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INTEGRATION AND VERIFICATION OF MINIATURE FLUID FILM PRESSURE SENSORS IN HYDRODYNAMIC LINEAR GUIDES

Previously, a 2D simulation model for hydrodynamic linear guides with two reduction factors has been developed to calculate oil film pressure and floating heights/angle numerically. However, no method was available to verify the oil film pressure experimentally but only with floating heights measurement. Therefore, different pressure sensor's integration methods were tested in a stationary Plexiglas rail to measure fluid film pressure inside the lubrication gap. The pressure sensors were statically and dynamically calibrated. However, floating heights could not be measured with the preliminary used Plexiglas rail. This paper reports the suitable integration of pressure sensors into a stationary steel rail to compensate this drawback. It focuses on the measurement of pressure rise using pressure sensors inside the lubrication gap in combination with the floating heights. Experimental results have shown that the oil film pressure inside the lubrication gap can be measured using pressure sensors, which draw conclusions about cavitation and lack of lubrication. The variation of oil film pressure measured along the length of the carriage can be used to improve the simulation model i.e. the reduction factors. The pressure measurement can help to identify the lubrication conditions and further actions can be taken to improve the lubrication cycle.

1. INTRODUCTION

1.1. HYDRODYNAMIC GUIDEWAYS AND INITIAL RESEARCH

Hydrodynamic guides have great importance in the machining industry to carry high normal loads. Hydrodynamic linear guides have a good damping coefficient, are low cost [1] and leads to high quality machined surfaces. Additionally, the constant supply of lubricant in hydrostatic bearings increases the running cost, which reduces efficiency. The disadvantages of hydrodynamic linear guides are unstable floating behaviour due to variable oil film pressure along the length of a carriage, mixed friction regimes (boundary lubrication) during starting and stopping of a carriage, and air entrapment inside the lubrication gap. Due to oil

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inlet pressure, a wedge geometry is formed by the produced moment [2]. A positive pressure is generated due to wedge-shaped film and relative movement between the surfaces of guide. Surfaces completely separated by lubricant (no asperity contact) have internal friction only due to the viscosity of the lubricant, which can be defined as shear stress over shear strain. The machine table's acceleration, velocity, and deceleration and oil inlet pressure, have a significant impact on unstable floating behaviour during operation.

Zhang et al.[3, 4] developed a single-phase 2-D simulation model based on the Reynolds equation to estimate oil film pressure and floating heights of hydrodynamic linear guides. Two reduction factors have been derived to compensate for the difference between 2-D and 3-D models and to take care of the simplification of the Reynolds equation and the losses in the experimental setup due to air entrapment and contact surface roughness. The reduction factors were developed by comparing numerical floating heights with experimental results (measured using eddy current sensors). In advance of the measurement strokes (both forward and backward), a lubrication cycle is used in which the lubricant was supplied through two lubrication grooves and the carriage was run back and forth to distribute the lubricant evenly. It was only possible to measure the floating heights of the machine table but the oil film pressure measurement enables to further improve the 2-D simulation model. Additionally, it will also help in optimization of a lubrication cycle, prediction of performance and identification of problems such as air entrapment, uneven wear or misalignment.

1.2. PRESSURE MEASUREMENT METHOD IN HYDRODYNAMIC GUIDEWAYS

One of the key parameters in hydrodynamic guides is the oil film pressure, which influences the operation of guides [5, 6]. The challenge was to measure the pressure separately from shear stress in the fluid. Previously, Sinanoglu et al. used sixteen manometer tubes to analyse the pressure development of journal bearing for different velocity variation and various texturing of shaft surface [7]. Additionally, thin film pressure sensors consisting of thin material layers (with a total thickness of 6 μm) on the sliding surface of the bearing were used to measure the oil film pressure of journal bearing by Ichikawa et al. [8], Mihara et al. [9, 10], Mihara and Someya [11], and Someya and Mahira [12]. Mihara and co-workers have carried out oil film pressure measurements as well as temperature and strain measurements of the bearing surface in an engine test. The sensors have been deposited by the physical vapour deposition (PVD) technique onto the bearing surface. Thin film pressure sensors are used to measure high pressure on the order of 1 GPa but the pressure is generated in engines on the order of 10–100 MPa, a sensor with reduced and uniform strain sensitivity has been developed by optimizing the composition of copper, manganese and nickel contained in the sensor alloy [11]. However, in hydrodynamic linear guides, it is not possible to use thin film pressure sensors because of very small pressure difference of 0.5 MPa [13].

Preliminary Ibrar et al. [13] tested and compared various integration methods of miniature pressure sensors in a Plexiglas rail (relatively shorter in length as compared to steel rail). In Fig. 1, the Plexiglas rail with three pressure sensors with finalized sensor integration method (also cut-view of sensors integration method) is shown. Plexiglas rail was used to simplify the manufacturing process. In this way, the test of different pressure sensor integration methods were easily possible, whereas, the eddy current distance sensors only worked with metals. Through the use of adopted sensor integration method (as shown in Fig. 1) a significant improvement has been observed that measures only oil film pressure. Another advantage of using a plexiglass rail and carriage was the ability to visualize inside the lubrication gap to study the flow. The main goal of the study in this paper is to introduce the finalized sensor integration method into the original steel rail to measure oil film pressure.

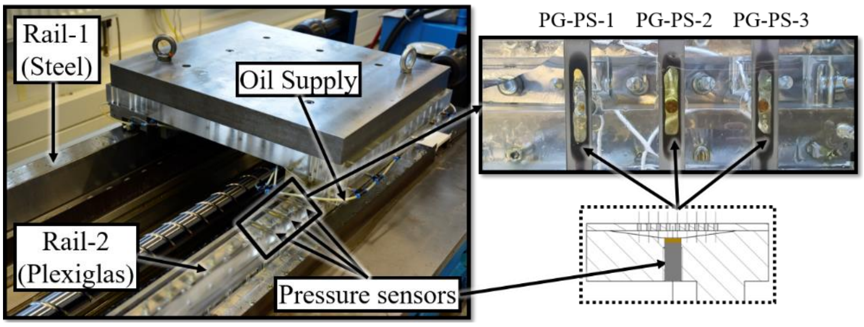

Fig. 1. Test-stand with Plexiglas rail and different integration methods of pressure sensors used by Ibrar [13]

The machine table has run for forward strokes for different feed-rates and pressure is measured through three sensors integrated in a Plexiglas rail. A surface pressure of machine table of 0.24 bar is considered to study the pressure inside the lubrication gap. Lubrication cycle shown in Fig. 12 is used with a pause time of 5 s. All three pressure sensors display six pressure peak corresponding to the six lubrication grooves depicted in Fig. 2. All three sensors measured same pressure behaviour and it is also visible that the pressure sensors capture the sudden changes in pressure within the lubrication gap. It indicates the ability of pressure sensors to measure dynamic pressure.

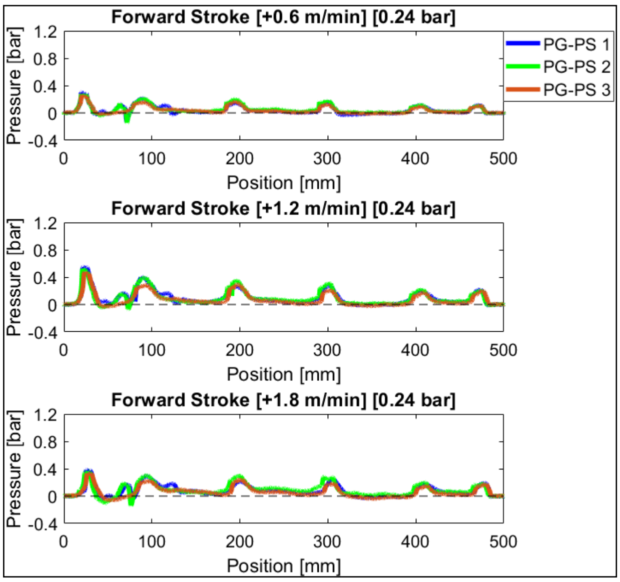

Fig. 2. Oil film pressure measured with sensors integrated in Plexiglas rail (shown in Fig. 1) with PG-PS-1, PG-PS-2 and PG-PS-3. Machine table with Surface pressure of 0.24 bar is considered in this case [13]

For the generation of oil film pressure, it is also important to improve a lubrication cycle. From the pressure curves measured using sensors integrated in Plexiglas rail, it has been observed that due to poor lubrication and relative movement, air is entrapped inside the lubrication gap. When there is air between the sensor and the sliding surface, pressure sensors do not produce any reasonable pressure values. To compensate for that, enough lubricant should be supplied before each stroke to avoid cavitation. Additionally, the variables used in the lubrication cycle, like lubrication time, the number of lubrication grooves, and the pause between lubrication and pressure measurement, should be improved through experimental means. With the measurement of oil film pressure, it will be easier to understand the air influence, and further actions can be taken to make improvements.

2. EXPERIMENTAL SETUP WITH STEEL RAIL

To measure the oil film pressure using the sensors integrated inside the steel rail, experiments were performed with the help of the test-stand shown in Fig. 6. The pressure sensors were again calibrated hydrostatically according to the method described by Ibrar [13], after they were integrated (with favourable integration method) into the steel-rail.

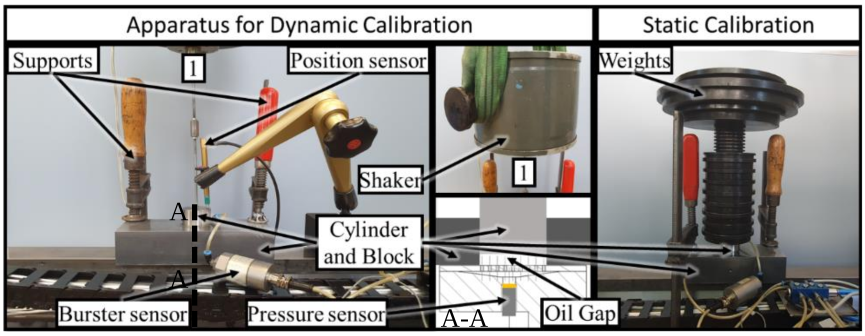

Fig. 3. Test-stand for both static calibration (Right) and dynamic calibration (Left) of sensors integrated in steel rail

After the integration of pressure sensors inside steel rail, it was crucial to perform hydrostatic and hydrodynamic calibration of sensors integrated in the steel rail. Test-stand for both static and dynamic calibrations are shown in Fig. 3, where a steel block with sealing ring is placed on the steel rail and to minimize leakage, supports are used. In hydrodynamic calibration cylinder is attached with the shaker to achieve three different amplitudes at four different frequencies (5, 10, 15 and 20 Hz). For hydrostatic calibration, the weights are directly applied to the cylinder to examine the pressure corresponding to weights utilized. Additional sensor named Burster is used for comparison purposes.

With the help of static and dynamic calibration apparatus, the sensors are calibrated to make sure that they measure the accurate oil film pressure inside the lubrication gap. In Fig. 4, it can be seen that the both PS-1 and Burster showed similar pressure with the error of less than 4%. Whereas, the error between actual weights and PS-1 is in between 8% and 14% which is due to oil leakage as it was not possible to avoid leakages completely using sealing ring and supports. During experimentation it has been also observed that oil flows out from the lubrication gap as the load increases on the cylinder. Additionally, the sensors are calibrated dynamically which is shown in Fig. 5. Pressure sensors PS-1 and Burster sensor showed similar pressure fluctuation at four different frequencies and 3 different amplitudes. The displacement of cylinder measured using the position sensor is shown in Fig. 5, which shows that the motion is transferred successfully from shaker to cylinder.

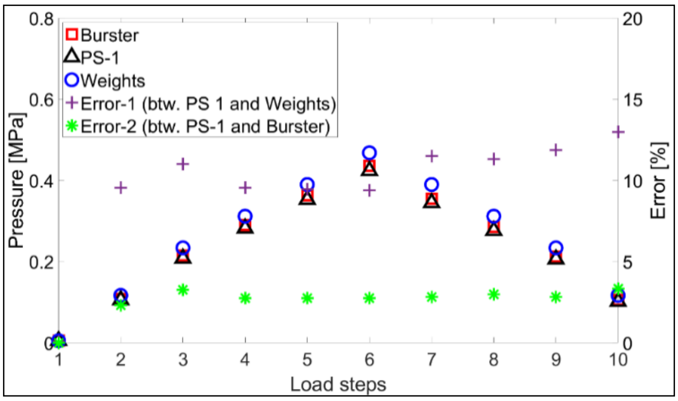

Fig. 4. Static calibration of pressure sensor in steel rail and comparison with Burster sensor and actual weights

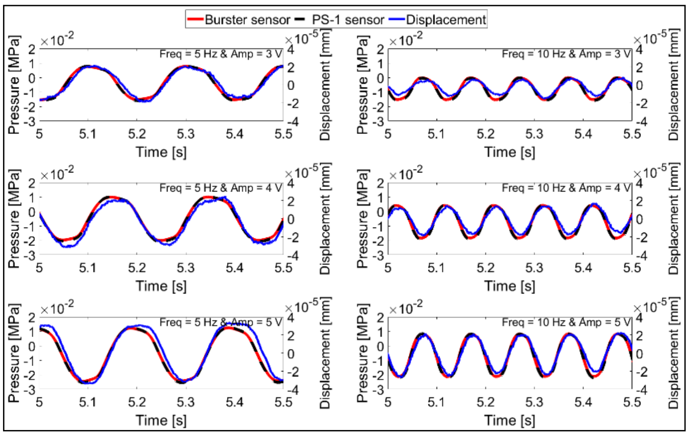

Fig. 5. Dynamic calibration of pressure sensor in steel rail and Burster sensor (for comparison) and displacement of cylinder

A hydrodynamic linear guide is shown in Fig. 6, where a carriage with sliding bars slide on two parallel steel rails. Distance eddy current sensors are attached to the carriage's four ends to measure the relative lubrication gap (or the carriage's roll and pitch) between the carriage and both rails.

Shell Tonna S-68 is used as a lubricant for this study, with a viscosity of 68 mm^2/s at 40°C (8.6 mm^2/s at 100°C) and a density of 879 kg/m^3. Strain gauge miniature pressure sensors have a temperature range of –20 to 70°C to measure oil film pressure inside the lubrication gap. Total seven strain gauge type (analogue) pressure sensors “PS-5KD” (maximum capacity of up to 5 bar) by Kyowa 0 are integrated in a steel rail to get detailed pressure information in the lubrication gap. Sampling rate for all pressure sensors in this study was set to 600 Hz.

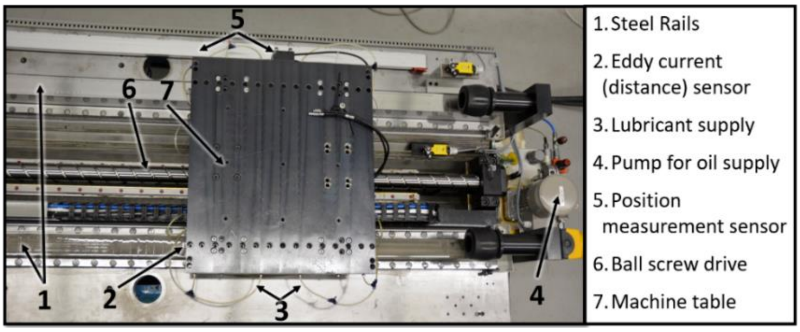

Fig. 6. Test-stand (Top-view) of hydrodynamic linear guides with pressure sensors integrated inside one steel rail

The integration method of seven pressure sensors in a steel rail and start position of the sliding bar can be seen in Fig. 7. The sensors with a round sensing surface (with a diameter of 6 mm and a sensing surface thickness of 0.6 mm) are able to measure oil film pressure along the length of the carriage, but no variation in pressure can be measured along the width of the rails.

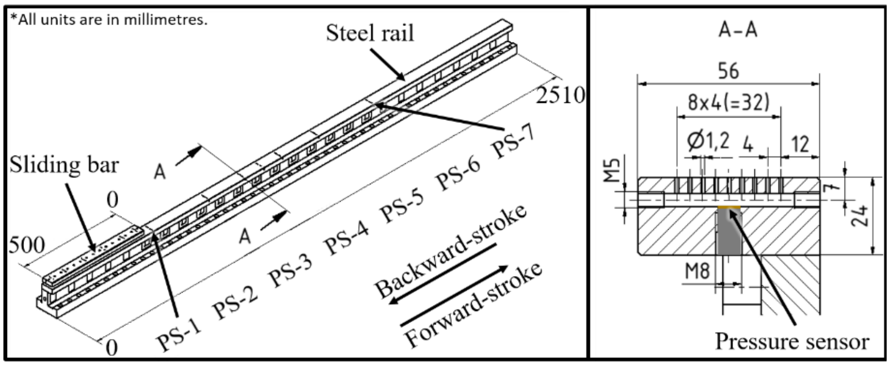

Fig. 7. Sketch of the steel-rail with integrated pressure sensors

After integration of sensors in the steel rail, the entire 1.7 m stroke length can be used at a maximum possible feed-rate of 100 m/min in both backward and forward directions. The steel rail has been produced from alloy tool steel grade W 1.2311, quenched, tempered to a strength of 1000 N/mm^2, gas-nitrided with a single-hardening depth of 0.3 mm and a hardness of 600 HV. A screw drive is used, and a force sensor is installed between the motor and carriage to measure the exact force applied during operation. The machine table has roller bearing side guiding on both sides with a steel rail (without integrated sensors) to avoid movement in width-direction. Whereas, the machine table has only one connecting surface with rail (with integrated sensors) that is separated by hydrodynamic lubrication.

Total six lubrication grooves have been introduced in each sliding bar, as shown in Fig. 8. There are four holes (1.8 mm diameter) in each lubrication groove and surfaces of both sliding bars were machined to have better and smooth contact with the steel rails. After preparation of sliding bar's surfaces the actual contact between two surfaces were measured by contact paper.

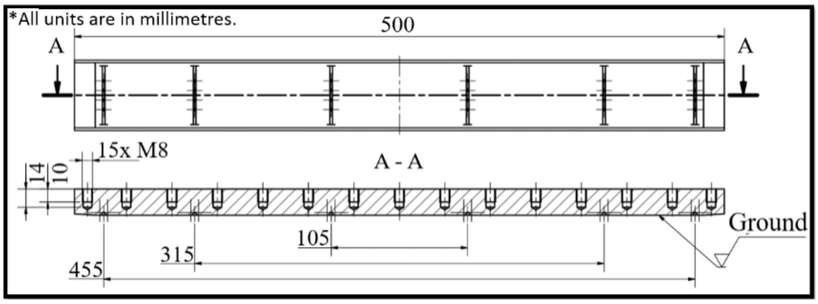

Fig. 8. Sketch of the sliding rails with 6 lubrication grooves

It is well understood that the oil film pressure and load-carrying capacity of linear guides are affected by the carriage feed rate. Pressure sensors are integrated in the steel rail in this study to analyse pressure at various feed-rates. The acceleration and deceleration of the machine table were kept as high as possible so that pressure could be measured for the maximum possible carriage's constant velocity range. However, due to the stick-slip effect or to avoid the carriage's jerk motion, a very short range of acceleration and deceleration is not possible. Furthermore, the minimum lubrication gap is determined not only by the feed rate but also by the surface pressure or load that must be carried. Three different surface pressures (0.24, 0.454, and 0.914 bar – due to the weight of the machine table) are considered, and oil film pressure has been recorded at different feed-rates. The experiments were performed at room temperature, and the change in viscosity of the lubricant due to a temperature rise inside the lubrication gap is negligible.

3. FLUID FILM PRESSURE MEASUREMENT

3.1. PREPARATION

Before the slide bars (with six lubrication grooves) were used to measure the pressure, it was important to examine the contact region between the sliding bar and steel rail (with integrated sensors). Spotted ink is pasted on the steel rail and the bars were cleaned so there is no oil inside the lubrication gap. The machine table is then placed on both steel rails and moved back and forth to examine the contact between both surfaces. Subsequently, the machine table is separated from the steel rail and the white papers were used to capture the ink from the sliding bar’s surfaces. Figure 9 illustrates the contact pattern of sliding bar with steel rail (with integrated sensors). The contact pattern with steel rail including sensors is irregular and non-uniform which leads to low or negative pressure values where there is poor contact between sliding bar and steel rail. But it reflects a real practical situation of machine tools.

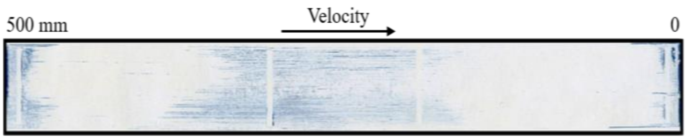

Fig. 9. Contact pattern developed between sliding bar and steel rail during sliding

3.2. RESULTS

Understanding the pressure curves along the length of the sliding bar can be challenging, particularly without focusing on the contact pattern between the sliding bar and rail. Without considering the contact pattern, the pressure curves may not provide a complete or accurate picture of the lubrication conditions within the system. Therefore, it is important to analyse the contact pattern alongside the pressure curves to comprehend the behaviour of the hydrodynamic linear guides. It leads to the assumption that the sliding bar has an inwardly curved surface in the longitudinal direction between 1 and 4 and between 4 and 6 lubrication groove. Both shape deviations will be affected the pressure curve.

The pressure curves measured using sensors´ integration method (described in Fig. 7) in steel rail are shown in Fig. 10. Whereas, the sensors integrated in steel rail show that the pressure curves correspond to the contact pattern, the contact is strong between second and fourth lubrication grooves and at the ends of the sliding bar. This can be verified by pressure curves as they show high-pressure values in between second and fourth lubrication grooves. However, in some regions of the sliding bar, there is no contact with the rail, which causes a pressure drop or negative pressure in those areas and cavitation occurs.

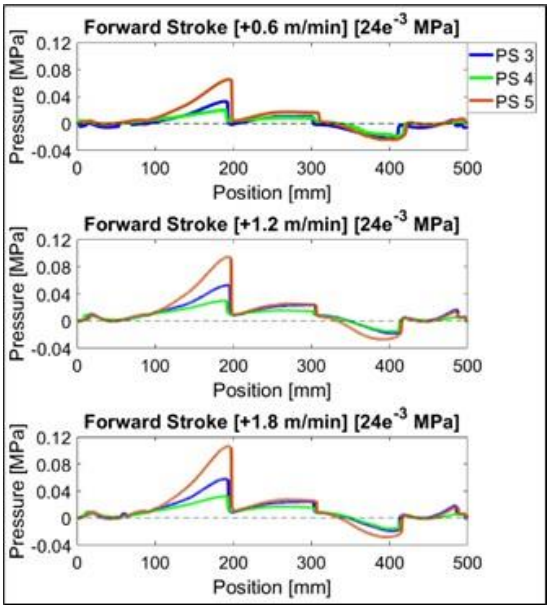

Fig. 10. Oil film pressure measured with sensors integrated in steel rail (shown in Fig. 7) with PS-3, PS-4 and PS-5. Machine table with Surface pressure of 0.24 bar is considered in this case

Figure 11 illustrates the pressure curves for the three different surface pressures showed that the pressure values increased slightly with the third surface pressure (0.7 bar), it is possible that the sensors did not capture the difference of hydrodynamic pressure for different weights. This could be due to various factors, including misalignment of the sliding bar or variations in the oil film thickness that affect the sensors' ability to accurately measure the pressure. Alternatively, the irregular contact pattern of the sliding bar with steel rail have led to changes in the hydrodynamic pressure distribution within the lubrication gap, resulting in higher pressure values in some areas. Although both sliding bars were prepared together, the resulting contact patterns between the sliding bars and the rail were found to be different. This discrepancy could potentially be attributed to the assembly of the sliding bars with the machine table, which may have caused variations in the alignment or position of the sliding bars relative to the rail. Further investigation would be needed to confirm the causes of the observed differences in oil film pressure between the machine table weights.

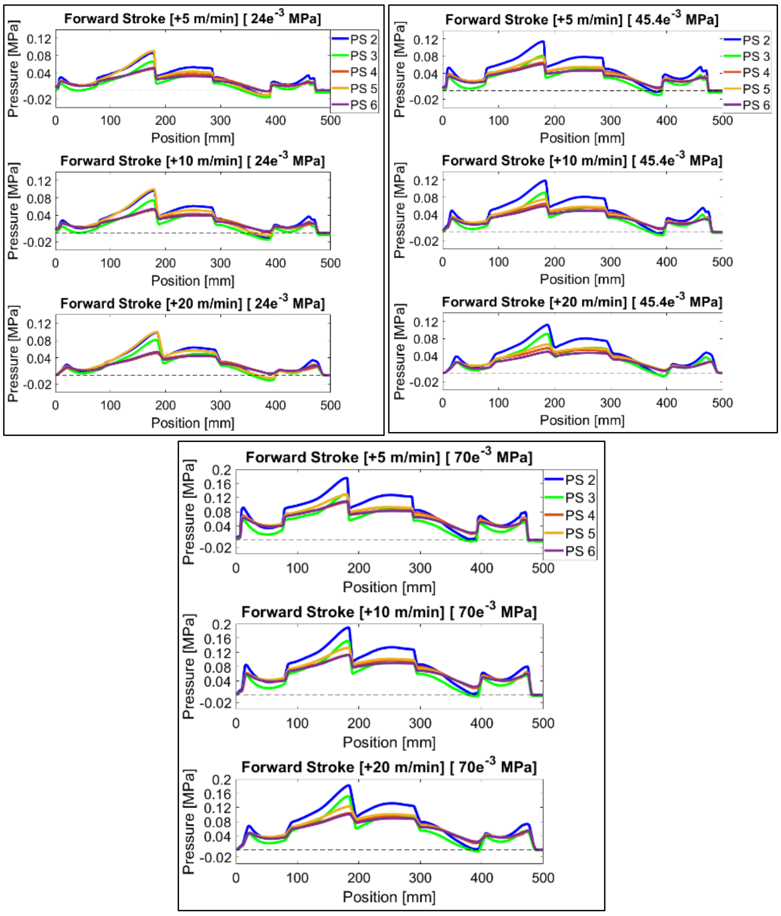

Fig. 11. Oil film pressure at different feed-rates (5, 10 and 20 m/min) with different pressure sensors (PS-2, PS-3, PS-4, PS-5, PS-6 and PS-7) integrated in a steel rail. Machine table with surface pressure of 0.24, 0.454 and 0.7 bar (Top-Left, Top-Right and Bottom respectively) is considered in this case

4. THE EFFECTS OF LUBRICATION CYCLE

4.1. LUBRICATION CYCLE

On one hand, air entrapment inside the lubrication gap reduces the load carrying capacity, and on the other hand, sensors do not show realistic pressure values in the presence of air between the sensor's surface and the moving carriage's surface. The influence of air entrapment has also been verified using hydrostatic lubrication. The air entrapped inside the lubrication gap can be easily removed (or the influence of air inside the lubrication gap eradicated) with the help of hydrostatic lubrication, but in this study only hydrodynamic lubrication is considered.

In order to reduce friction and measure accurate oil film pressure through sensors, it is also important to improve the lubrication cycle of hydrodynamic linear guides. In the test stand (see Fig. 6, six tubes were connected to six lubrication grooves in the carriage to supply lubricant inside the lubrication gap with the help of a pump. The volumetric flow rate through all six tubes is identical. To understand the effects of the number of lubrication grooves, it was important to study pressure and floating heights with different numbers of lubrication grooves.

Fig. 12. Lubrication cycle used to conduct experiments for oil film pressure measurement

The lubrication cycle used in this study is shown in Fig. 12, where it can be seen that lubricant is supplied before each stroke for measurement. As soon as the lubricant supply stops, the carriage moves downward due to gravity force, and the oil squeezes out of the lubrication gap. Therefore, it is important to have a pause between lubrication and the measurement stroke to get stable floating heights. This pause time has a great influence on pressure measurement and on the floating heights of the machine table. If the pause time is too long then it has a negative influence on pressure measurement but positive influence on floating heights. Due to lubrication, the machine table is lifted and as soon as lubrication stopped, the oil squeezes out of the lubrication gap due to gravity force on the machine table. After a long pause, there will be less oil inside the lubrication gap, which is not optimal condition for pressure measurement. However, for floating heights it is suitable condition as the maximum lubrication gap will be reduced and there is less room for irregularity of floating heights during operation. It is also difficult to control to initial floating heights of the machine table and the pressure build inside the lubrication gap depends on the initial floating heights.

On the other hand, if the pause time is too short then the initial floating heights are not stable and the squeezing of oil due to gravity takes place during the strokes, which is not suitable. Although, a short pause time helps in measuring oil film pressure inside the lubrication gap. The pressure sensors work better in the presence of adequate lubricant inside the lubrication gap.

4.2. VARIATION OF PAUSE TIME

In this study, the effect of different pause times (1, 5, 10, 20, and 30 s) on pressure measurement inside the lubrication gap of hydrodynamic linear guides is investigated. A surface pressure of 0.24 bar is used to compare different pause times. Total seven miniature pressure sensors are integrated in steel-rail at different positions. In Figs 13 and Fig. 14

pressure curves for different pause times between the lubrication and measurement stroke have been compared for 5 m/min and 10 m/min feed-rates respectively. Pressure curves for both forward strokes (left side) and backward strokes (right side) have been shown for different pressure sensors PS-1, PS-3 and PS-7 from top to bottom.

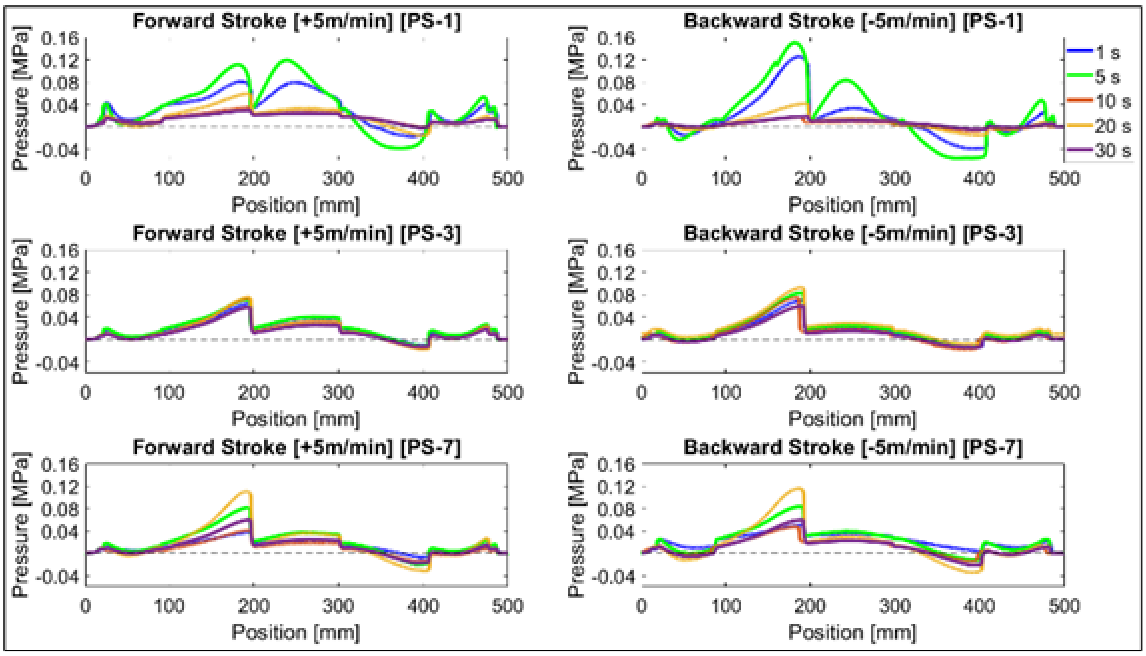

Fig. 13. Oil film pressure inside lubrication gap at different pause times for pressure sensors PS-1, PS-3 and PS-7 from top to bottom respectively. Left: Forward strokes and Right: Backward strokes, at 5 m/min feed-rate

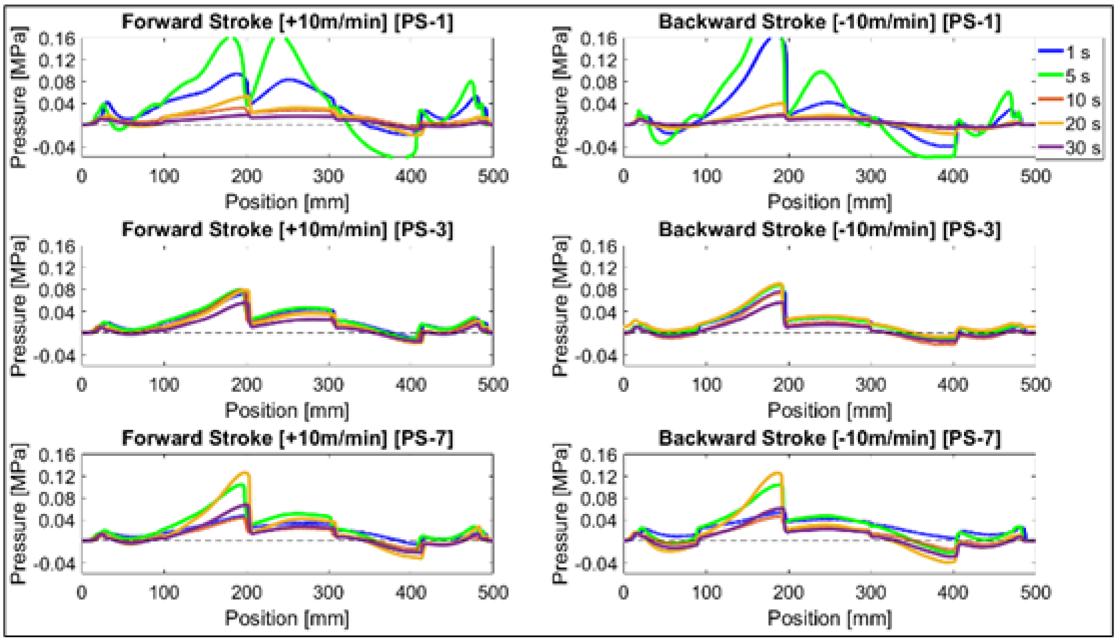

Fig. 14. Oil film pressure inside lubrication gap at different pause times for pressure sensors PS-1, PS-3 and PS-7 from top to bottom respectively. Left: Forward strokes and Right: Backward strokes, at 10 m/min feed-rate

When the pause time is longer, the pressure values measured by sensors become significantly smaller, as evidenced by the pressure curves in both Figs 13 and Fig. 14. From the results it has been observed that 5 s pause time is suitable for measurement of pressure inside the lubrication gap. Although 20 s pause time also shows the same results as 5 s pause time but sensor PS-1 (which is the closest to machine table for forward stroke) shows that the pressure measured after 20 s pause time are relatively smaller. Whereas, the pressure curves for 5 s pause time shows even better results than one second pause time. The reason why the pressure values are high for a short pause time is that the lubrication gap contains a sufficient amount of oil. An increase in the amount of oil present in the lubrication gap leads to improved lubrication and more accurate pressure measurement. Additionally, the irregular floating heights due to short pause time is a significant problem which cannot be ignored.

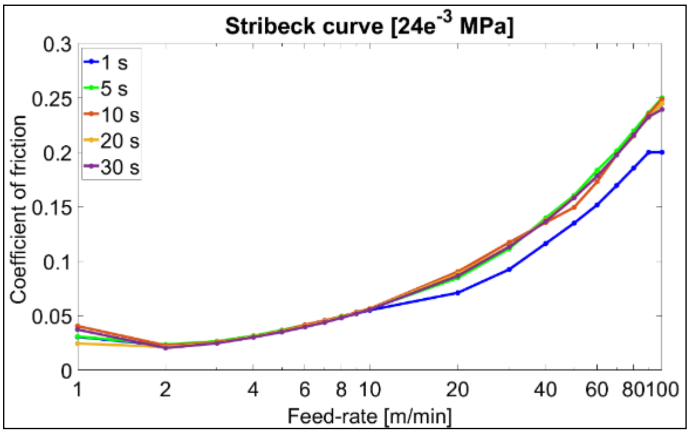

Fig. 15. Stribeck curve for different pause times at different feed-rates. Surface pressure of 0.24 bar is considered

Stribeck curves have been also generated for different feed-rates and different pause times for a surface pressure of 0.24 bar. It is clear in Fig. 15 that the curves for each pause time are in the fluid friction regime and the coefficient of friction increases with velocity from 2 m/min. In between 1 m/min and 2 m/min the curves have a down trend which shows that they are in mixed-friction regime. Friction coefficient for each pause times shows almost the same values and trend except 1 s pause time, which is understandable because the pause time is too short and therefore, the floating heights are greater than other pause times.

5. CONCLUSION

Previously, different sensors integration methods have been tested to measure film pressure of hydrodynamic linear guides using Plexiglas rail and sliding bars. After finding the best variant, which showed realistic pressure curves, it is then integrated into steel rail to measure film pressure to realise the lubrication conditions inside the lubrication gap. Sensors integrated in steel rail were also calibrated statically and dynamically. Oil film pressure at different pause times between the lubrication and measurement stroke has been compared. It has been observed that the pause time has a great influence over film pressure measurement.

Additionally, the oil film pressure do not show any large variation in pressure due to different surface pressure of machine table which can be due to the poor contact pattern between the rail and sliding bar. Because of irregular contact, it is possible that the small holes above the sensor are not completely covered by the sliding surface and oil flows out from some holes, which leads to low pressure values.

Further improvements are needed to improve the contact by pattern by manufacturing a machine table with sliding bar as one part so there is no variation between the contact pattern of both sliding bars and rails. Additionally, further experimentations are needed to study the effects of number of lubrication grooves on lubrication conditions. Because the lubrication conditions play a very important role in irregularity of the floating heights of the machine table.

ACKNOWLEDGEMENTS

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Appendix I. for manuscript No. JME0000027-2023 – Integration and Verification of Miniature Fluid Film Pressure Sensors in Hydrodynamic Linear Guides

DISCUSSION ON THE METHOD OF PRESSURE MEASUREMENT

It seems to the editors that the issue of pressure measurement method in Hydrodynamic Linear Guides along the length of the slides for both forward and backward strokes is very complex and difficult to arbitrarily resolve. Therefore, we suggest publishing the article together with a discussion between reviewers and authors.

Reviewers comments:

Reviewer #1
Reviewer #1 has not referred to the Method of Pressure Measurement

Reviewer #2:
I think that your calibration method is valid for the measuring system sensing the pressure variation as far as at the entrance of each oil hole. However, the pressure at the entrance of the oil holes on the guide way surface does not indicate the correct oil film pressure caused by wedge action in hydrodynamic lubrication, because the wedge action is greatly disturbed by the existence of the oil holes especially in case of thin oil film thickness in mixed lubrication regime and the volume of thin oil film which should be sensed is extremely smaller than the oil volume in the pressure measuring system. The ratio of the oil volume in the measuring system to that in the thin oil film is proportional to the effect of oil compressibility as well as the elastic deformation of the pressure sensor on the sensing of the oil film pressure due to hydrodynamic lubrication. Therefore, I do not have much confidence in the experiment results.

Reviewer #3:
Measurement of oil film pressure:
In short, the measuring method shown in Fig. 1 is not acceptable, because the transducer detects the oil pressure within the small chamber, which is not in direct relation to the pressure of the oil film at the slide way. In fact, there is certain time lag caused by the oil capacitance in small chamber along with the mass-effect of the oil.

Authors' response

In Fig. A1, various integration methods of strain-gauge type pressure sensors have been compared by measuring the oil film pressure inside the lubrication gap. For geometry variant I, the sensors were integrated at two different width positions. In geometry of section A-A, the sensing surface was directly connected to lubrication gap. In section B-B, it can be seen that the sensing surface is connected to lubrication gap through a small single hole. Purpose of the variants was to understand which influences or side effects have on the pressure measuring like shearing oil flow and cavitation.

The geometry variant II (section D-D) was designed to avoid the influence of side effects to the measuring results.

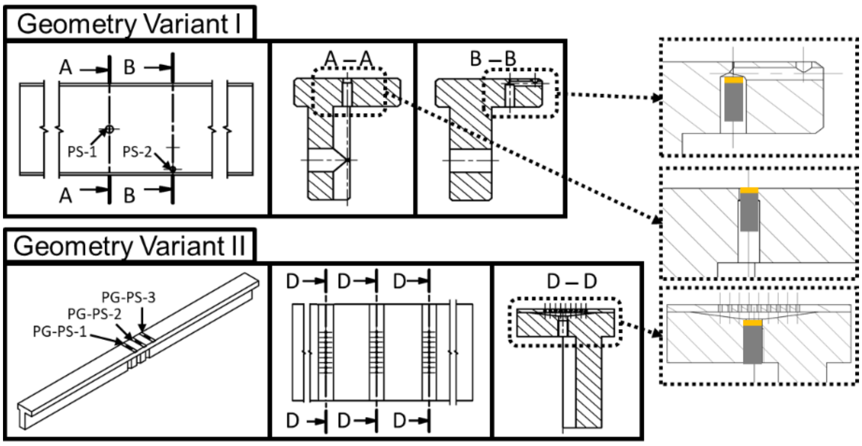

Fig. A1. Integration method of sensors for both geometry variants

However, the sensors integration method shown in geometry variation I were not suitable as it can be shown in results in Fig. A2. One of the reason was that the sensing surface was too small (equal to the diameter of sensor, which is 6 mm). After that, geometry variation II was developed to reduce the air influence on oil film pressure measurement and it showed very promising results as shown in Fig. A3. Where the sensors show six pressure peaks (dynamic) in response to six lubrication grooves in the sliding bars of the carriage. Contrary to that, the result for variant I (Fig. A2) does not allow any conclusion to the geometry of the sliding bars. The only drawback of variant II is that it does show variation in the direction of rail's width. Another interesting thing was that all three sensors with same integration method in geometry variation II have shown almost similar behaviour and pressure peaks, which is the dynamic behaviour. All variations were tested in Plexiglas rail first and after seeing improvements with geometry variation II, the sensors were integrated in steel rail.

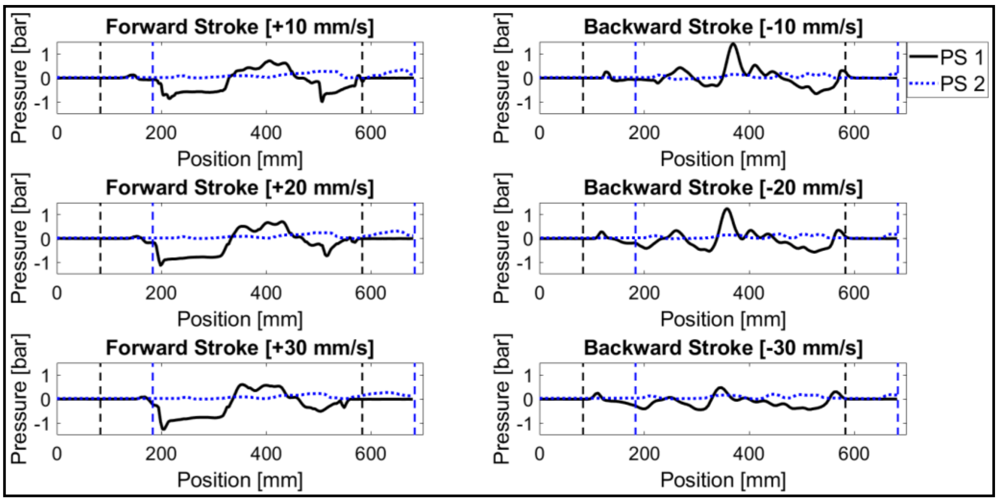

Fig. A2. The pressure measured for forward and backward strokes inside the lubrication gap using geometry variant I

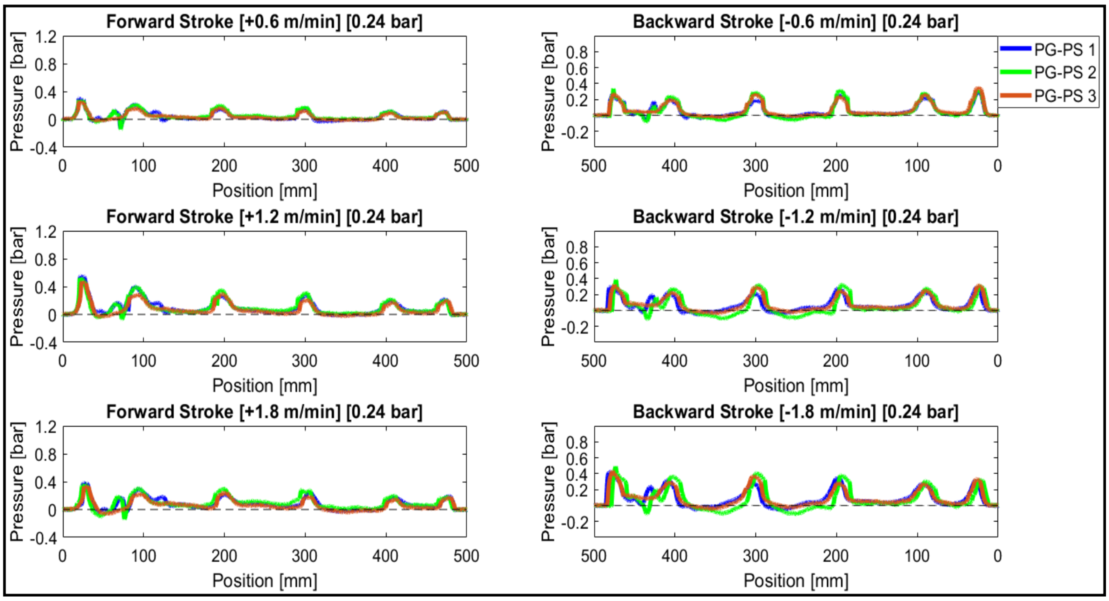

Fig. A3. The pressure peaks (6 peaks for 6 lubrication grooves) measured for forward and backward strokes inside the lubrication gap using geometry variant II

Additionally, different pause times (5, 10, 15, 30, and 60 s) between the lubrication and pressure measurement have been studied and the results shown in Fig. A4. Where it can be clearly seen that the small pause time lead to higher pressure values which is due to the more oil inside the lubrication gap. It have been previously stated that the lubrication cycle plays an important role in oil film pressure measurement. This is the reason why the lubrication grooves have been increased to achieve homogeneous lubricant presence inside the lubrication gap.

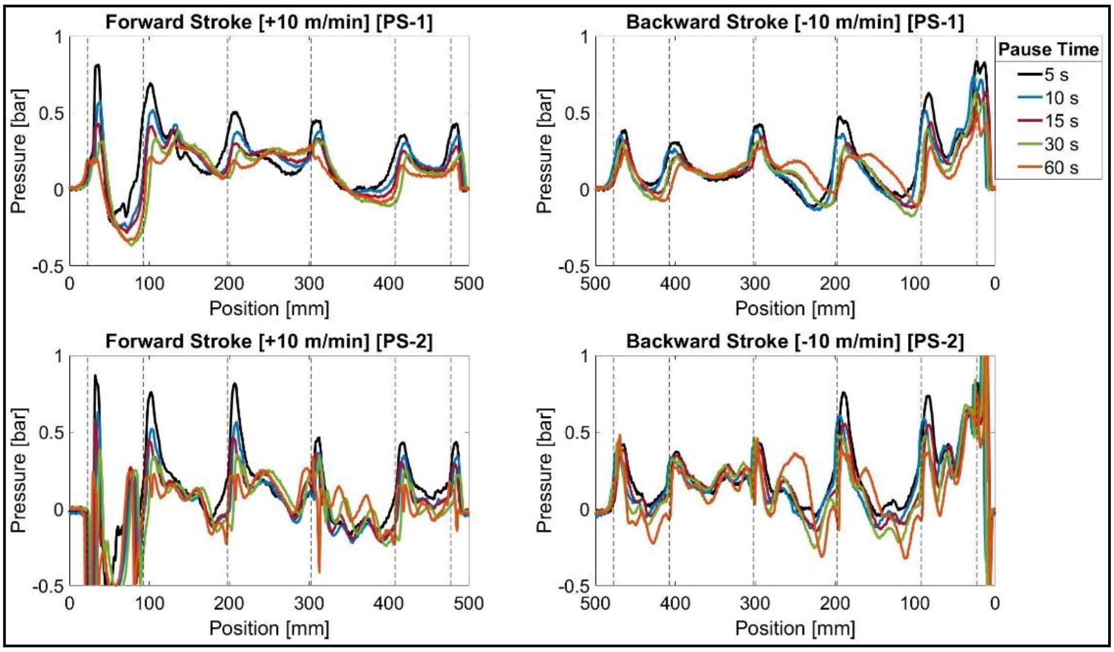

Fig. A4. The pressure peaks (6 peaks for 6 lubrication grooves) measured for forward and backward strokes inside the lubrication gap using geometry variant II for different pause time at 10 m/min

The sensors were calibrated (static + dynamic) individually (independent of rail) which we also published in WGP 2022 and in Plexiglas rail. Later we again calibrated (static + dynamic) the sensors integrated in steel rail as shown in geometry variation II and in Fig. A1.

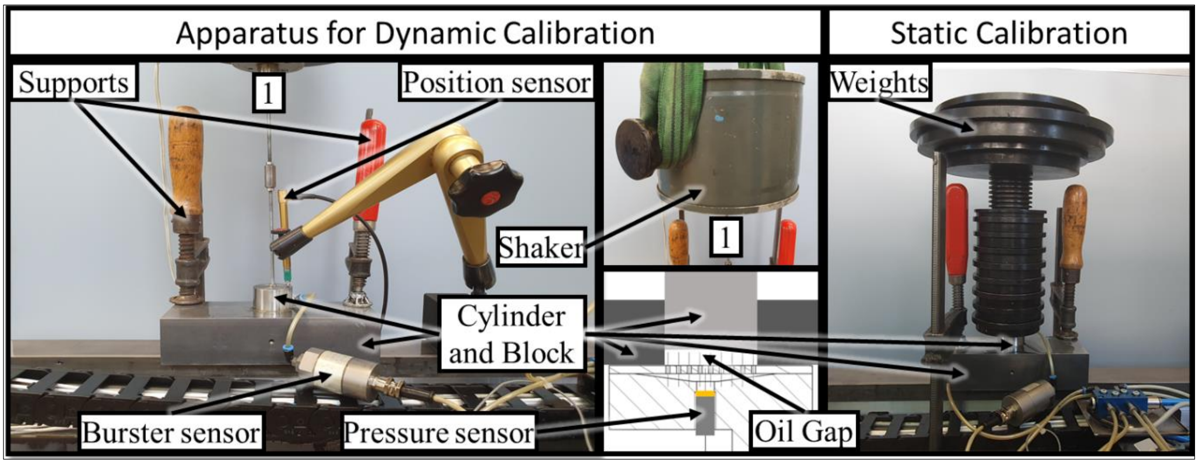

Fig. A5. Test stand for both static calibration (Right) and dynamic calibration (Left) of sensors integrated in steel rail

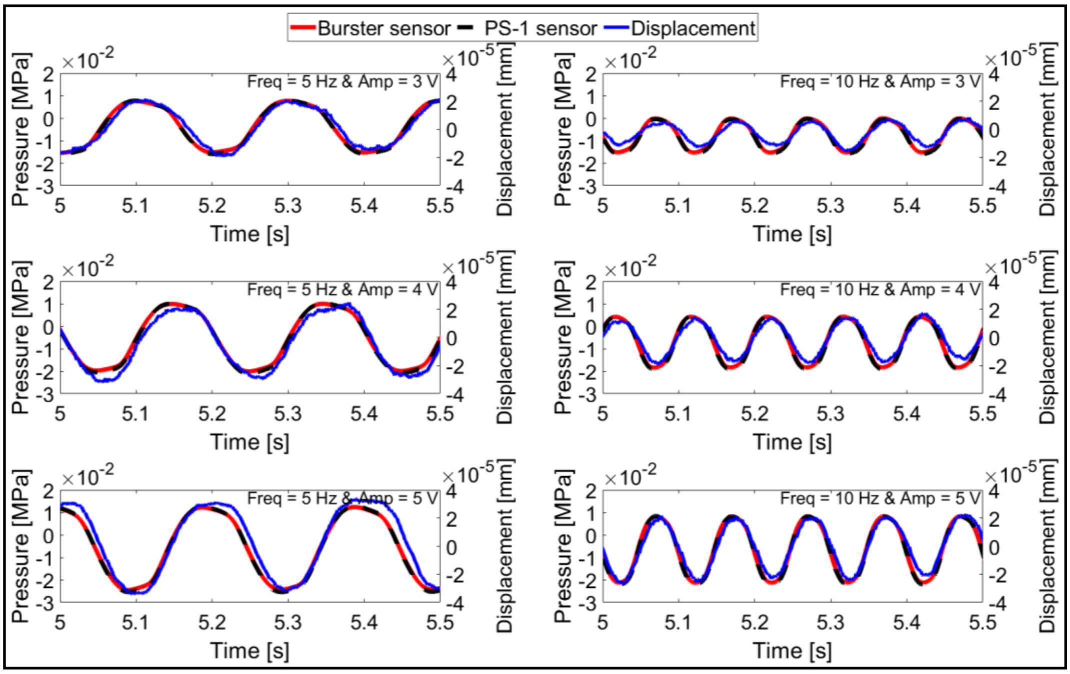

Fig. A6. Dynamic calibration of pressure sensor in steel rail and Burster sensor (for comparison) and displacement of cylinder (Experimental setup shown in Fig. A5)

For dynamic calibration, another sensor named Burster is also connected to the oil gap, which is independent of any integration method, which can be seen in Fig. A5. Both sensors (PS-1 and Burster) in dynamic calibration are in very good match, which proves that the sensor integrated in steel rail with nine small holes work. The sensors were calibrated for frequencies 1 to 25 Hz and three different amplitudes and no difference has been seen between PS-1 and Burster as shown in Fig. A6.

Regarding the concern about thin film sensors described in [11 and 12], they used those sensors to measure very high pressure (+10 MPa) but in our application the pressure is very small (static surface pressure < 0.1 MPa) so using thin film sensors is not appropriate to measure oil film pressure along the length. Your concerns are valid but the results we measure are promising and different integration methods were tested to prove our statement.

The large volume with small nine holes is always filled with oil so it does not have any negative influence in pressure measurement. In addition, the sensors can only be installed in the stationary rail to measure oil film pressure along the length of the slides for both forward and backward strokes. The pressure measured with steel rail (results shown in Fig 10 in JME Paper) is also very promising and showed the pressure rise and fall for each gap between two lubrication grooves. Moreover, the results are in very good agreement with the contact pattern between sliding surface as shown in Fig. 9 (JME Paper).

Additionally, with the help of CFD and experiments, it has also been validated that which size of small holes do not have any negative influence on oil film pressure measurement. Below different size of holes (1, 1.5 and 2 mm) have been compared by dropping an oil drop on top of the strip. The oil flow through small holes is shown after 3 seconds for all three diameters of small holes in Fig. A7. It is clear that the size of small holes have great influence on oil film pressure measurement and it should be carefully decided.

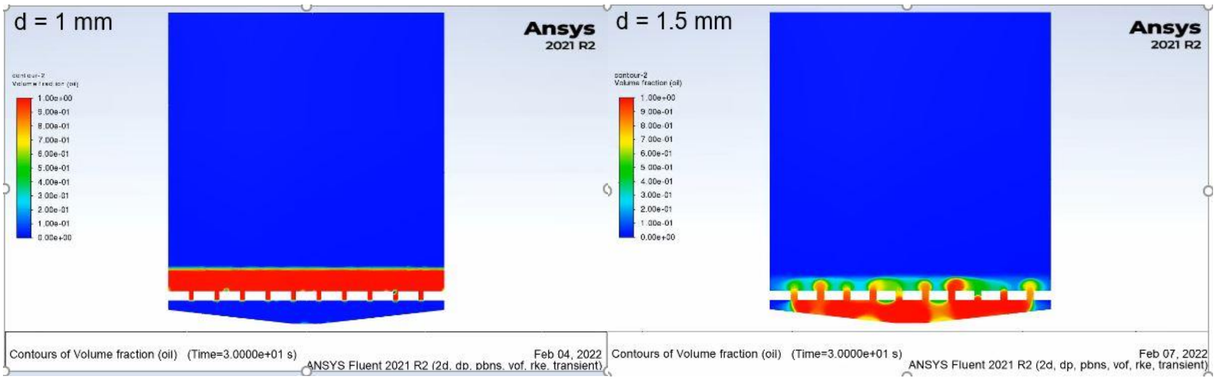

Fig. A7. CFD simulation of drop test with different small holes to see its influence