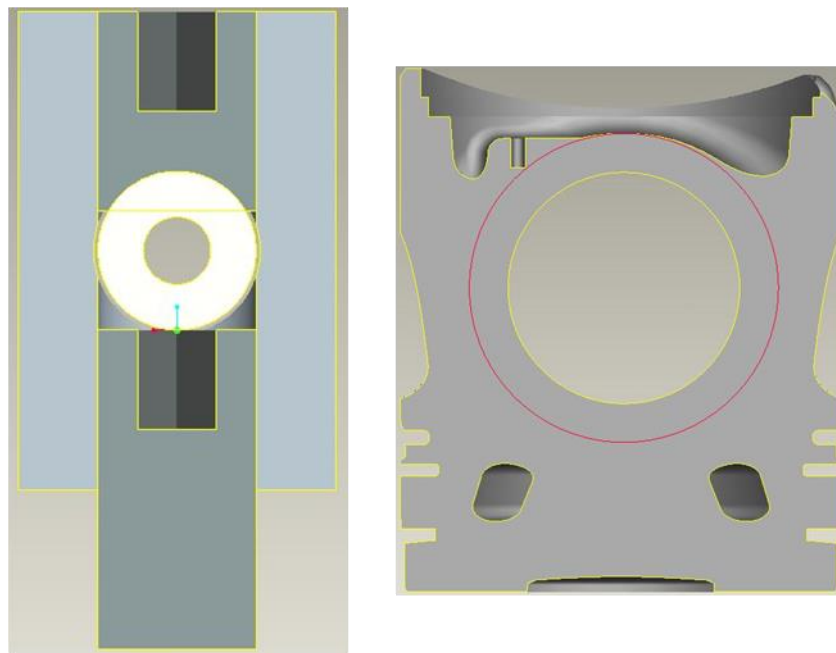


Figure 3.26 : Design #3: Cross section of the holder and piston drilling.

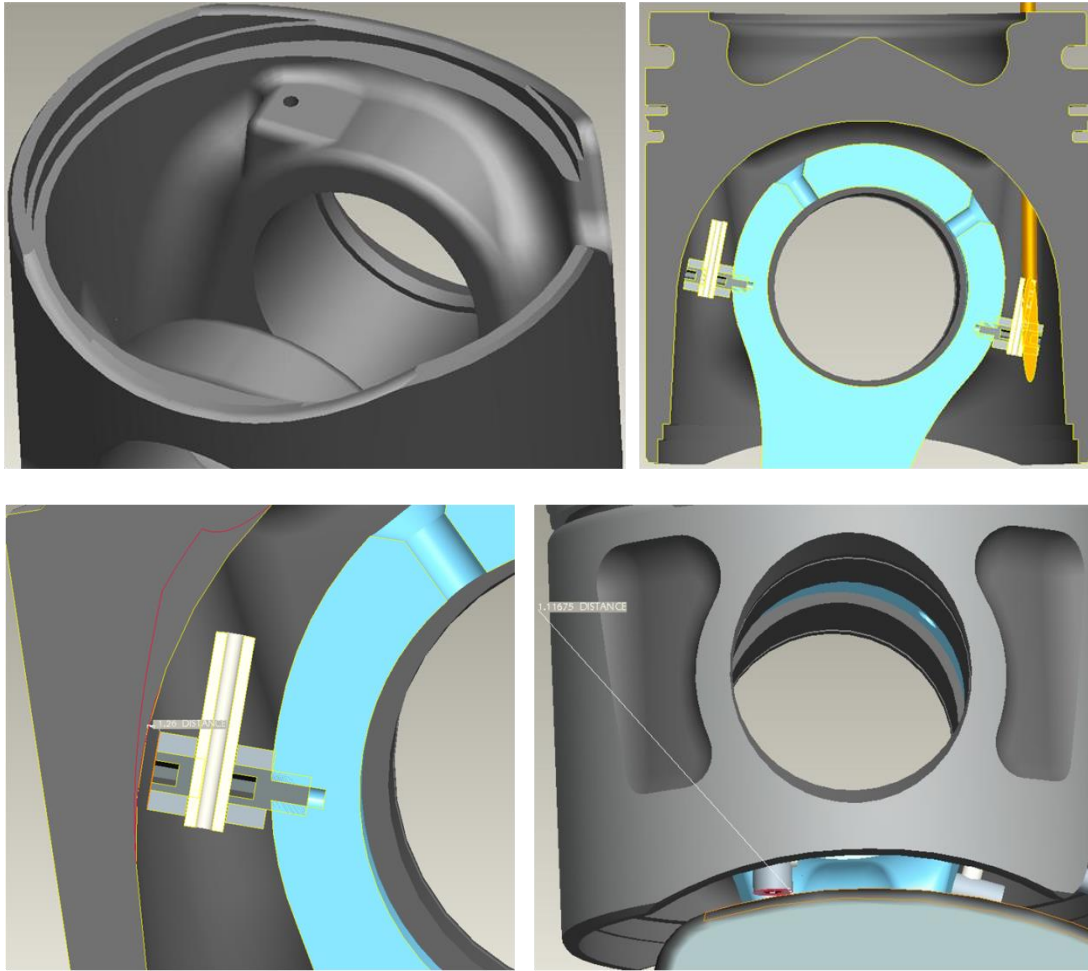


Figure 3.27 : CAD pictures and packaging checks for design #3.

Following engine dynamometer testing, the Teflon tube was found to be scratched and the sensor cables inside the tube were exposed. This was found to be due to an error during installation. Since one screw is used to hold the cylindrical holder in place, the threading of the holder and base part determines the angle of the holder part. This angle affects the Teflon tube orientation. The holder on the connecting rod must point directly upwards to allow the tube to travel around the small end. The tube will then bend slightly to meet with the holder on the piston, so the holder on the piston must be slightly angled towards the rod. In the first trial, the angles of the holders were incorrect and caused the tube to bend excessively. This caused the tube to get squeezed between the connecting rod and piston, hence the scratches on the tube. Nevertheless, the test proved that the setup is robust enough to hold the parts in place. Figure 3.28 shows the result of testing.

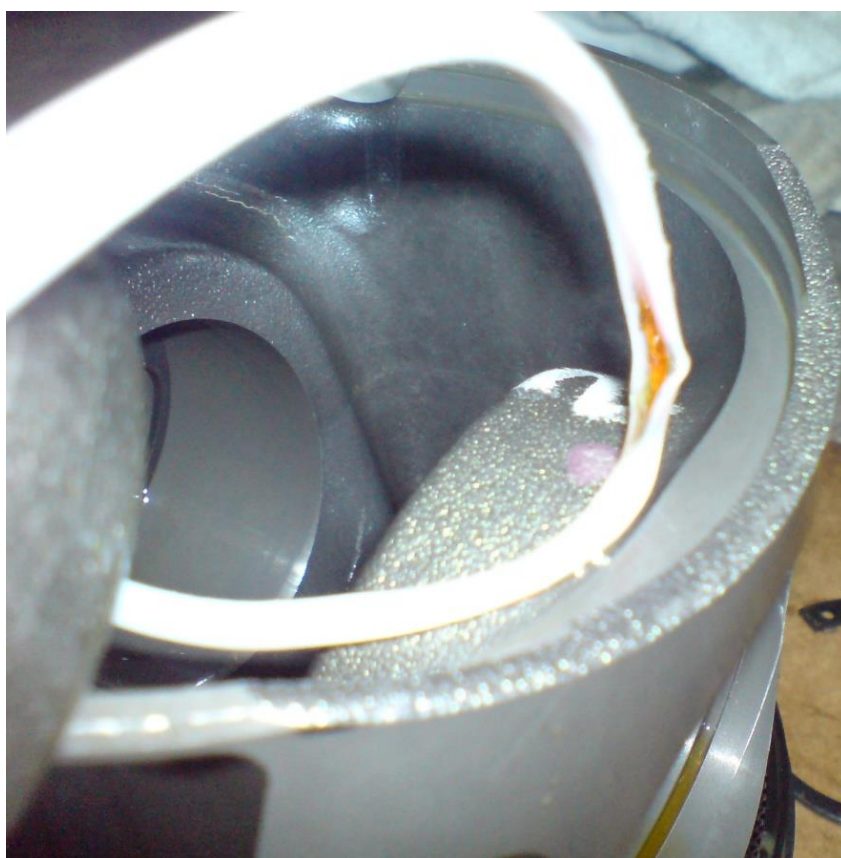


Figure 3.28 : Design #3: Result of dynamometer testing.

Design #4 contains a sheet metal holder with two bolts each for piston and rod. Due to its design, the problem observed with cylindrical holder does not apply. Design passed engine testing without any damage, however it was decided not to use this design because of possible damage to piston or connecting rod due to usage of two bolts. Engine testing revealed that as long as the routing of the tube is done correctly and holder parts stay intact, no problems are observed. The holder parts in design #3 can easily be modified to obtain correct angle. Moreover, design #4 holds the tube in position by applying pressure and relying on the friction between the tube and the sheet metal. Due to the pressure applied to the tube, the cables are squeezed, which can cause damage to the measurement. On the other hand, design #3 uses shape bonding between the screw and the tube, which allows the tube to be fixed without applying pressure onto the cables. Because of these reasons, the decision was to use design #3.

The layout for design #4 is given in Figure 3.29. Post-test results are given in Figure 3.30.

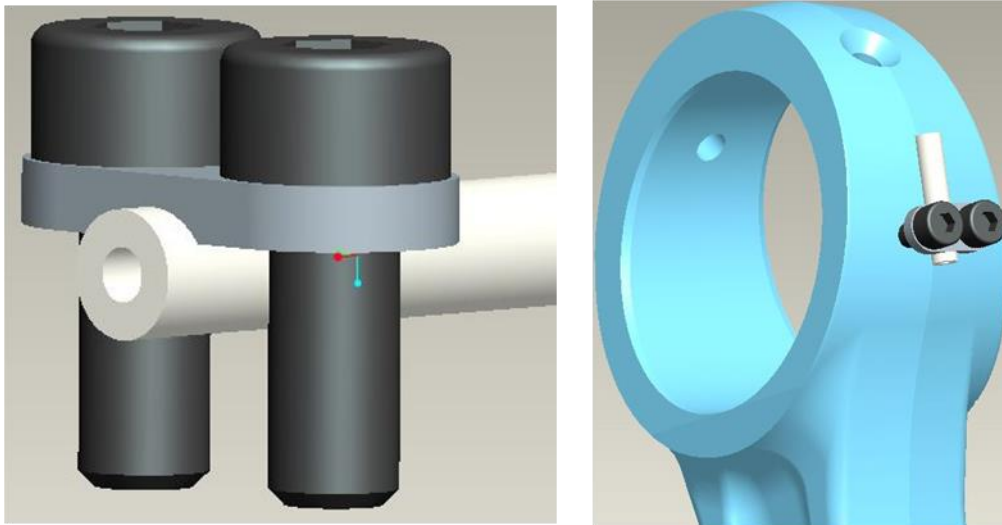


Figure 3.29 : Design #4: CAD Pictures.

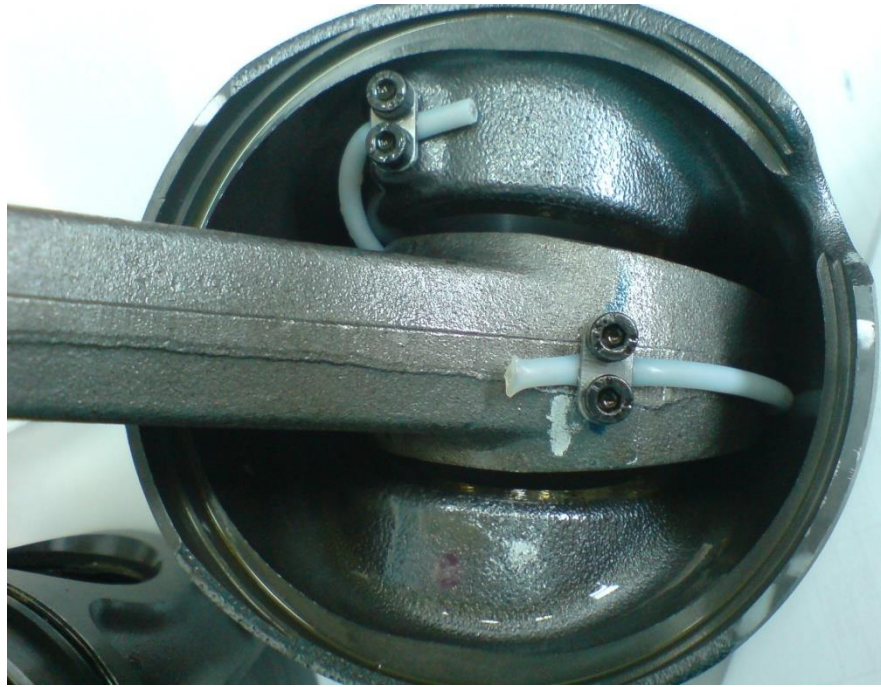


Figure 3.30 : Design #4: Pictures after the dynamometer test.

3.5 Engine Block Modifications and Instrumentations

3.5.1 Modifications for pegging sensor

In order to correct the measurements of the in-cylinder relative pressure sensor, an absolute pressure sensor was implemented near BDC. This technique is called

pegging. Same Kulite pressure sensor that is used for piston 2nd land pressure measurement was selected.

In order to insert an absolute pressure transducer in the cylinder towards the bottom of the stroke, it was needed to drill a hole in the cylinder block. This hole would pass through the water jacket, therefore an adaptor system is required to prevent coolant water and cylinder gas interaction and coolant leakage to the outside of the engine. In addition, the adaptor should gently place the pressure sensor close to the cylinder and hold it in its place. For this task, the adaptor system in Figure 3.31 was designed.

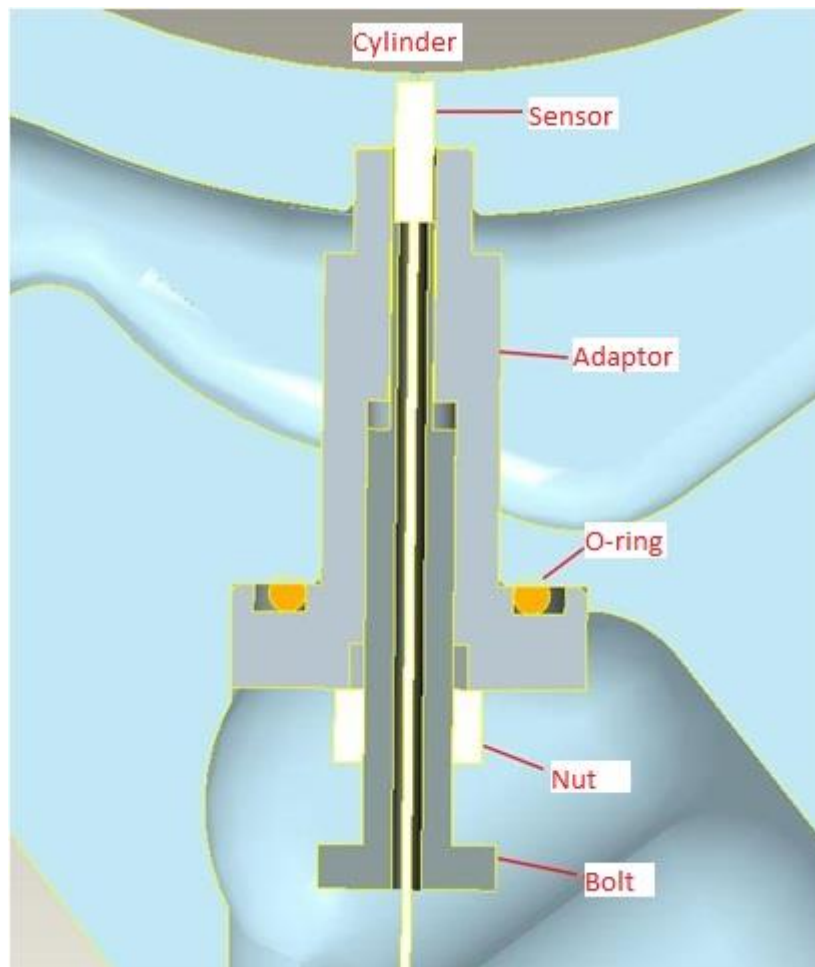


Figure 3.31 : CAD picture of pegging adaptor system.

The adaptor is bolted inside the block and covered with sealant (Loctite was used) to avoid cylinder gas and coolant water interaction. The o-ring prevents coolant water leakage to the outside. The bolt is inserted inside the adaptor to push and hold the sensor in place. This bolt has reverse teeth to avoid loosening of the adaptor. The nut avoids loosening of the bolt.

The opening to the cylinder needed to be very small. This was achieved using the tip of the drill. Drilling operation was stopped immediately after the hole was visible. The reason for this is to make as small a hole in the cylinder block as possible, to prevent malfunctioning of the piston and piston rings.

Figure 3.32 shows the location of the adaptor in cylinder block.

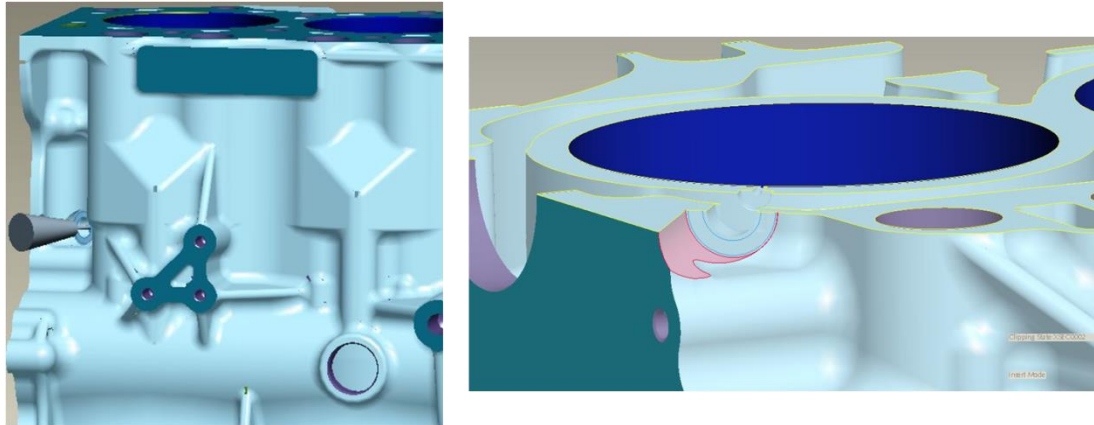


Figure 3.32 : CAD pictures of pegging adaptor installations in cylinder block.

This system was successfully tested in a dummy engine using a steel cylinder as representative of the sensor. No leakage issues were observed and the sensor was able to be kept in place. The pictures of the system are shown in Figure 3.33 and Figure 3.34.

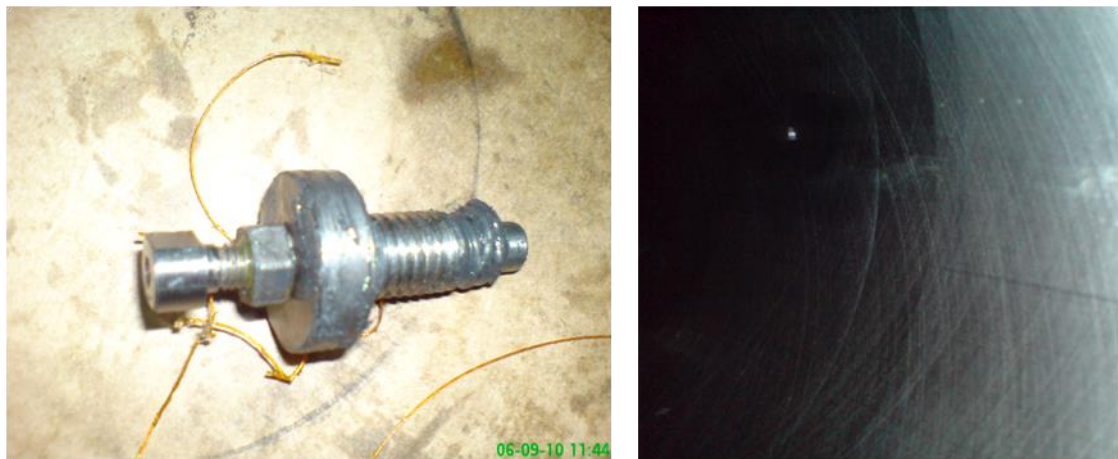


Figure 3.33 : Picture of the adaptor after testing and the drill in the cylinder block.

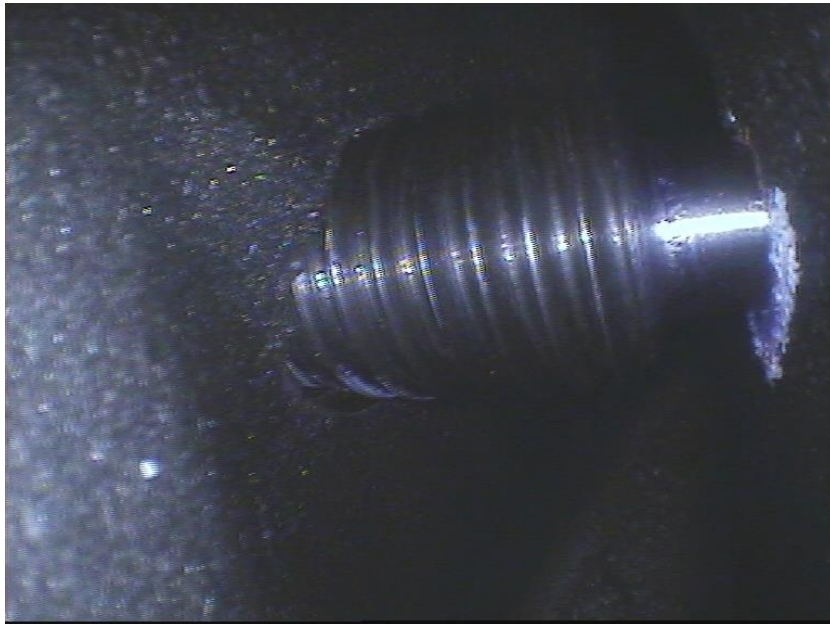
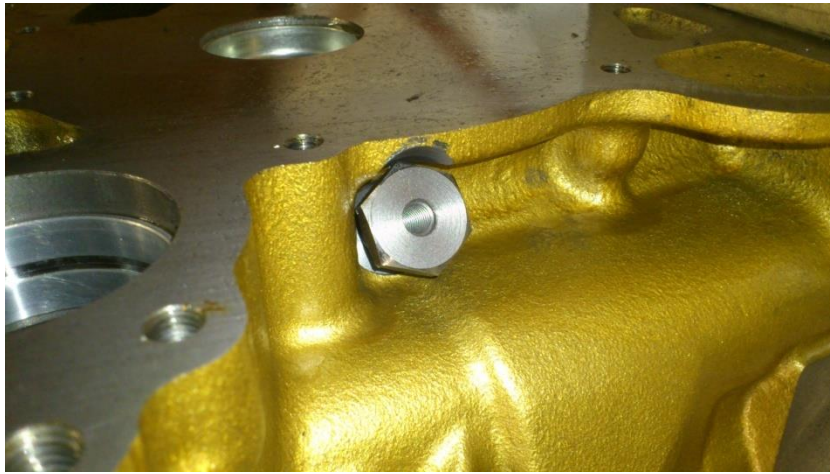


Figure 3.34 : Pictures showing block drilling and the adaptor inside the block.

3.5.2 Engine antenna placement

Antennas are required to be placed inside the cylinder block in order to pick-up the transmission signals sent by the transmitter on the connecting rod. These antennas need to follow the connecting rod and be as close as 2-3 mm to the transmitting antennas on the connecting rod. This is due to the low electrical power consumption requirement, which also helps in reducing the noise.

Figure 3.35 displays this recommendation.

The green coloured regions show the layout of antenna brackets. These brackets hold the antennas in the appropriate position. They are to be produced from glass fibre, either machined from a solid block or cut out of thin plates and glued together.

The block surfaces need to be smoothened before fitting the antenna brackets. Two methods are available for holding the brackets in place; using spot welding or using a bolted system.

After the antenna is placed on the block, matching resistors need to be connected between the antenna and electrical ground. This is called antenna termination and is used to terminate the signals that travel back from the end of the antenna, which would otherwise cause interference with the received signal and thus increase noise. In this application, engine block is considered as the electrical ground; the resistors need to be placed between the antenna and the block.

The antenna system is to be covered using JB Weld to provide robustness against engine vibration and protection against external factors.

Final step is to connect the BNC jacks to connect the antennas with the receiver equipment.

These steps are shown in Figure 3.36 and Figure 3.37.

The antenna modifications have been done previously by Datatel, however it's not possible to use that block, therefore they need to be re-done in a new block.

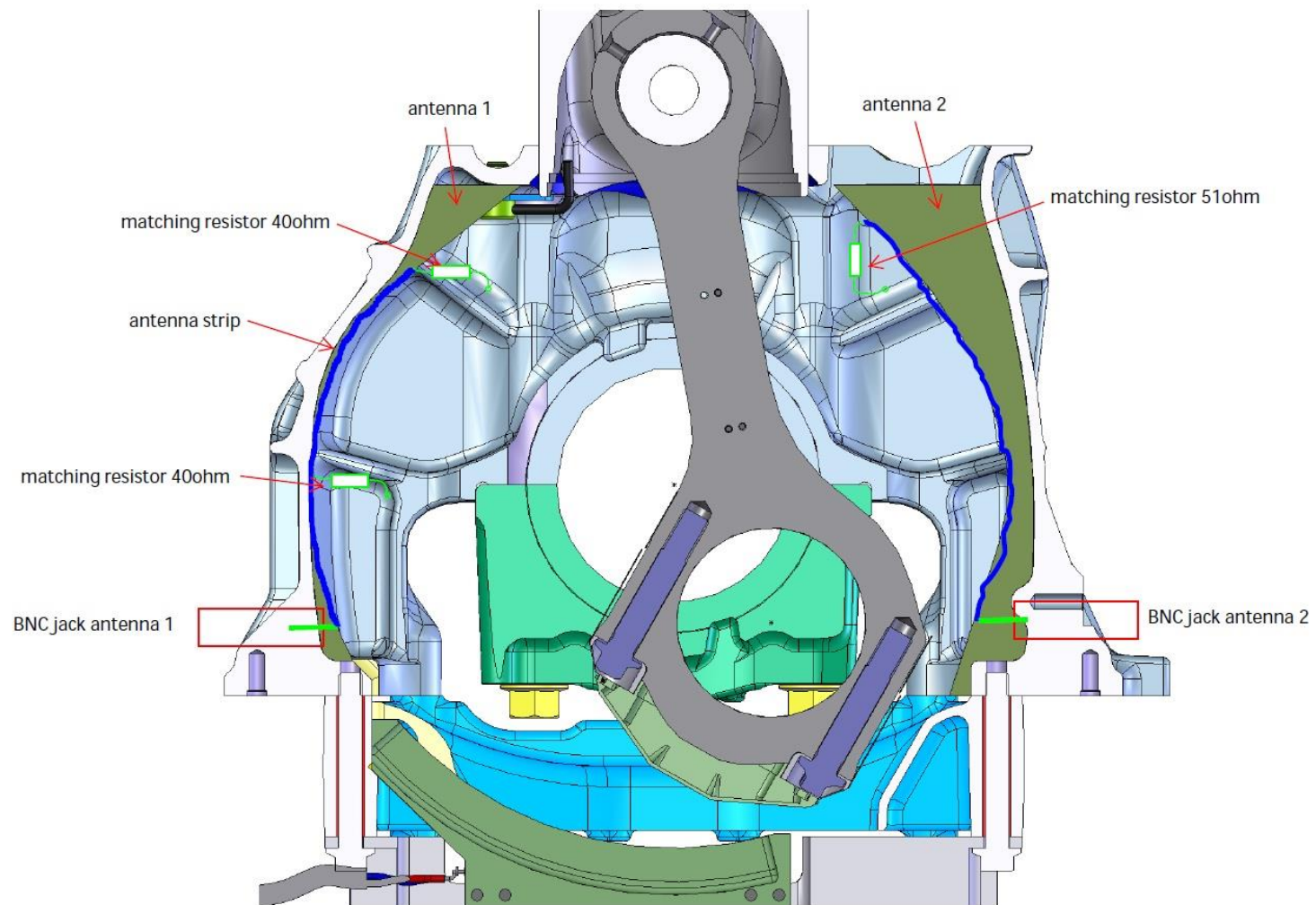


Figure 3.35 : CAD representation showing the block antenna requirements.



Figure 3.36 : Engine block surface after smoothing and antenna brackets.

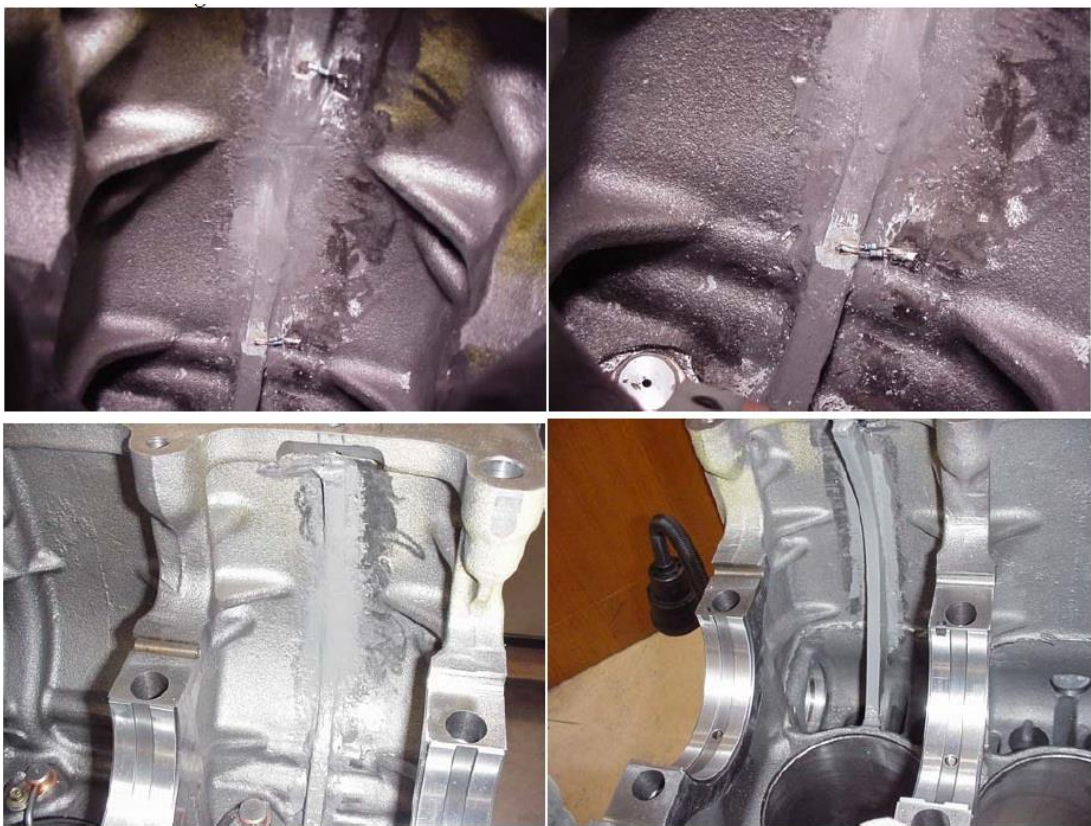


Figure 3.37 : Pictures showing antenna and resistor installation.

3.5.3 Block and ladderframe modifications for stator coil

Ladderframe is a stiffener part placed under the engine block to the block skirt. The objective is to increase the stiffness of the block to reduce the distortions in crank bores and cylinders.

The last modifications to be done on the engine are for the placement of the stator coil. Datatel installed brackets onto the stator coil for easy installation on the ladderframe using original ladderframe bolts. Small sections on the block and

ladderframe are machined to prevent clash of the rotor coil on the connecting rod. Final modifications were made at the bottom of the engine block to provide the pathway for stator coil cables. These modifications do not affect the functions of the parts. These are shown in Figure 3.38, Figure 3.39 and Figure 3.40.

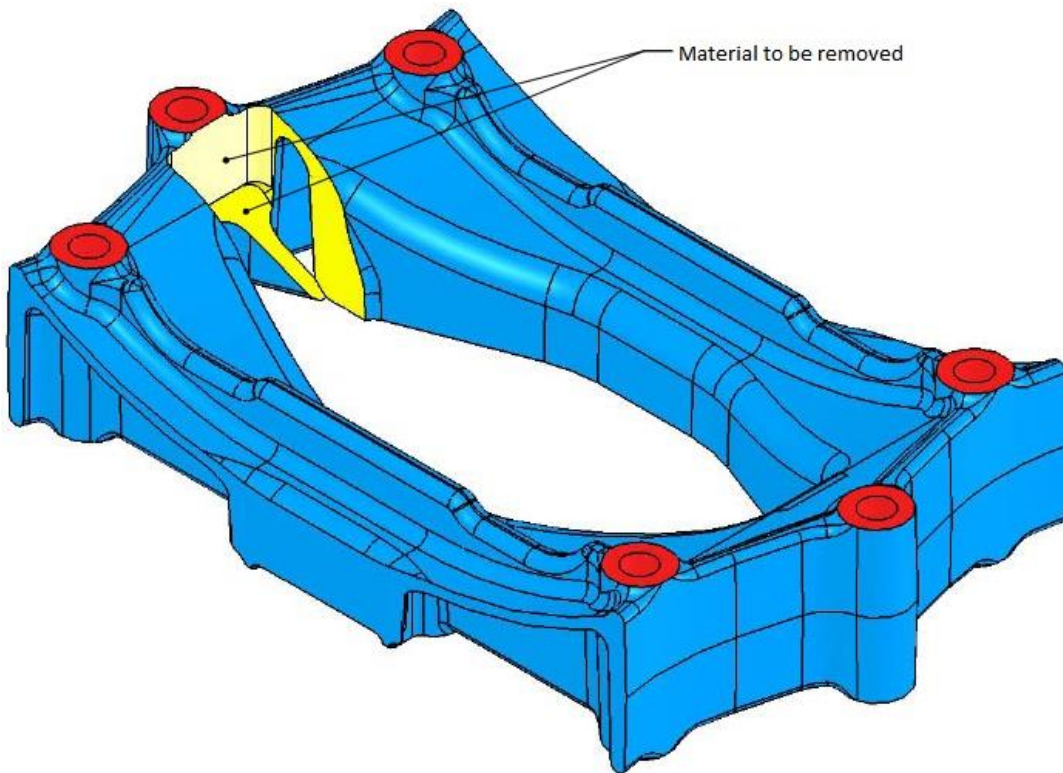


Figure 3.38 : Ladderframe modifications.



Figure 3.39 : Small machining at block underside to take out stator coil cables.

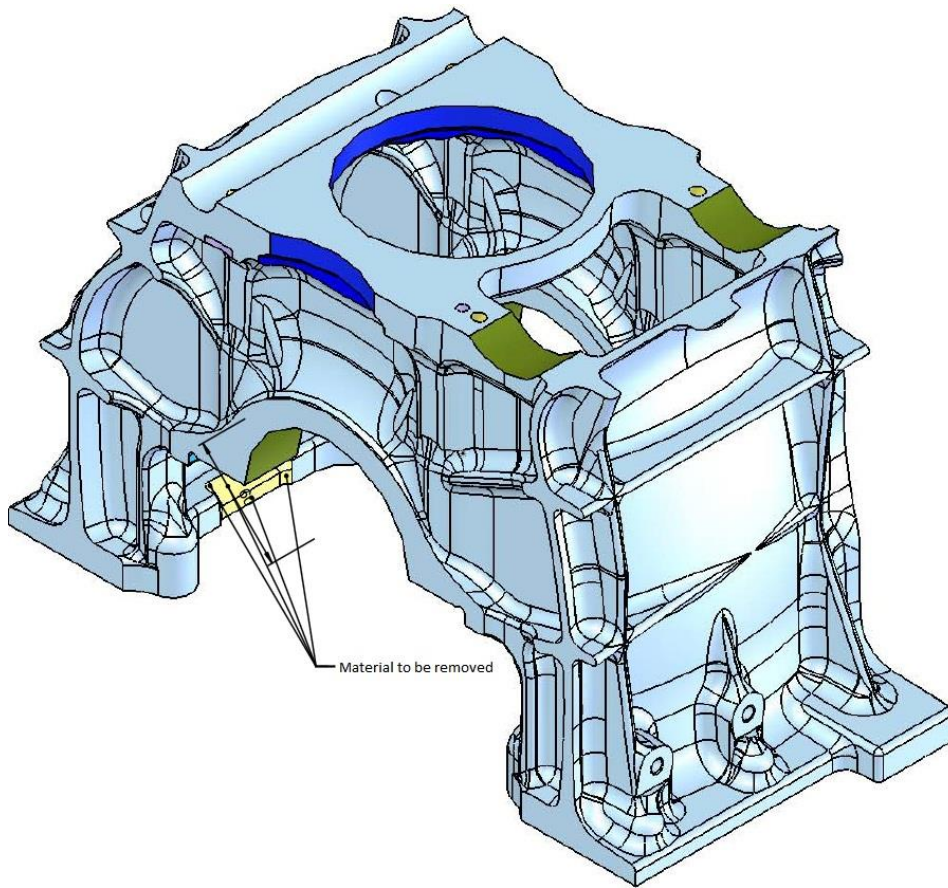


Figure 3.40 : Engine block modifications.

3.6 Calibration

Calibration was performed on all 4 channels that data collection will be held:

- Strain-gage data from the connecting rod
- Two pressure transducers in the piston
- Thermocouple on the transmitter: This sensor is factory installed inside the transmitter and will be used to make sure that transmitter temperature does not exceed recommended limit.

The output from calibration must reveal the output voltages at the receiver corresponding to sensory input. Other signals that are generated in other parts of the system are not of concern.

The objective is to represent the engine conditions as fully as possible. In addition, the accuracy of the calibration is very important since it has direct influence over the accuracy of the results.

Before the calibration work was started, the complete telemetry system setup was tested in laboratory. The parts were not in the engine, but it was possible to activate the system by placing the stator coil close to the rotor coil. The stator coil was connected to the receiver, data acquisition and inductive power generator units, hence it was possible to activate the telemetry system and verify that there are no problems.

3.6.1 Strain-gage calibration

For this study, it was decided to place the instrumented connecting rod on a rig that can generate a known amount of force. Two data sets were recorded, the force and the output voltage at the receiver. With this method, it is possible to find the relation between rod force and output voltage. All the other effects are being neglected, including the strain on the connecting rod, the voltage across the strain-gage bridge, and so on. During engine testing, this will allow the receiver output to be immediately converted into connecting rod force and insert in the calculations for piston friction.

Istanbul Technical University ROTAM (Rotorcraft Design) facility provided a MTS testing rig that can apply both constant and alternating load up to 100 kN. This value is less than the maximum compressive force applied to the connecting rod during engine operation, which is 180kN. This is not thought to be an issue from calibration point of view, since the connecting rod is always kept at the elastic region where there's a linear relation between stress and strain. Thus the calibration curve can easily be extended to cover the whole range.

Maximum tension force applied to the rod is 20 kN. The rig is able to produce more force so care was taken not to exceed this value.

The rig contains a gripping mechanism with two clamps for holding parts in place. Adaptors were designed and produced specially for this rig. Engine components were used wherever possible; including piston, pin, circlips, block cap and bolts. Figure 3.41 shows the adaptor designs.

Even when the machine is not applying any force, the strain-gages will be reading a small strain due to the weight of the components above the strain-gage, including the weight of the adaptors and the small end section in the rod. These were calculated and subtracted from the results to obtain good accuracy.

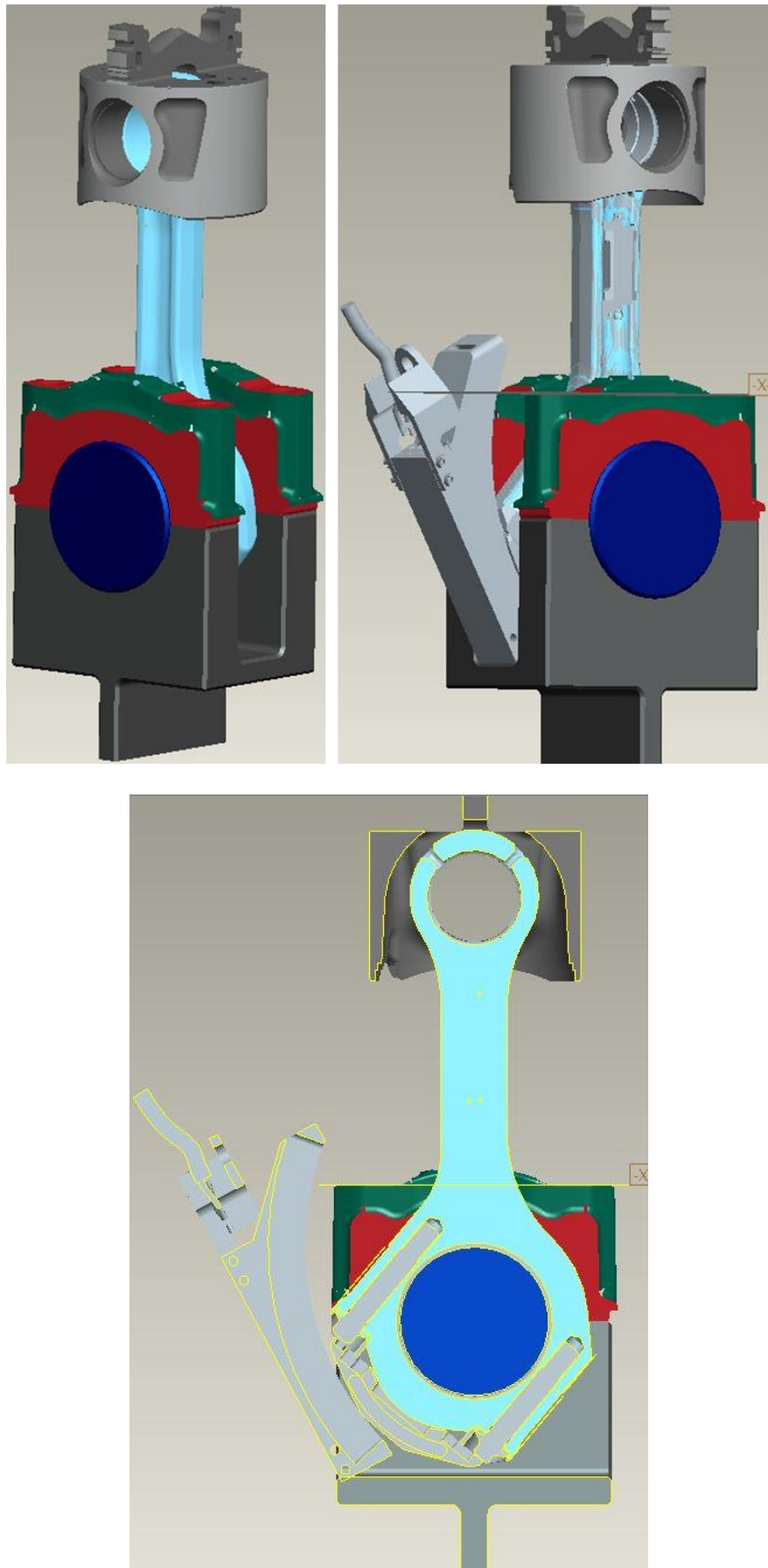


Figure 3.41 : CAD screens showing the adaptor design.

The adaptor design does not allow for full opposition of the stator and rotor coils, however it was tested in laboratory and confirmed that enough voltage is generated in the transmitter for successful operation.

The data acquisition system for collecting data from the receiver was provided by the ROTAM facility. BNC cable was used to connect the receiver and the data acquisition system. The system has the capability to sample with 500 Hz frequency.

The gripping area on the piston bracket was checked to make sure that it won't break or slip. This study is shown in Figure 3.42.

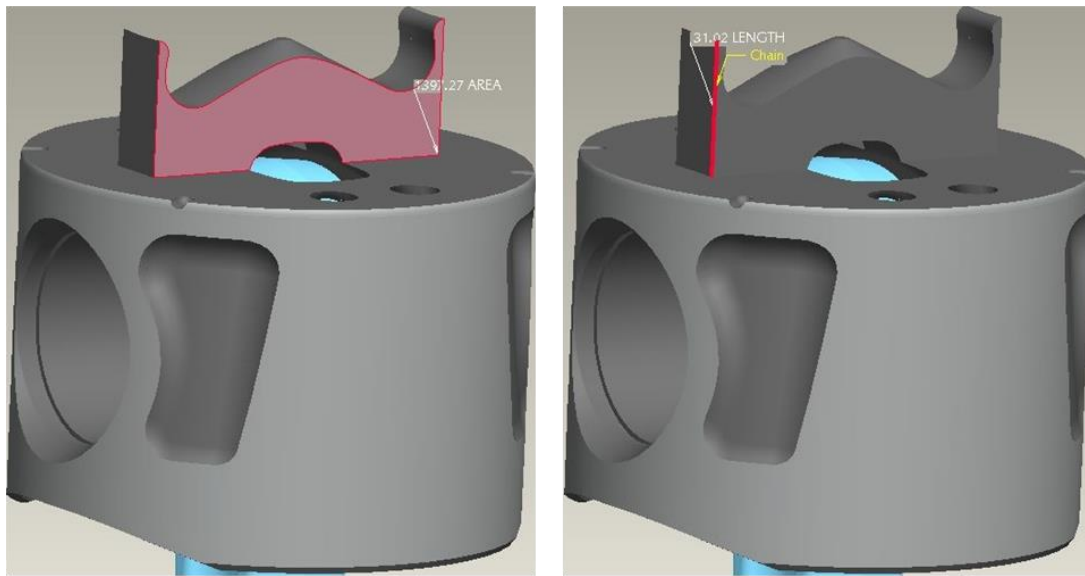


Figure 3.42 : CAD checks for gripping area in the piston.

A trial run has been performed with a dummy connecting rod to make sure that the adaptors are suitable for this usage. No problems were found for 100 kN compression force and 20 kN tension force. Figure 3.43 shows the setup placed in the rig.

First calibration trial resulted in very noisy results. The root cause was found to be a faulty solder on the connecting rod coil. This setback showed the importance of careful instrumentation.

The second trial has been completed successfully. The tuning knobs at the receiver were used to make small adjustments on the gain and location for zero voltage. It was found that even at maximum gain, the output voltage limit of $\pm 10V$ cannot be reached, so the gain was maximized for highest signal ratio.



Figure 3.43 : Pictures showing the setup placed in the rig.

The zero adjustment was performed to obtain 0V for 0N of force. The force range has been swept in both directions. Figure 3.44 shows the calibration curves. Note the hysteresis curve shown on the second figure. Since steel shows very little elastic hysteresis, it can be concluded that this is due to the time lag caused by the telemetry system.

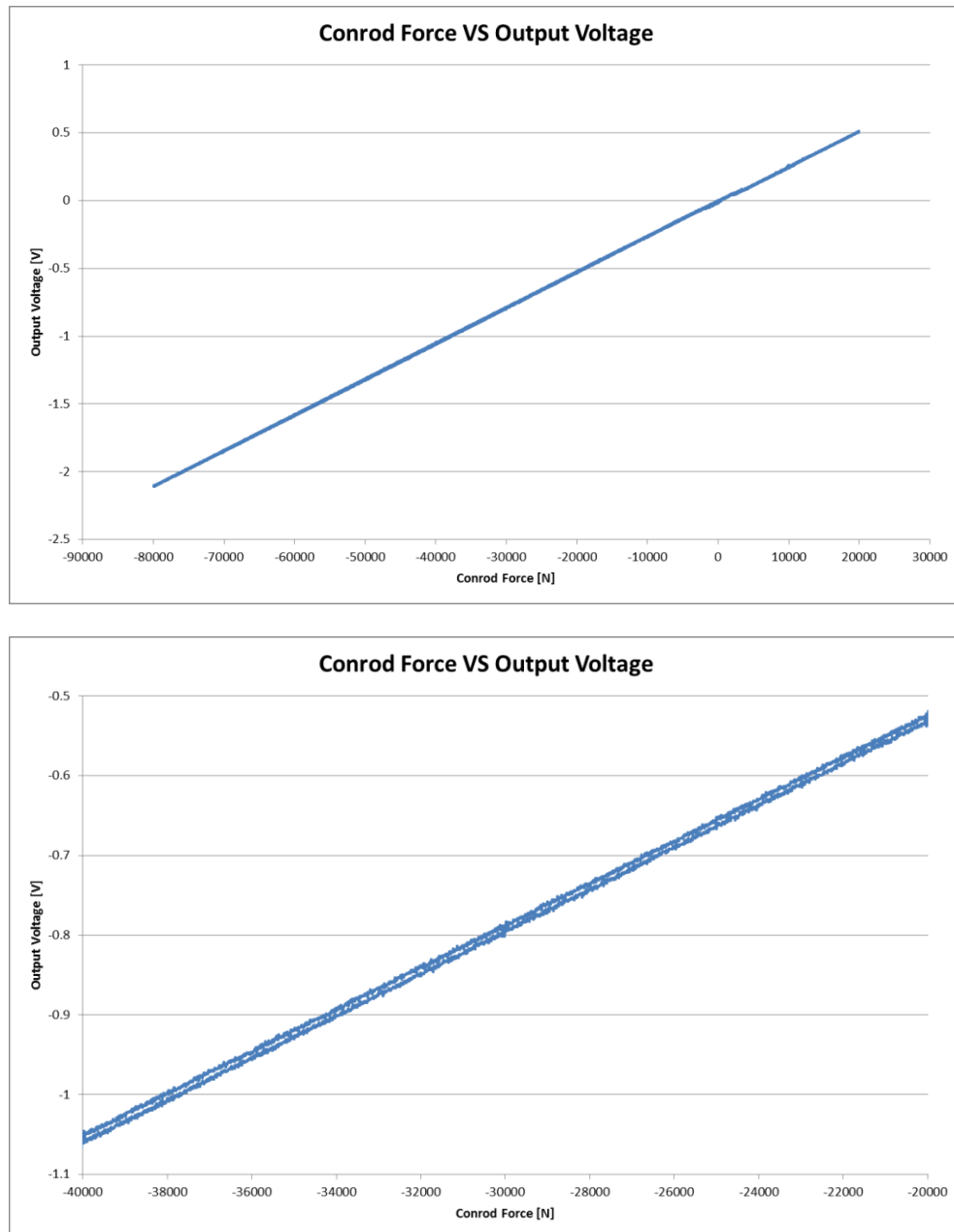


Figure 3.44 : Calibration curves for strain-gage.

3.6.2 Pressure sensors calibration

Similar to the strain-gage calibration, the ideal scenario for pressure sensor calibration is to connect the pressure sensors to the transmitter, apply pressure and

measure the output at the receiver. However due to the absence of a device that could apply controlled pressure to the piston, it was decided to externally input the pressure sensor information to the transmitter. This is an acceptable method as well, since all of the pressure sensors are factory calibrated and the exact relation between pressure and sensor voltage is given.

For this task, three options are recommended by Datatel:

- Using Shunt Calibration Box: This option requires a shunt calibration tool, which was not able to be found.
- Using Voltage Source: The voltage source will be used to simulate sensor output voltage, which is a very small voltage. This option requires a very high precision voltage source and relies heavily on the voltage that is supplied. Small errors in the voltage result in bigger error in the output voltage. Due to this high sensitivity to voltage, this option was not selected.
- Using Function Generator: This option requires AC signal to be generated and entered into the transmitter using an external electrical circuit. This option was selected since the National Instruments data acquisition device available in ITU was able to generate the required signal.

Accuracy of the National Instruments “NI USB-6259 BNC” data acquisition device was checked against the full sensory input and it was decided to be suitable for use. Table 3.1 shows the accuracy values.

Table 3.1: Accuracy of the NI DAQ device

Sensor Calibration			
Input Value (V)	Absolute Accuracy (μ V)	Absolute Accuracy @ 70 Bar (bar)	Absolute Accuracy @ 70 Bar (%)
0.1	51.8	0.03626	99.95%
0.02	37.96	0.13286	99.81%
0.015	37.095	0.17311	99.75%

The selected option requires an electrical circuit to be constructed, using:

- A 100 k Ω resistor
- A 10 μ F capacitor

- Four resistors: Resistance to be calculated based on sensor properties.

Figure 3.45 shows the representation of the electrical circuit as shown in Datatel manual.

The circuit closely represents the Wheatstone bridge configuration. For this purpose, the four resistors are placed in the same configuration. The circuit is fed by the 5V supply from the telemetry system. One leg of the bridge is excited using the AC signal from the function generator. A capacitor is used to make sure that the excitation voltage only contains AC component and no DC component. This excitation causes an output voltage to be generated, which represents the sensor output voltage.

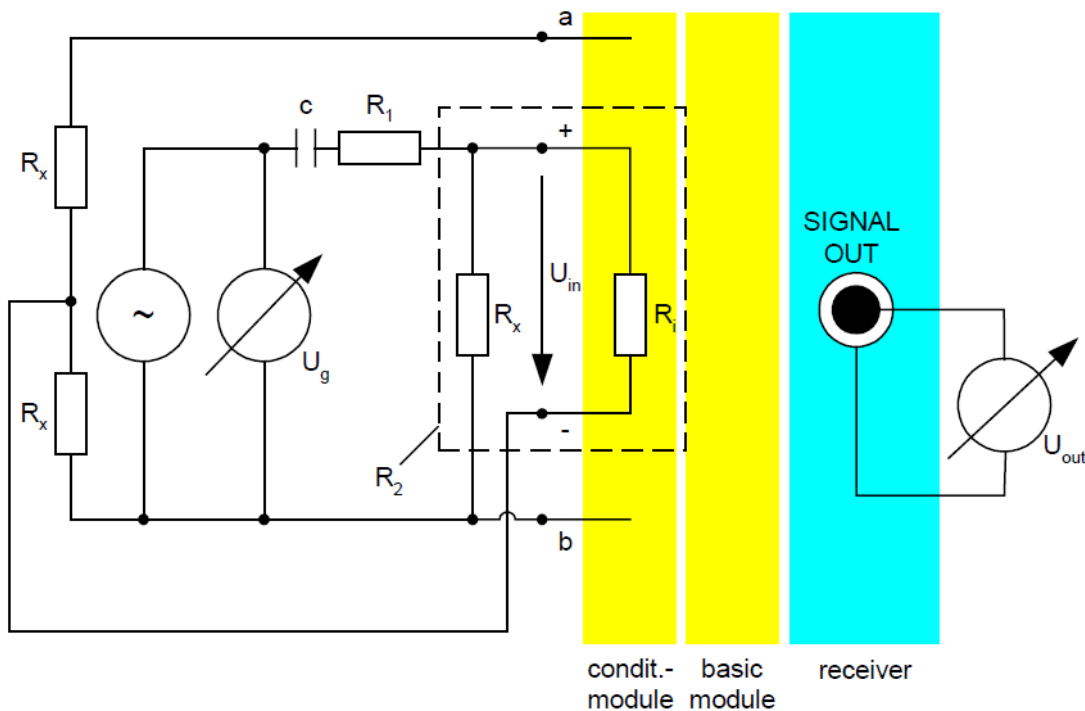


Figure 3.45 : The circuit shown in Datatel manual.

The electrical circuit has been constructed and simulated in PSpice computer program. First trial revealed an error in Datatel manuals, this was corrected. The simulation results showed that the desired input voltage is sent to the transmitter. Figure 3.46 shows the screenshots from the simulation.

Circuit board was implemented and connected to the transmitter. NI data acquisition device was used as both to send AC signal to the circuit and collect data from receiver. The picture of the implemented electrical circuit is given in Figure 3.47.

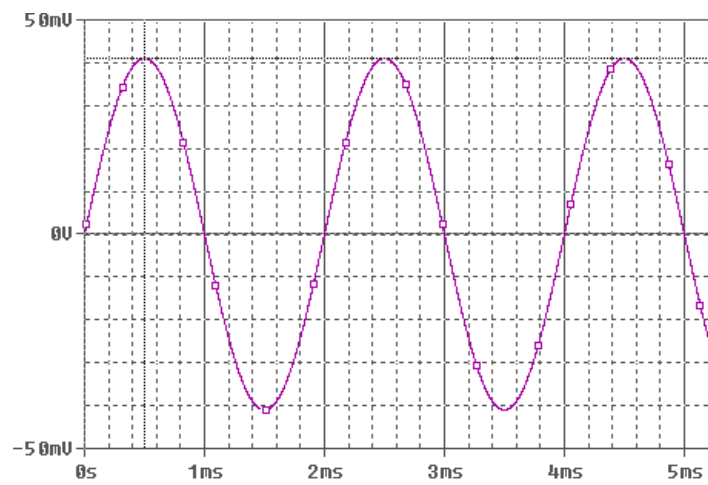
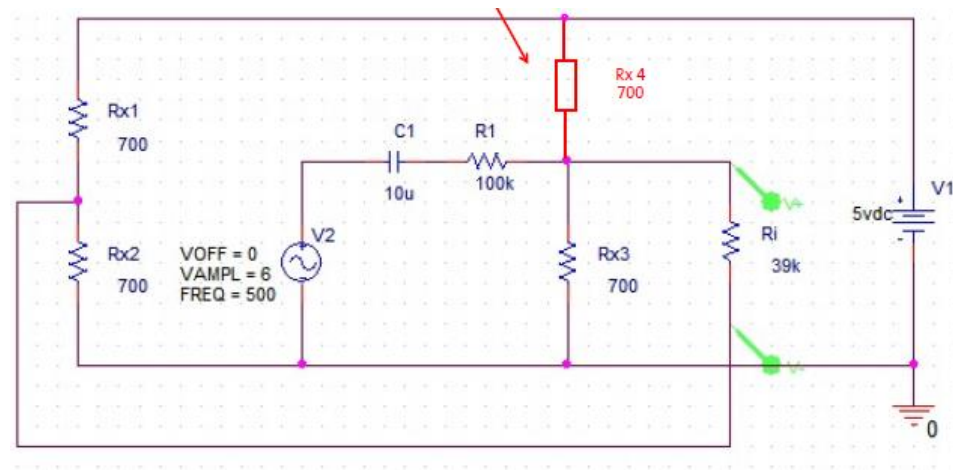


Figure 3.46 : The circuit constructed in PSpice and resulting input voltage.

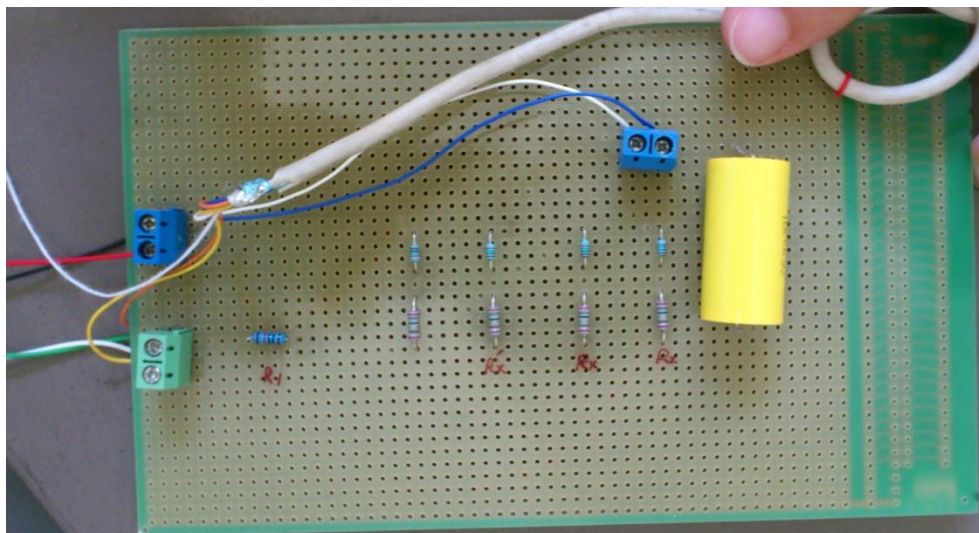


Figure 3.47 : Picture of the electrical circuit.

NI DAQ device was controlled using Lab View computer program. Figure 3.48 and Figure 3.49 show the screenshots from the program.

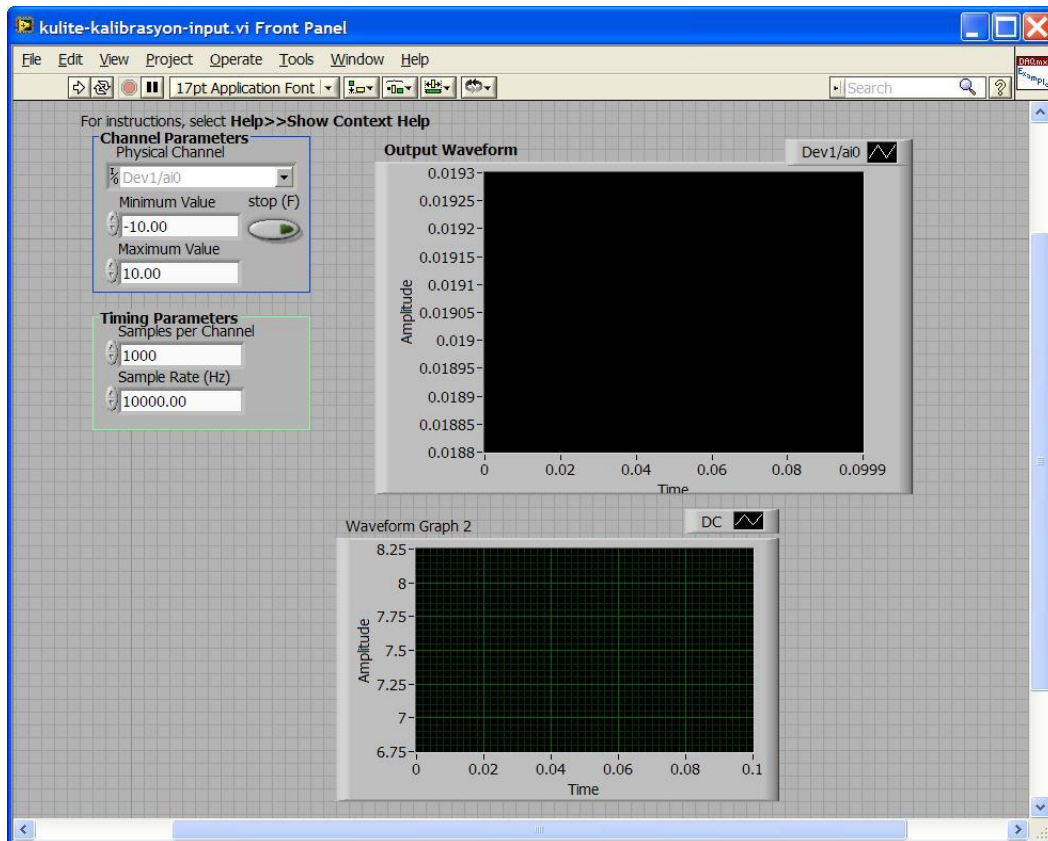


Figure 3.48 : Screenshot of LabView control panel.

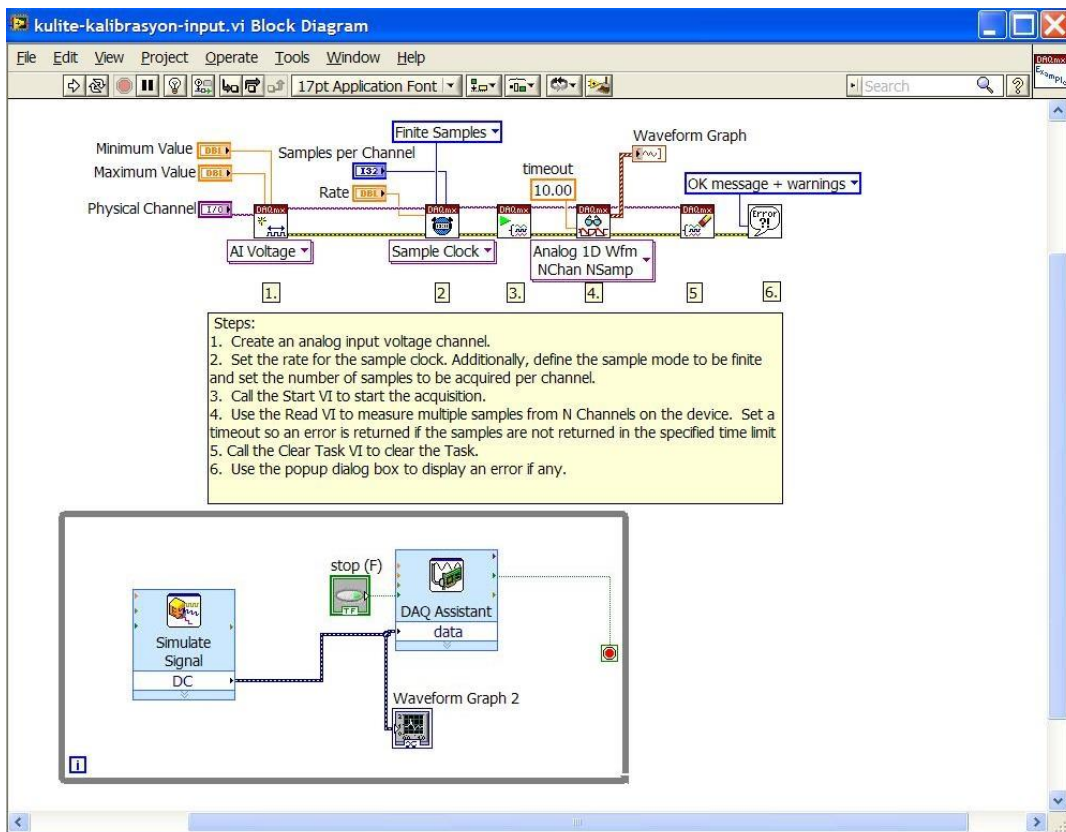


Figure 3.49 : Screenshot of LabView block diagrams.

Calibration curves have been obtained by running the LabView program, as shown in Figure 3.50.

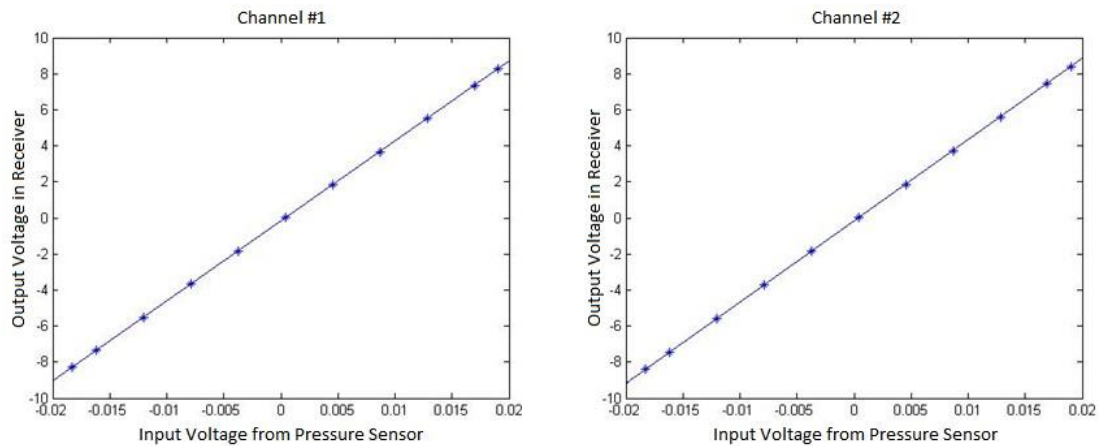


Figure 3.50 : Calibration curves for pressure sensors.

3.6.2.1 Telemetry system lag

As the sensory input is generated in the sensors, a certain amount of time has to pass before this signal can be read at the receiver, because of all of the systems that work in between. This lag is crucial for the measurements, since both strain-gage and pressure measurements must be matched to crank angle for the measurements to be beneficial. This is especially crucial for friction calculation where small delays can cause big errors in the friction.

During the pressure sensor calibration, since NI DAQ device was used to both send sensory input and collect data from the receiver, the lag between these two inputs were able to be measured. The lag was found to be approximately 0.6 ms, which corresponds to 1° crank angle during engine running at 2400 rpm. Figure 3.51 shows the screenshot for this study.

During engine testing, this value will be taken into account in order to match all of the measurements to the crank angle. It's possible to double check this correction using the pegging sensor. Right after the top ring passes through the pegging sensor in the cylinder, the pressure sensors inside the piston and the pegging sensor will read pressure from the same region. This information can also be used to verify telemetry system lag.

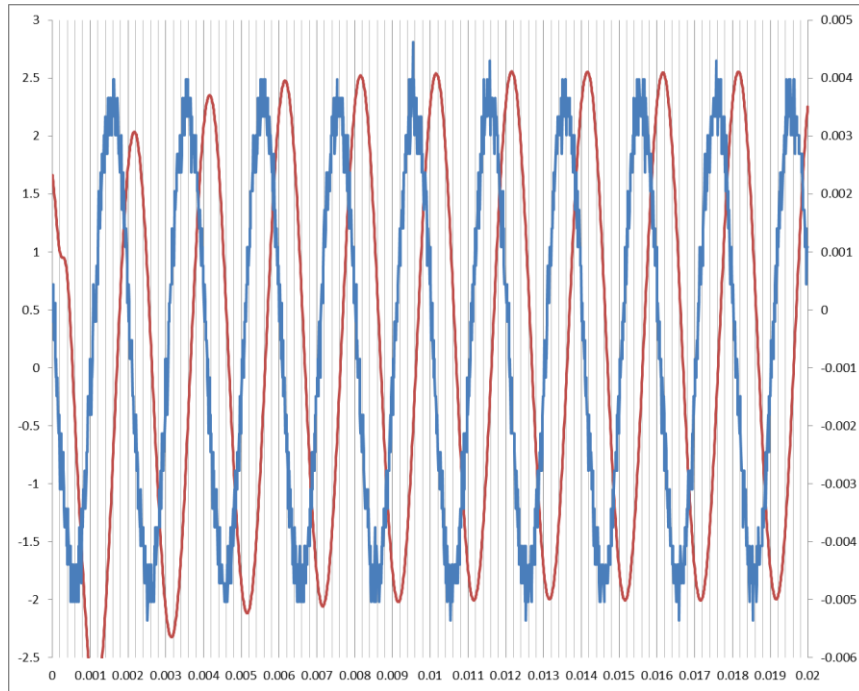


Figure 3.51 : Figure for telemetry system lag.

3.6.3 Thermocouple calibration

Final stage in calibration is to calibrate the thermocouple inside the transmitter, which will be used to control the temperature on the connecting rod to make sure that the electronics do not exceed the recommended temperature.

Thermocouple calibration was performed using an external electrical heater. The actual temperature was read using a calibrated thermocouple. The voltage from the receiver was read to obtain the calibration curve. Figure 3.52 shows this curve.

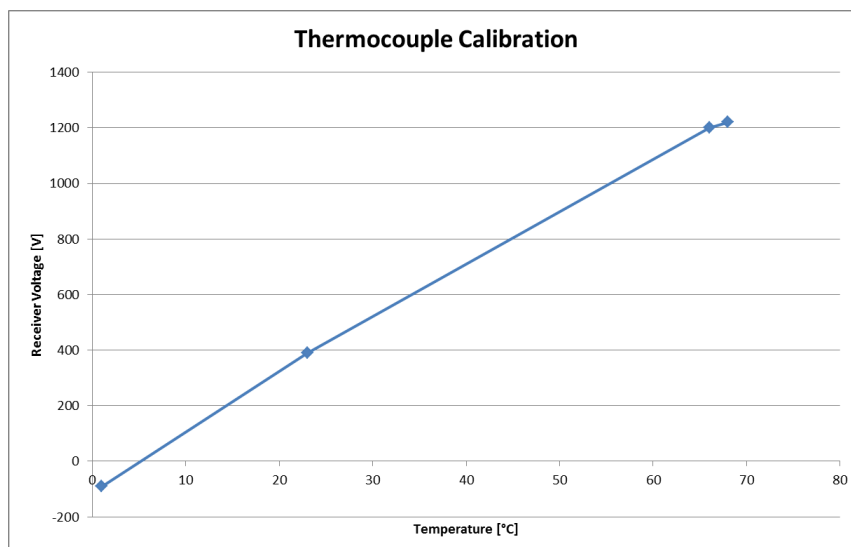


Figure 3.52 : Calibration curve for transmitter thermocouple.

4. SIMULATED RESULTS

In order to calculate piston assembly friction, MATLAB codes were generated using the calculations explained in section #2. However sensor data is not yet available to perform this calculation, because of delays in engine block instrumentation.

In order to check that the MATLAB codes are working properly, data from a previous study was used, where the piston assembly friction have been simulated in AVL Excite. This data was used to obtain connecting rod sensor data using another MATLAB code. This sensor data was then fed back into the original MATLAB program to check that the same friction values were found. These MATLAB codes are written in the appendix section.

This study was not performed for piston 2nd land pressure measurement, since the calculation is straightforward. The receiver output voltage will easily be converted to sensor voltage using the calibration curves, and then the sensor voltage will be converted to pressure using the calibration datasheets of each sensor.

Two sets of MATLAB codes were generated. The first two, as shown in Appendix A and B were used for symbolic formulation. The first code formulates the piston assembly friction force, assuming engine operation where strain-gage force is provided. The second code formulates the strain-gage force, which is to be used in this study for reverse calculation. The results of these formulations are shown in Appendix C. These formulae will be used after engine data is available.

The second set of codes that are shown in Appendix D and E were used for numeric calculations used in this study. The first code in Appendix D performs the reverse calculation to obtain strain-gage force, whereas the second code is used to regenerate the friction force. These calculations are explained in the next sections.

4.1 Inputs

4.1.1 Constants

The inputs to the program consist of constants and variables. The variables contain data that vary according to time / crank angle. These are limited to the cylinder pressure, strain-gage force, friction force and crank angle. All of these values were entered with respect to time.

The values for the constants are shown below. All lengths are entered in meters, masses are in kilograms and pressures are in Pascals.

- R (crank throw): 0.072 m
- L (connecting rod length): 0.233 m
- s (σ : piston pin offset): 0.0006 m
- mP (piston assembly mass including rod small end): 3.167 kg
- mR (rod shank mass up to the strain-gages): 0.226 kg
- r1 & r2 (distance from small end centre to beginning of rod shank and strain-gage locations respectively): 0.035 m & 0.075 m
- g (gravitational constant): 9.81 m/s^2
- ACyl (cylinder area): Calculated from bore size of 115mm

The masses that are included in the piston assembly mass are:

- Bare Piston: 1591 g
- Piston Pin: 1001.5 g
- Circlips: 13.4 g
- Top Ring: 31.7 g
- 2nd Ring: 29.5 g
- Oil Ring: 18.4 g
- Connecting Rod Small End Section (including the bush): 481.5 g

Figure 4.1 shows the regions on the connecting rod where the distances and masses were obtained.

Piston assembly mass contains the connecting rod small end mass, which is the region on the right of the rod shank beginning (small end bush is not shown in this figure). Connecting rod shank mass is the mass between the beginning of rod shank up to the location of the strain-gages.

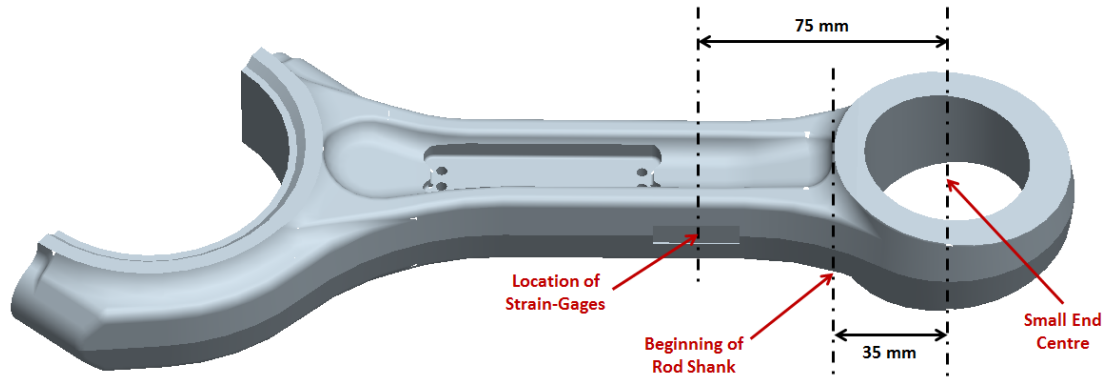


Figure 4.1 : Connecting Rod Regions.

For the calculation in this thesis, the following assumptions were made:

- Crank angle speed fluctuations were ignored.
- Only the crank angles between 90° before and after TDC were taken into account since that region is where most of the forces are seen.
- All of the dimensions and masses were either obtained from technical drawings, or calculated from 3D CAD models. Due to the tolerances involved, the parts that are tested in the engine will have different dimensions and weights. For engine testing, all of the dimensions will be measured accurately to get the exact values. Moreover, piston assembly mass will be measured, together with the instrumentation, to have better accuracy. However it's not possible to measure the masses in the connecting rod, since the part must be cut into three pieces; therefore the 3D CAD model calculations will be used.
- Connecting rod mass and the distances within the connecting rod were taken roughly from the 3D CAD models. For engine testing, the distances will be measured and the mass will be calculated accurately.
- Crankcase pressure data was not available, therefore the pressure was assumed to be zero.

4.1.2 Variables

The only difference between the two codes is the variables that are entered into each code. The first code will calculate the strain-gage force; therefore it uses friction force and cylinder pressure variables. The second code inputs the strain-gage force

that is calculated from the first code; and together with the cylinder pressure, calculates the friction force.

The friction force was taken from a previous CAE analysis done in AVL Excite. For simplicity, only the data at 2200 rpm was used and only half of the compression and expansion stroke was used (from 90° before TDC to 90° after TDC). This study only covered the friction for the rings, piston friction (through piston skirts) was not calculated in this study. Figure 4.2 shows the friction curve that was used.

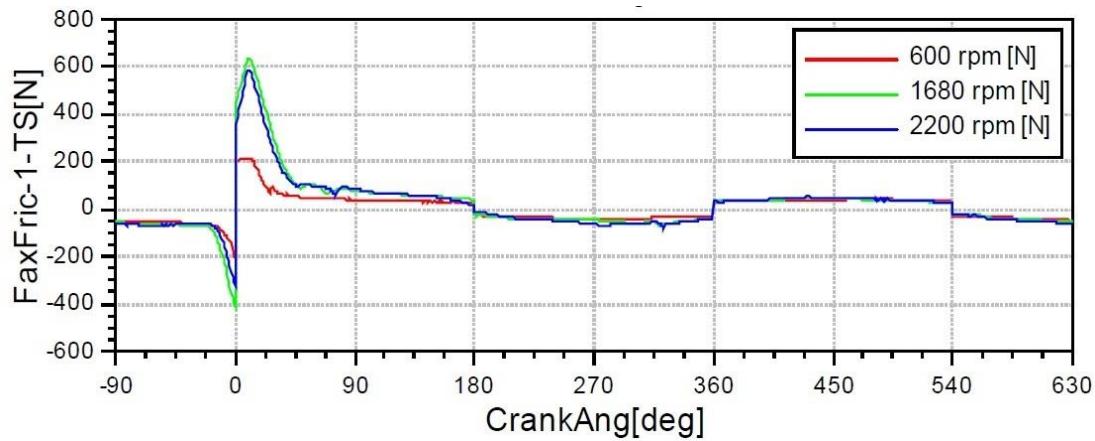


Figure 4.2 : Friction curve used for this study.

The second input to the program is cylinder pressure curve, as shown in Figure 4.3.

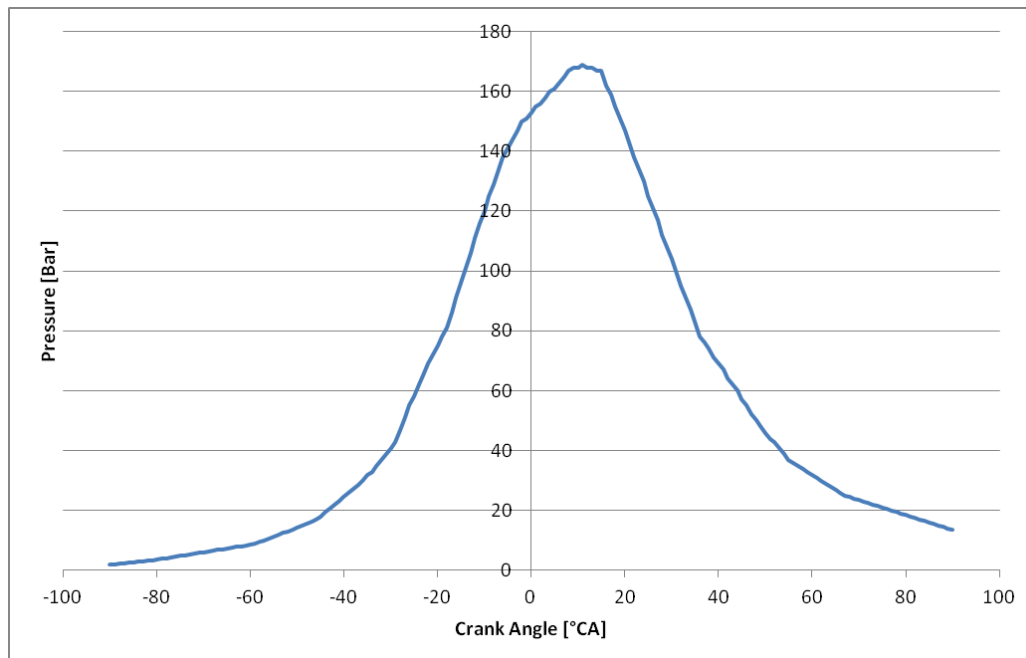


Figure 4.3 : Cylinder pressure curve used for this study.

4.2 Results

The first two calculated values were piston motion and connecting rod angle. Using the equations (2.5), (2.6) and the constants shown earlier, the graphs in Figure 4.4 were obtained.

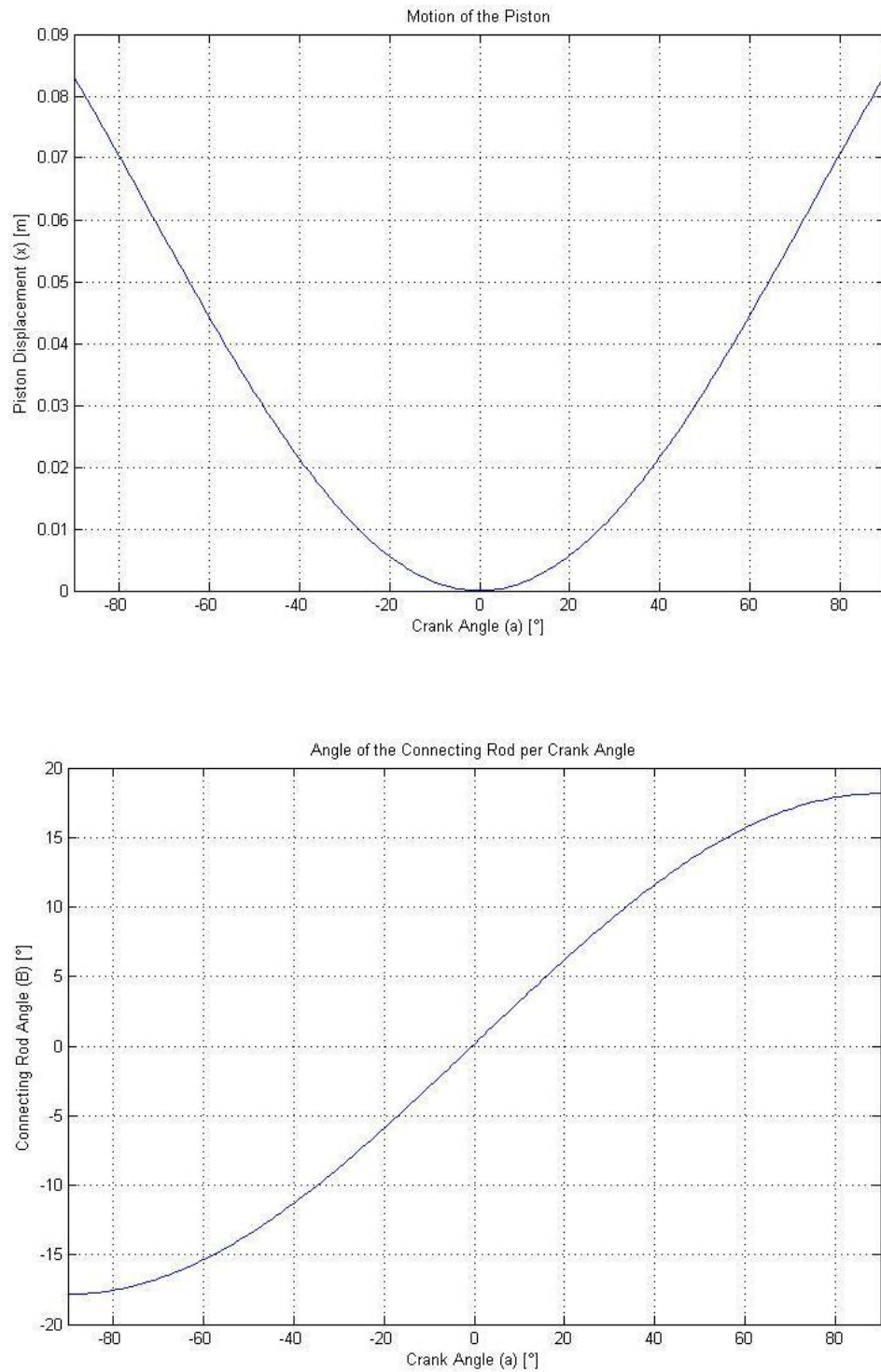


Figure 4.4 : Graphs for Piston Motion and Connecting Rod Angle.

In order to obtain piston acceleration, the second derivative of piston motion data was calculated. Since working under numeric conditions, while performing derivation, MATLAB only calculates the difference between the two consecutive values in the given data set. Naturally, this causes one data point to be lost for each derivation operation. To prevent data mismatch, the piston motion was calculated again using a wider range of crank angle, between 91° before and after TDC.

This derivation operation only provides the “dx” and “d²x”. Therefore the data was divided by “dt” two times, in order to finalize the derivation operation with respect to time. “dt” value was obtained using engine speed, where the time to rotate the crank by one degree was calculated.

The piston acceleration data was multiplied by piston assembly weight (including connecting rod small end weight) to obtain the piston system inertia, which is shown in Figure 4.5.

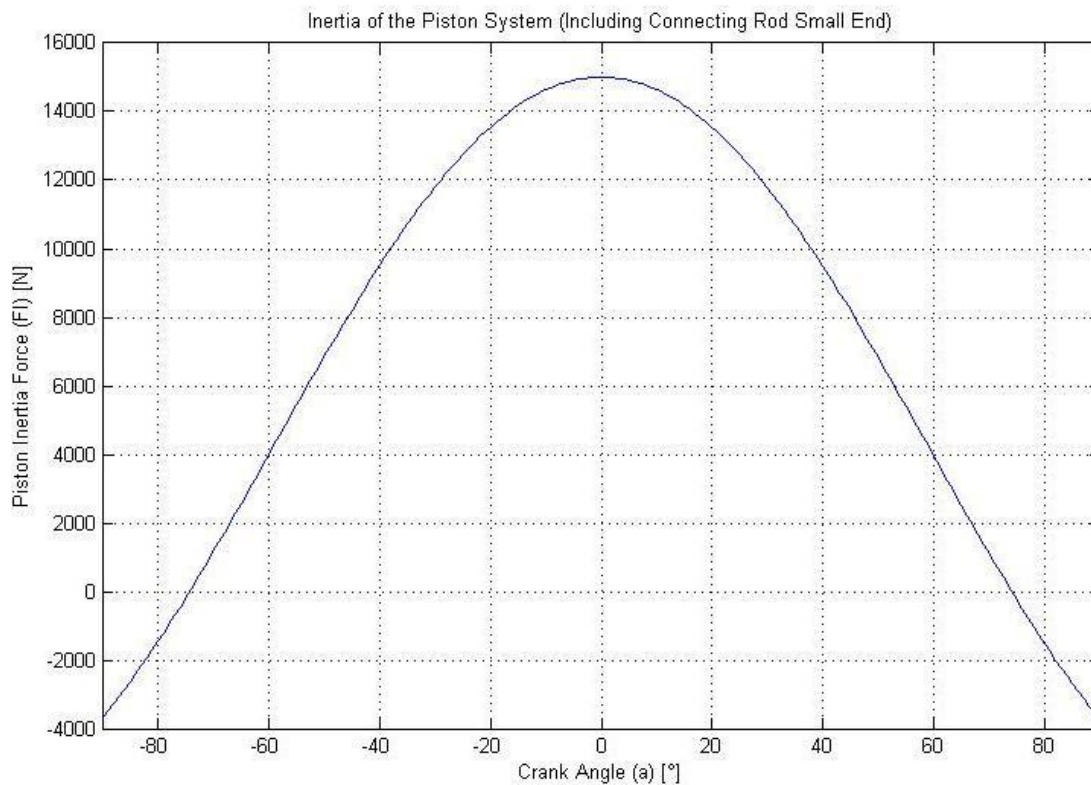


Figure 4.5 : Graph for Piston Inertia.

Similar to piston acceleration, connecting rod angular velocity was calculated using the expanded crank angle data. This time, one data point was removed from the final data vector, since derivative operation is used only once.

The angular velocity data was used to calculate the inertia for the connecting rod shank region, which is shown in Figure 4.6.

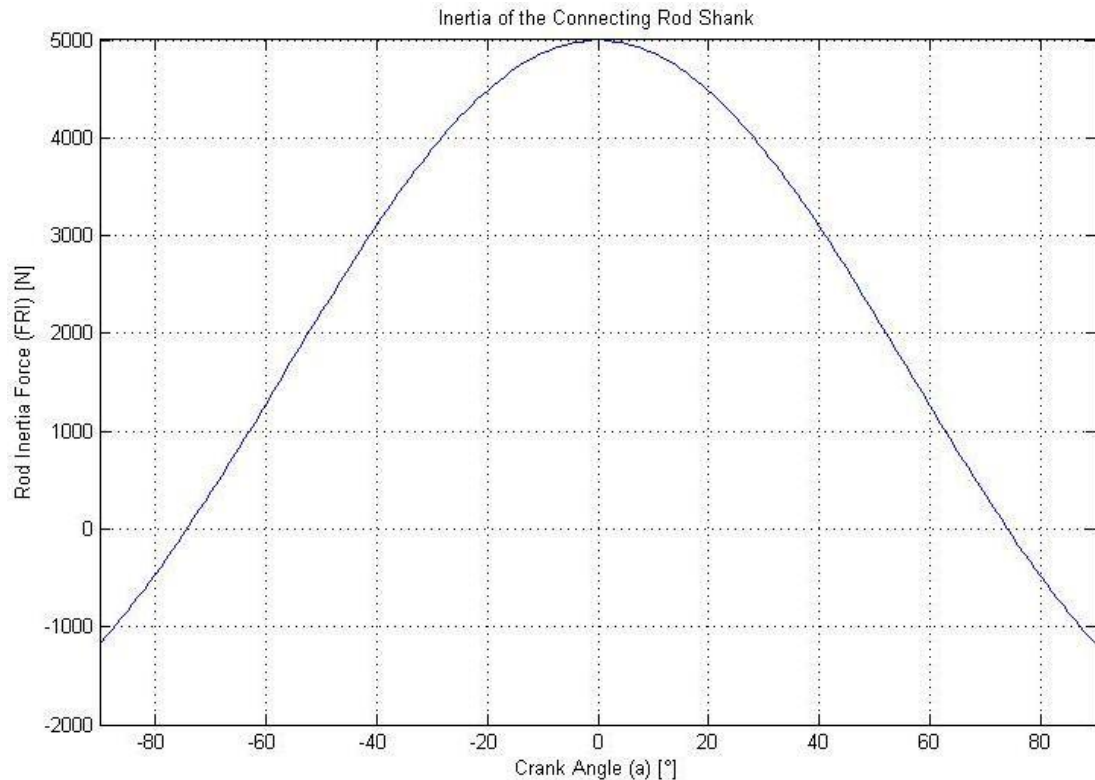


Figure 4.6 : Graph for Connecting Rod Inertia.

Next step was to calculate net gas force exerted on the piston. Cylinder pressure data was entered into the program and multiplied by cylinder bore area (obtained using bore diameter). Then the forces applied to the connecting rod and strain-gages were calculated using force balance.

Figure 4.7 shows the comparison of the net gas force and strain-gage force. Note the reduction of force in majority of the region, which shows the force spent on friction and inertia forces. The inertia phenomenon works in favour of the strain-gage force before -70° and after 70° , where the strain-gage force is higher than piston force due to the deceleration of moving components.

Final step is to enter the strain-gage data into the second code to re-generate the friction data. This was performed successfully with no difference between the original friction data and the regenerated data. Figure 4.8 shows the result.

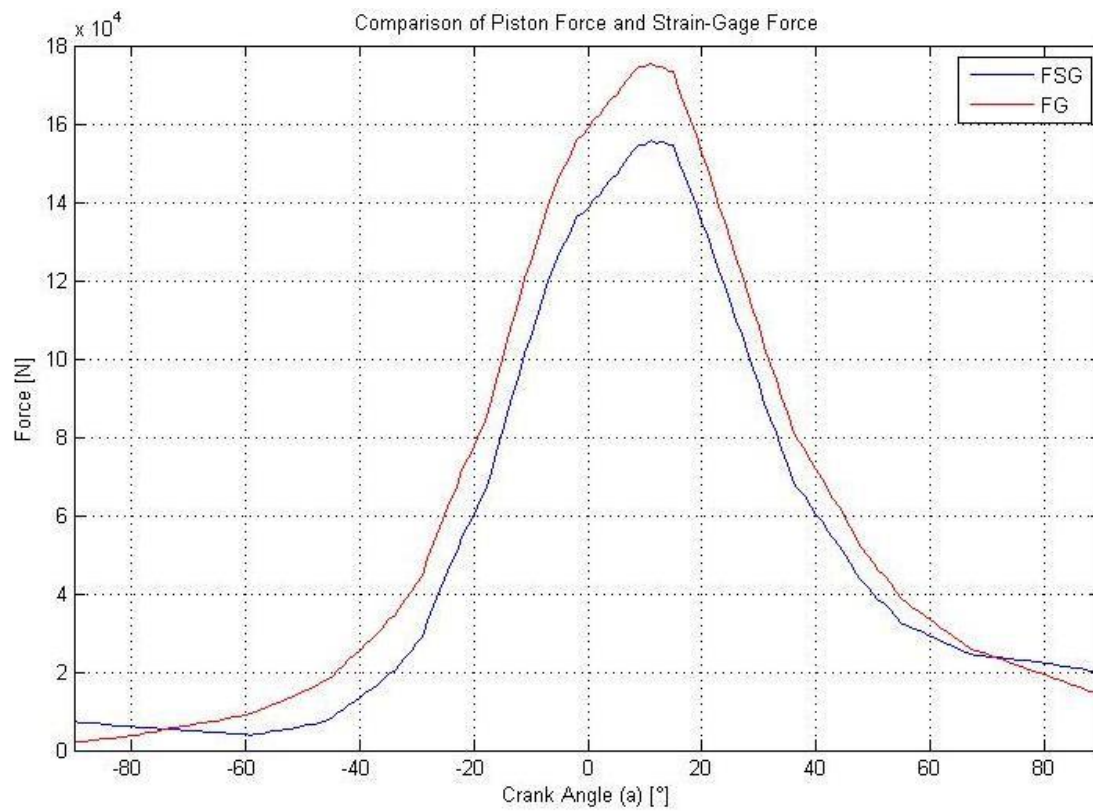


Figure 4.7 : Graph for comparing Piston Force and Strain-Gage Force.

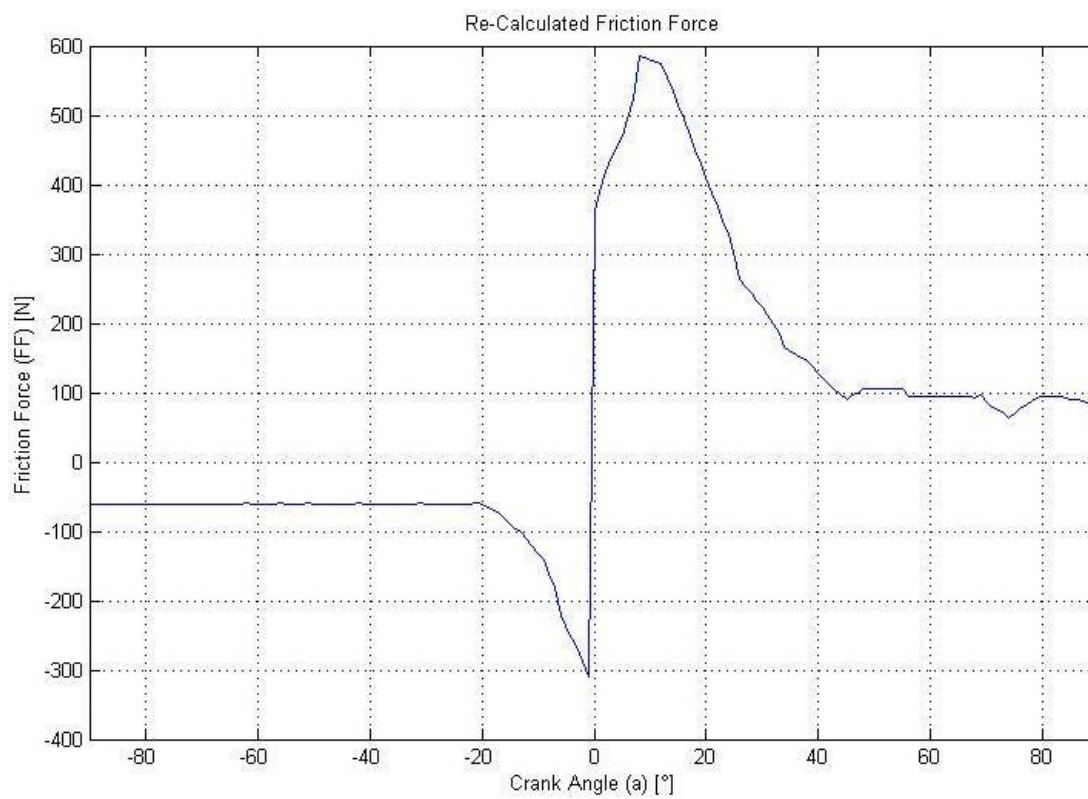


Figure 4.8 : Regenerated Piston Assembly Friction Force.

The data has been further assessed to better understand the results. Firstly, the difference between the gas force and strain-gage force has been calculated in percentage, as shown in Figure 4.9.

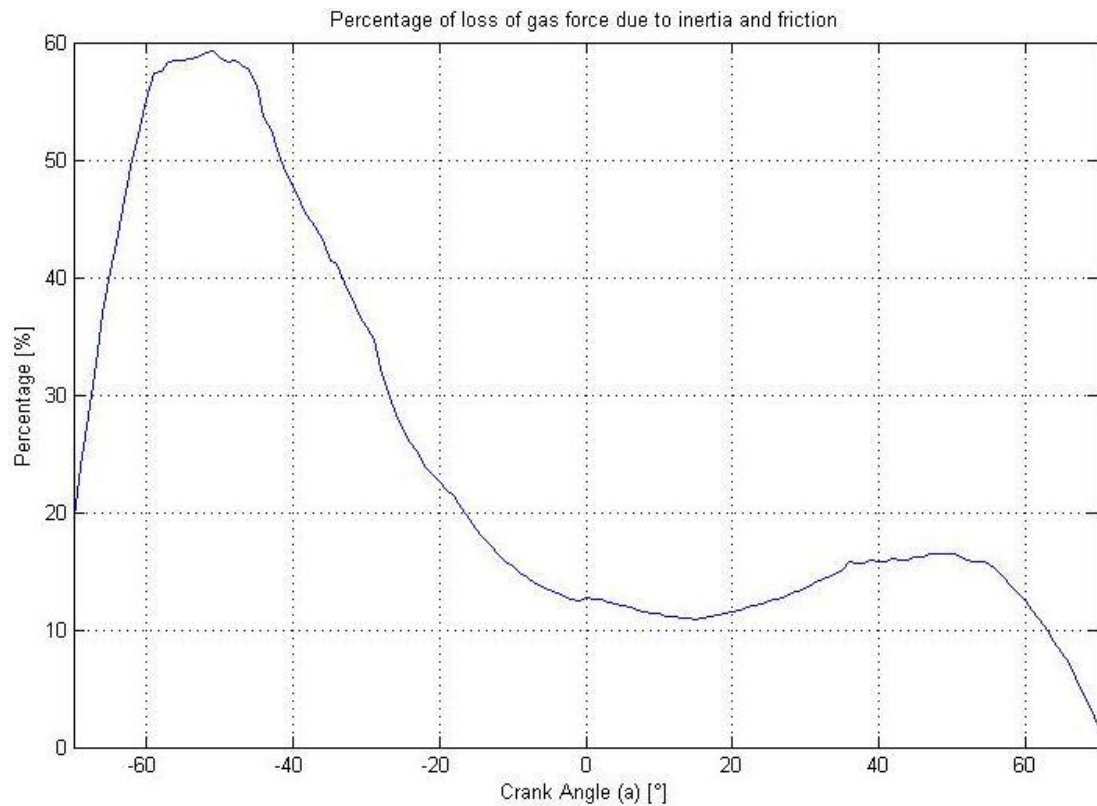


Figure 4.9 : Percentage of loss of gas force.

This graph shows how much of the gas force is being used for inertia and frictional forces. The losses are maximized during compression stroke around 50° before TDC, where the losses reach to 60% of the total gas force. In the compression stroke, the gas pressure naturally follows an exponential curve due to the linear decrease in volume (this curve is generally interrupted in diesel engines via the usage of pilot fuel injection, which aims to create a smooth transition to the pressures reached during ignition, thereby improving the NVH properties of the engine). Around 50° before TDC, the gas force is still at a low value, around 10 bars, which is around 6% of the maximum pressure reached during the engine cycle. The frictional losses are also low at this point. The top and 2nd rings are pushed onto the cylinder wall via the pressure at the back of the rings, which is low due to low cylinder pressure. Therefore the dominating effect in this region is the inertia losses.

As the piston travels towards TDC, the percentage of loss decreases due to increasing gas force. Despite the increase in inertia and friction losses, the cylinder pressure

dominates. The losses are reduced to around 11% in 10° after TDC, where maximum cylinder pressure is reached.

In the expansion stroke, the percentage of losses starts to increase again due to decreasing gas force, however the value cannot reach above 17% since the gas contains higher pressure than in the compression stroke, due to increased temperature and number of gas molecules.

Figure 4.10 shows the comparison of the inertia and friction forces.

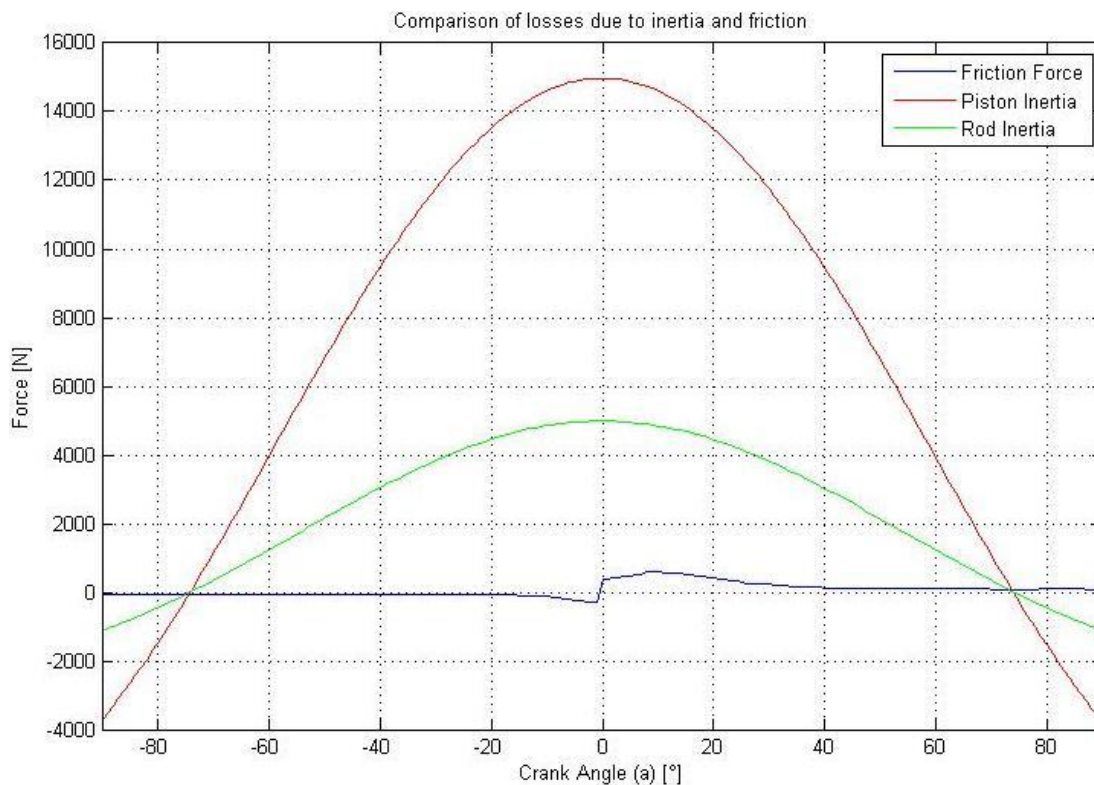


Figure 4.10 : Comparison of losses due to inertia and friction.

The friction data obtained for this study contains friction losses that maximizes around 10° after TDC, around the same time as the cylinder pressure peak; however the maximum value is around 600N, which is a low value when compared with gas force and inertia forces. Piston skirt friction needs to be calculated in CAE to get a higher and more representative friction force.

Inertia forces are naturally maximized at TDC, where connecting rod shank inertia reaches 5 kN and piston assembly inertia reaches 15 kN. At TDC, piston ring friction is 12% of the rod inertia, 4% of the piston inertia and 3% of the total inertia of rod and piston. Since at TDC, total losses are 11% of the gas forces, it's concluded that piston ring friction is 0.33% of the gas forces reached at TDC.

Looking at the complete cycle between -90° and $+90^\circ$, inertia losses are found to be 17% of the gas forces, whereas the friction losses are 0.24%. This study has been done on the maximum speed of the engine (where maximum inertia forces are reached) and at full load (where maximum gas pressures are reached). During low speed and low load operation, the effect of friction would be greatly increased due to decreased inertia and gas forces.

In these calculations, it must be noted that engine performance is negatively affected by the forces in compression stroke (between -90° and TDC). During this period, the piston is forced to go up, which causes the cranktrain to do work (which is supplied by the combustion in other cylinders and the total rotating inertia of the engine). After TDC, the effect is reversed and the piston does work on the cranktrain.

The inertia losses cause a decrease on the total force, which works in favour of the engine during the compression stroke, but in return the work done by the piston is also reduced during expansion stroke. Looking at the total cycle, the energy in the system is not changed due to the inertia forces, the negative and positive effects cancel each other out. Therefore it can be concluded that the inertia forces does not directly affect engine performance (the effect would be on engine speed limitation where heavier components would not allow high engine speeds to be reached).

The friction however, always causes negative effect. Note that during compression stroke, the friction forces are entered as “negative”. This causes the total force to be increased during this cycle, which increases the losses. During expansion stroke, friction forces are positive, which decreases the total force, causing the piston to do less work. Therefore it can be concluded that decreasing friction has a direct influence on engine performance and fuel economy, especially at low load and low engine speeds.

5. CONCLUSIONS AND RECOMMENDATIONS

The piston system has a very complicated working structure and it is very difficult to fully understand what is going on. Despite the history of over one hundred years, the system continues to preserve some of the mystery. The engine design world is dominated by computer analysis, yet the analysis done on the piston continues to lag behind in terms of realism and representation of engine conditions. This study aims to assist in solving this problem. Via making a real life measurement in a running engine, one of the most important objectives is to improve the CAE analysis done on piston assembly. The telemetry system constructed for this study uses advanced technology to maximize measurement accuracy, which would otherwise be not beneficial.

To summarize the work performed in this study:

- Decision to perform real-time measurement in a running engine was given. Piston assembly friction and 2nd land pressures were decided to be measured. 2nd land pressure measurement was to be used to improve understanding and the analysis done on the ring dynamics, whereas the friction measurement was to be used for analysis involving both ring and piston dynamics.
- The idea was to preserve the engine as much as possible and perform the measurements without affecting engine operating condition or any of the engine parameters. For this goal, decision to use a telemetry system was given.
- To obtain as high accuracy as possible for friction measurement, Instantaneous IMEP method was used together with cylinder pegging technique.
- The telemetry system was purchased from one of the world's leading manufacturers, Datatel Telemetry.
- The principles of the telemetry system was studied to understand the physics involved and to make adjustments wherever necessary.

- Piston and connecting rod instrumentations have been performed with utmost care. Each adhesive and solder was carefully selected for best performance in engine operating conditions. After each step, engine dynamometer testing was performed using dummy parts, to make sure that no problems are observed in actual test.
- Modifications were performed on various parts of the engine including cylinder block, ladderframe and crank.
- Packaging checks have been performed to check that neither the telemetry equipment, nor the instrumentations contact with the neighbouring components.
- Telemetry system was fully constructed in laboratory environment and it was proven that the system is functional.
- Sensor calibrations were performed with great care on the accuracy of the calibration. Relation between the end results was sought; where for strain-gages, this is the relation between connecting rod force and receiver output voltage; and for pressure sensors, it is the relation between pressure and receiver output voltage.
- Engine was prepped ready for engine dynamometer testing.
- For friction calculations, MATLAB codes were written, taking into account the tiniest details like engine speed fluctuations. These codes were checked using a previous study on piston friction.

This study is to be continued, firstly to obtain the sensory data from the engine. After the data is obtained, a lot of work is required to understand and fully assess the meaning of this data. The data is then recommended to be used to validate various CAE analyses done on the piston itself, as well as the engine. Following these studies, it is possible to perform design changes and improvements on this engine.

It is possible to improve this work to get higher measurement accuracy. Cylinder bore deformations can be taken into account while calculating the gas force applied to the piston. Bore deformations can be calculated using the combination of cylinder block measurements and CAE studies for hot bore deformations. Moreover, connecting rod inertia can be calculated accurately using CAE tools, to remove the

constant cross-section assumption for rod shank and cylinder assumption for small end.

Such an understanding will provide the basis for the future studies done on the same or other engines of similar type. Furthermore, obtaining such a telemetry capability brings out further possibilities to perform other measurements on the same or different engines, to continue improving the designs.

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APPENDICES

APPENDIX A: MATLAB Code to Formulize Piston Assembly Friction

```
%Variables and constants are defined symbolically.
%From left to right:
%Time, crank throw, connecting rod lenght, pin offset,
%piston assembly mass (including rod small end), rod shank mass,
%distance from small end centre to beginning of rod shank,
%distance from small end centre to strain-gage location,
%gravitational constant, force read by strain-gages,
%cylinder bore, cylinder pressure, crankcase pressure.
syms t R L s mP mR r1 r2 g FSG ACyl PCyl PCC

%alpha, crank angle, is defined as a function of time,
%to capture the effect of speed fluctuations.
a=sym('a(t)');

%x: Equation of motion for the piston
x = R + L - R*cos(a) - (L^2 - (R*sin(a) + s)^2)^.5;

%Beta: Connecting rod angle
B = asin((R*sin(a) + s)/L);

%Weight of piston and rod shank.
FW = mP * g;
FRW = mR * g;

%Inertia of rod shank
FRI = mR * (diff(x,t,2)*cos(B) - (r2^2-r1^2)/2*diff(B,t)^2);

%Piston assembly inertia
FI = mP * diff(x,t,2);

%Net gas force
FG = ACyl * (PCyl - PCC);

%Rod force
FRod = FRI + FSG - FRW;

%Frictional Force
FF = FG + FW - FI - FRod*cos(B);
```

APPENDIX B: MATLAB Code to Formulize Strain-Gage Force

```
%Variables and constants are defined symbolically.
%From left to right:
%Time, crank throw, connecting rod lenght, pin offset,
%piston assembly mass (including rod small end), rod shank mass,
%distance from small end centre to beginning of rod shank,
%distance from small end centre to strain-gage location,
%gravitational constant, force read by strain-gages,
%cylinder bore, cylinder pressure, crankcase pressure.
syms t R L s mP mR r1 r2 g FF ACyl PCyl PCC

%alpha, crank angle, is defined as a function of time,
%to capture the effect of speed fluctuations.
a=sym('a(t)');

%x: Equation of motion for the piston
x = R + L - R*cos(a) - (L^2 - (R*sin(a) + s)^2)^.5;

%Beta: Connecting rod angle
B = asin((R*sin(a) + s)/L);

%Weight of piston and rod shank.
FW = mP * g;
FRW = mR * g;

%Inertia of rod shank
FRI = mR * (diff(x,t,2)*cos(B) - (r2^2-r1^2)/2*diff(B,t)^2);

%Piston assembly inertia
FI = mP * diff(x,t,2);

%Net gas force
FG = ACyl * (PCyl - PCC);

%Rod Force
FRod = (FG + FW - FI - FF)/cos(B);

%Strain-gage measurement
FSG = FRod + FRW - FRI;
```

APPENDIX C: The Formula for Piston Assembly Friction and Strain-Gage Force

FF =

$$\begin{aligned}
 & ACyl*(PCyl-PCC)+mP*g- \\
 & mP*(R*\cos(a(t))*diff(a(t),t)^2+R*\sin(a(t))*diff(a(t),\$ (t,2))+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(3/2)}*(R*\sin(a(t))+s)^2*R^2*\cos(a(t))^2*diff(a(t),t)^2+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*R^2*\cos(a(t))^2*diff(a(t),t)^2-1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*(R*\sin(a(t))+s)*R*\sin(a(t))*diff(a(t),t)^2+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*(R*\sin(a(t))+s)*R*\cos(a(t))*diff(a(t),\$ (t,2)))- \\
 & (mR*((R*\cos(a(t))*diff(a(t),t)^2+R*\sin(a(t))*diff(a(t),\$ (t,2))+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(3/2)}*(R*\sin(a(t))+s)^2*R^2*\cos(a(t))^2*diff(a(t),t)^2+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*R^2*\cos(a(t))^2*diff(a(t),t)^2-1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*(R*\sin(a(t))+s)*R*\sin(a(t))*diff(a(t),t)^2+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*(R*\sin(a(t))+s)*R*\cos(a(t))*diff(a(t),\$ (t,2)))*(1- \\
 & (R*\sin(a(t))+s)^2/L^2)^{(1/2)}-(1/2*r^2^2- \\
 & 1/2*r1^2)*R^2*\cos(a(t))^2*diff(a(t),t)^2/L^2/(1-(R*\sin(a(t))+s)^2/L^2))+FSG- \\
 & mR*g*(1-(R*\sin(a(t))+s)^2/L^2)^{(1/2)}
 \end{aligned}$$

FSG =

$$\begin{aligned}
 & (ACyl*(PCyl-PCC)+mP*g- \\
 & mP*(R*\cos(a(t))*diff(a(t),t)^2+R*\sin(a(t))*diff(a(t),\$ (t,2))+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(3/2)}*(R*\sin(a(t))+s)^2*R^2*\cos(a(t))^2*diff(a(t),t)^2+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*R^2*\cos(a(t))^2*diff(a(t),t)^2-1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*(R*\sin(a(t))+s)*R*\sin(a(t))*diff(a(t),t)^2+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*(R*\sin(a(t))+s)*R*\cos(a(t))*diff(a(t),\$ (t,2)))-FF)/(1- \\
 & (R*\sin(a(t))+s)^2/L^2)^{(1/2)}+mR*g- \\
 & mR*((R*\cos(a(t))*diff(a(t),t)^2+R*\sin(a(t))*diff(a(t),\$ (t,2))+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(3/2)}*(R*\sin(a(t))+s)^2*R^2*\cos(a(t))^2*diff(a(t),t)^2+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*R^2*\cos(a(t))^2*diff(a(t),t)^2-1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*(R*\sin(a(t))+s)*R*\sin(a(t))*diff(a(t),t)^2+1/(L^2- \\
 & (R*\sin(a(t))+s)^2)^{(1/2)}*(R*\sin(a(t))+s)*R*\cos(a(t))*diff(a(t),\$ (t,2)))*(1- \\
 & (R*\sin(a(t))+s)^2/L^2)^{(1/2)}-(1/2*r^2^2- \\
 & 1/2*r1^2)*R^2*\cos(a(t))^2*diff(a(t),t)^2/L^2/(1-(R*\sin(a(t))+s)^2/L^2))
 \end{aligned}$$

APPENDIX D: MATLAB Code to Numerically Calculate Strain-Gage Force

```
%Numeric Calculation
%Find Strain-Gage forces using Friction data
%FF and PCyl are entered beforehand.

%Constants
R = 0.072;
L = 0.233;
s = 0.0006;
mP = 3.167;
mR = 0.226;
r1 = 0.035;
r2 = 0.075;
g = 9.81;
ACyl = 0.115^2*pi/4;

%Crankcase pressure data is not yet available, so was assumed as
zero.
PCC = 0;

%Enginespeed
rpm = 2200;

%Crank angle, only the data between 90° before and after TDC will be
used.
a=[-90:90];
ar=deg2rad(a);           %In radians
%Secondary crank angle,
%To compensate for the data loss during derivative operations.
ar2=deg2rad([-91:91]);

%Time interval is calculated according to engine speed
dt=60/rpm/360;

%x: Equation of motion for the piston
x = R + L - R*cos(ar) - (L^2 - (R*sin(ar) + s).^2).^5;
%Derivative is calculated using secondary crank angle
x2 = R + L - R*cos(ar2) - (L^2 - (R*sin(ar2) + s).^2).^5;
diffx2=diff(x2,2)/dt/dt;

%Beta: Connecting rod angle
B = asin((R*sin(ar) + s)/L);
B2 = asin((R*sin(ar2) + s)/L);
diffB=diff(B2)/dt;
diffB=diffB(1:181);
%One more point is removed from the data since derivative is used
once.

%Weight of piston and rod shank.
FW = mP * g;
FRW = mR * g;

%Inertia of rod shank
FRI = mR * (diffx2.*cos(B) - (r2^2-r1^2)/2*diffB.^2);

%Piston assembly inertia
FI = mP * diffx2;
```

```

%Net gas force
FG = ACyl * (PCyl - PCC);

%Rod Force
FRod = (FG + FW - FI - FF) ./ cos(B);

%Strain-gage measurement
FSG = FRod + FRW - FRI;

```

APPENDIX E: MATLAB Code to Numerically Calculate Piston Assembly Friction

```
%Numeric Calculation
%Find Friction forces using Strain-Gage data
%FSG and PCyl are entered beforehand.

%Constants
R = 0.072;
L = 0.233;
s = 0.0006;
mP = 3.167;
mR = 0.226;
r1 = 0.035;
r2 = 0.075;
g = 9.81;
ACyl = 0.115^2*pi/4;

%Crankcase pressure data is not yet available, so was assumed as
zero.
PCC = 0;

%Enginespeed
rpm = 2200;

%Crank angle, only the data between 90° before and after TDC will be
used.
a=[-90:90];
ar=deg2rad(a); %In radians
%Secondary crank angle,
%To compensate for the data loss during derivative operations.
ar2=deg2rad([-91:91]);

%Time interval is calculated according to engine speed
dt=60/rpm/360;

%x: Equation of motion for the piston
x = R + L - R*cos(ar) - (L^2 - (R*sin(ar) + s).^2).^5;
%Derivative is calculated using secondary crank angle
x2 = R + L - R*cos(ar2) - (L^2 - (R*sin(ar2) + s).^2).^5;
diffx2=diff(x2,2)/dt/dt;

%Beta: Connecting rod angle
B = asin((R*sin(ar) + s)/L);
B2 = asin((R*sin(ar2) + s)/L);
diffB=diff(B2)/dt;
diffB=diffB(1:181);
%One more point is removed from the data since derivative is used
once.

%Weight of piston and rod shank.
FW = mP * g;
FRW = mR * g;

%Inertia of rod shank
FRI = mR * (diffx2.*cos(B) - (r2^2-r1^2)/2*diffB.^2);

%Piston assembly inertia
FI = mP * diffx2;
```

```
%Net gas force
FG = ACyl * (PCyl - PCC);

%Rod force
FRod = FRI + FSG - FRW;

%Frictional Force
FF = FG + FW - FI - FRod.*cos(B);
```


CURRICULUM VITAE



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