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PIV Study of Fluid Flow Inside A Gearbox

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ABSTRACT

For dip lubrication system, simple act of reducing the amount of lubricant inside the gearbox can reduce the load-independent losses but at the same time reduce the lifetime of the gear itself due to increasing bulk temperature [1]. Finding a proper lubricant level require understanding of the lubricant flow inside a gearbox. Therefore, 2D-2C PIV of a gearbox model has been done to study more detail the flow. The measurements were taken at three different lubricant levels (centreline, two module of gear, two module of pinion) and three different pitch line velocities (0.6, 1.1, and 1.62 m/s). Viscosity of the test oil (Nytex 810) is maintained at approximately 64 cSt by maintaining temperature at 20°C. The results suggest that flow inside a gearbox is complex, three dimensional and involves effects of rotation, inertia, free-surface dynamics, formation of bubbles and droplets, and also phenomenon of laminar-turbulent transition.

1. Introduction

One efficient way to transmit power mechanically is by using gears. Gears are well-known for their high efficiency capability in transmitting power. The efficiency of a single mesh gear can reach more than 90% [2], and for set of gears in the automatic gearbox, the total efficiency can reach 95% in its optimum condition [3]. In order to achieve its optimum working condition, gears has to be lubricated to reduce friction and dissipate heat.

There are several ways of making a lubrication system, but the most widely used in automotive application is a dip lubrication system. In dip lubrication, gears are put inside an enclosure (gearbox) and the gearbox is filled with lubricant until gears partially immersed into the lubricant. The immersion depth is determined by a balance between friction reduction and heat dissipation.

Friction reduction in gear is done by eliminating metal to metal contact. A thin layer of lubricant (a lubricant film), higher than surface roughness of the gear surface, is sufficient to eliminate metal to metal contact, hence reduce wear (elastohydrodynamic lubrication, see [4] and [5]). Unfortunately the lubricant film is affected by gear bulk temperature because viscosity of the lubricant is changing with changing temperature (viscosity is getting lower when temperature is getting higher). Low viscosity leads to thinning film thickness, hence increase the risk of metal to metal contact [1].

A large quantity of lubricant is good for heat dissipation however losses due to splash (churning losses) will be high [13]. On the other hand, a small quantity of lubricant will lead to high temperature and reduce lifetime of the gear [1]. Thus to find a proper lubricant level, one needs a better understanding of the lubricant flow inside a gearbox.

2. Method

Flow visualization technique was used as a method to preliminary study the flow inside a gearbox and oil distribution. After that study, some cases were chosen and studied further by using PIV to obtain the information on the flow velocity.

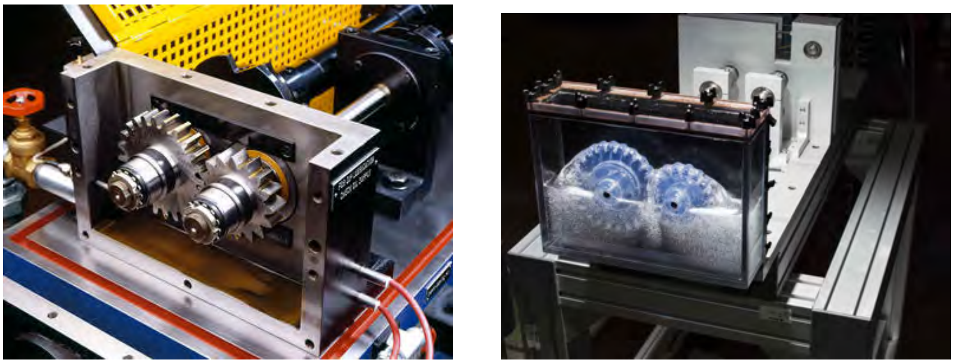
2.1 Experimental Rig

To be able to visualize the flow, a new experimental rig (Chalmers gearbox rig) with good optical access has been built at Chalmers. The rig design is based on the FZG back to back gear test rig. The FZG back to back gear test rig is one method closest to practice on predicting scuffing and wear properties of gear oils [7]. That is why the Chalmers gearbox rig design was based on this well established gear test rig.

The Chalmers gearbox rig basically a version of the FZG back to back gear test rig with maximized optical access. Some modifications were made in the rig. The contact between the gears was eliminated to remove a gear friction loss. The pinion was driven by a slave gear with belt. Five walls of the box and gears were made transparent to make it suitable for fluid flow study. Figure 1 shows both rigs side by side.

The Chalmers gearbox rig has an overall dimension 1000 mm in length by 600 mm in width and 1150 mm in height. The test section is made from 10 mm plastic with black anodized aluminium back wall. The inner dimension of the test section is 270 mm × 180 mm × 56 mm.

The test gears which were used are the FZG gears type C (see [7] for detail of FZG gear type C). The test lubricant was severely hydrotreated process oil, Nytex 810 (see [8] for detail information). This particular oil was chosen due to its optical properties. This oil



(a) FZG Test Rig (b) Chalmers Gearbox Rig

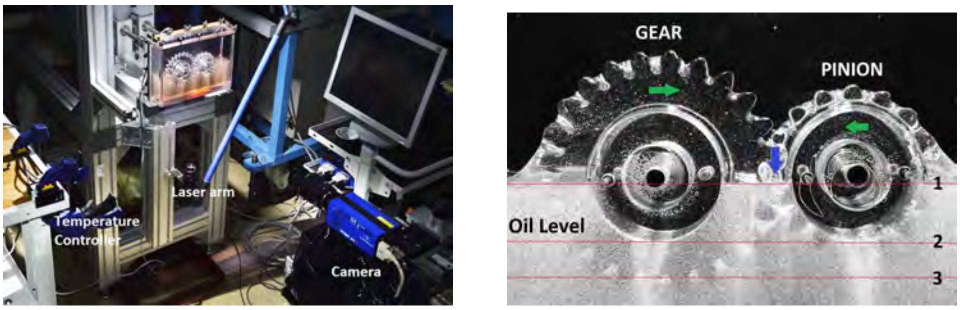
Figure 1: Comparison between the FZG gear test rig and the Chalmers gearbox rig

is transparent, colorless and has refractive index that matches the refractive index of PMMA. The viscosity of this oil at temperatures 20 °C corresponds to viscosity of Castrol Syntrans 75W-85 at 40 °C.

2.2 *Experimental Setup*

Figure 2a shows the experimental setup when doing 2D2C-PIV measurement. A CCD camera (Imager Pro X 4M, 2048×2048 pixel^2 resolution) was faced perpendicularly to the test section plane. The fields of view were 270 × 270 mm^2 (for overall flow) and 92 × 92 mm^2 (for corner flow). The laser sheet was introduced via the bottom wall of the test section. The illumination from below was found optimal because it did not get disturbed by splashes that were created when the gears were rotating. The flow was seeded with fluorescent particles (PMMA Rhodamine B) and the fluorescent light of particles is separated by low-pass filters mounted on the camera.

Figure 2b illustrates the cases that were studied. Three different oil levels and three different pitch line velocities (V_t) have been studied. Red lines and numbers indicates lubricant level that has been studied in this paper, which is centerline (1), two module of pinion (2), and two module of gear (3). Green arrows indicate rotational direction for both the gear and the pinion and blue arrow indicates the pitch line location. The pitchline velocities were 0.55, 1.1 and 1.62 m/s.



(a) Laser and camera setup (b) Case setup

Figure 2: Experimental setup

In this study, oil viscosity was approximately 64 cSt. It was achieved by maintaining the test section temperature at 20 °C so that the viscosity of the test oil match the viscosity of Castrol Syntrans 75W-85 at 40 °C (The Castrol Syntrans 75W-85 is the type of transmission oil that is approved by Volvo tTrucks [9]).

2.3 *PIV Processing Setup*

For each case 100 double frame images with external triggering were taken. Time difference between frames (dt) was adjusted for each pitch line velocity. The requirement was to keep dt small enough to allow particles to move maximum 5 pixels.

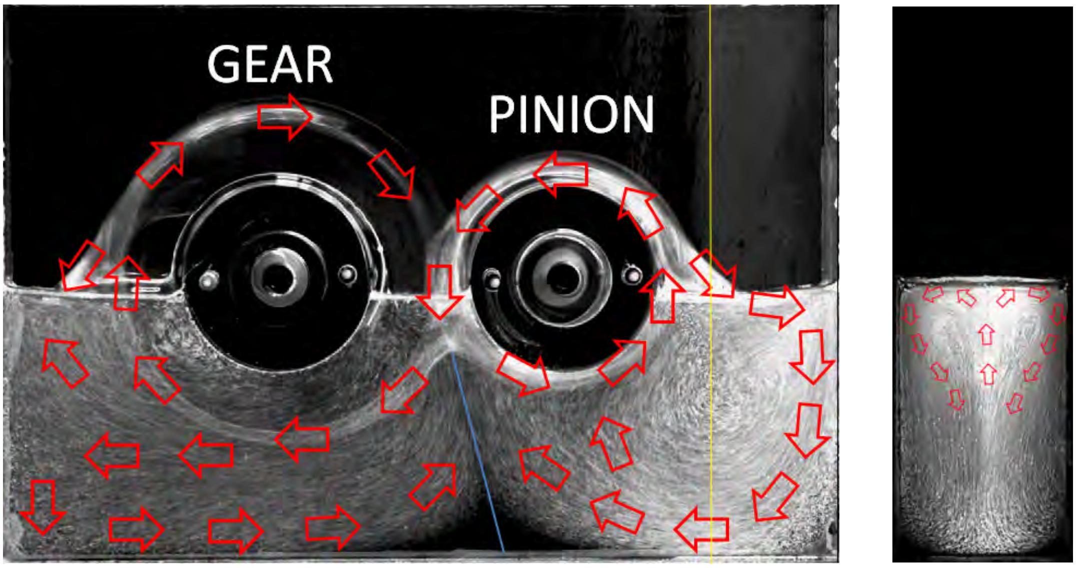
Resulting PIV images were masked geometrically, to remove some still objects and algorithmically, to remove reflections on the background. Multipass cross correlation with decreasing window size was used for processing the PIV images. An initial pass 64 × 64 interrogation window size with 50% overlap was used. The final pass was on 8 × 8 interrogation window size with 50% overlap and high accuracy mode.

3. Results and Discussion

The resulting images show that the flow inside a gearbox is complex and challenging to be studied. Visibility is the main issue, especially in high pitch line velocity, due to bubble formation and splashes that block the field of view.

3.1 Flow Visualization

Before studying the flow with PIV, a flow visualization study was done as a preliminary study. Figure 3 shows visualization of the flow. The images were taken at pitch line velocity, 0.35 m/s. At this low velocity the formed bubbles are small in size and the bubble concentration is low even at long running time. This means that the bubbles can be used as tracers that follow faithfully the flow. By using a long exposure image one is able to capture the trajectories of the bubbles.



(a) Front view (b) Cross view

Figure 3: Flow Visualization of test oil at 20°C, $V_t = 0.35$ m/s. Blue line emphasizes two separate regions created by the rotation of the gear and the pinion. Yellow line indicates the cross sectional region. Red arrows indicate the flow direction.

The visualization in figure 3a reveals that the rotating gear and pinion create two separate regions in the oil bath, one on the left and one on the right side. The flow in the bath has similarity with the lid-driven cavity flow (see [10] for more information about the lid-driven cavity flow). The rotating gear and pinion act like a moving lid in the lid-driven cavity case. Closer inspection of the flow below the pinion shows that there are three distinct recirculation regions. The first is the biggest recirculation region with the clockwise rotation seen in figure 3a. The two other recirculation regions are highly three-dimensional regions located near the pinion. The rotation shown in figure 3b occurs at the plane highlighted in figure 3a. The size of these recirculation regions and the main region are highly dependent on the flow regime. Particularly, in the considered case, for the gear the flow pattern is already different. Due to the larger radius and longer fluid path along the gear surface the gear produces thicker boundary layers and involves larger amount of fluid into the motion compare to the pinion. The thickened viscous layer results in increased size of the secondary recirculation regions and decreased size of the main recirculation region. In this case the flow is first approaching the left wall and then splits into the upward and downward moving flow.

In each of four bottom corners of the bath the stagnated flow regions are created. Another stagnated flow region can be seen is in the middle of the box, in place where the dividing flow streamline (highlighted by the blue line in figure 3a) is split up near the bottom wall.

There is no oil splash created in the top part of the box when the rotation speed is low. The oil which is dragged by the gear and the pinion leave the gear and the pinion smoothly as highlighted by arrows in figure 3a and enters the bath. The oil re-entry occurs to the left of the gear and to the right of the pinion.

The oil splash is created at higher speeds of rotation. To visualize how the oil distributions look at different operating conditions short-exposure images were taken as shown in figure 4. The flow visualizations were obtained for three different oil levels and velocity range $0.88 \leq Vt \leq 17.58$ m/s. Note that when these pictures were obtained the rig was equipped with an earlier version of the gear and the pinion manufactured with rapid-prototyping technique which is the reason for the different colour of the gear and the pinion on the pictures. The visualizations show that the influence of the oil level on the oil distribution is remarkable. For the oil at centreline level the splashed oil is approaching the top wall already at 1.76 m/s. At the speed of 5.28 m/s and above the oil distribution becomes fairly similar for all speeds and the splashed oil is distributed around the entire box. For 2 × module of gear case, the pinion is not touching the oil surface and does not contribute to the splash formation. As a result the splashed oil is occupying only the left side of the box. Remarkably, though, that the shape of the oil free surface formed by the splash on the left side of the box is nearly same as in case of centreline oil level. In case of 2 × module of pinion oil level the splash is considerably reduced and confined in the left bottom corner for all speeds. The amount of splashed oil appears nearly constant for speeds from 2.64 m/s and above.

The visualizations clearly show the limitations for optical methods. Visibility is the main issue due to bubble formation and splashes. With increasing pitch line velocity the visibility become poorer and transparency of the oil is greatly reduced due to a large amount of air bubbles. After just a few cycles of rotation the colour of the oil changes into a milky white. The bubbly oil combined with splashes, due to the fluid is expelled by the gear and the pinion, block the visibility. The expelled fluid hits the side and top walls and return making a fluid curtain covering the visibility from the front. Also some fluid that was dragged outside the oil bath by the gear and the pinion and hit the top wall creates raining oil from the top. This problem is the most severe for oil level at centreline.

3.2 PIV

Previous study concludes that there was a visibility issue associated with bubbles in the oil. To visualize the bubbles in the oil at different pitch line velocities, PIV images are shown in figure 5. For PIV the flow is seeded with fluorescent particles and the camera is equipped with a low-pass filter. As a result of the used approach the bubbles appear as contours when they reside in the laser plane. The bubbles located outside of the laser plane are illuminated by the diffuse light from the fluorescent particles. These out-of-plane bubbles are also out of the focal plane of the camera which results in typical unsharp images.

Closer inspection of figure 5a reveals that there are three types of bubbles. First are the large bubbles trapped in between the teeth of the gears. The second are medium bubbles which have size between 0.5 and 2 mm which are rather rarely distributed in the oil with concentration 1-10 bubbles per square centimetre. These bubbles can be clearly seen in front of the gear in figures 5b and 5c. The third are the small bubbles which have the largest concentration.

Comparing figure 5a, 5b and 5c, the conclusion are: all type of bubbles grows in quantity and size when the pitch line velocity increase. For $V_t = 0.55$ m/s (figure 5a), the visibility is good because the quantity of bubbles are in the non-disturbing level. For $V_t = 1.1$ and 1.62 m/s the quantity of medium bubbles are increase. These medium bubbles when passing through the light sheet leave thick shadow as seen in figure 5b and 5c.

Figure 6 shows the mean absolute velocity relative to the pitch line velocity at the centreline of the gear (figure 6a) and the pinion (figure 6b). The kink at the line indicates the changing direction of the flow (see figure 9 and 10 for better understanding of flow direction at the gear and the pinion side).

For gear at oil level at centreline, there are no counter flow, hence no recirculation region. All fluid at the mid-plane goes according to the rotational direction of the gear. At $V_t = 0.55$ m/s, the transition of the flow velocity is smooth. The line profile almost resemble Blausius laminar boundary layer profile. For gear at oil level 2 $\times$ module of pinion the counter flow is present. For oil level $2\times$ module of gear, the region of counter flow is the largest, but slowest. Notice the kink position at the plot, also see figure 9 for better understanding of flow direction at the gear side.

For pinion, the velocity at the centreline is dominated by counter flow. Notice the kink of the line plot are almost in the same location for every cases. This is because below the pinion there is more fluid compare to the gear, due to radius of the pinion is smaller. See figure 10 for better understanding of flow direction at the pinion side.

Figure 7 shows the overall mean velocity fields of the flow inside a gearbox. The figure of the flow above the oil level is considered as qualitative result. It is because in that region the PIV algorithm is applied to the oil drops not to the seeding particle, and the illumination intensity is not uniform.

For higher pitch line velocity case, most of the fluid is being expelled by the rotation of the gears. The higher the velocity the higher the splashes. The energy of the splash was dissipated when the fluid hit the side wall. In very high speed the energy is enough to make the fluid return and hit back the gears (see figure 4). Closer inspection of figure 7 shows that in high pitch line velocity there are a transition from viscosity dominated flow to inertial dominated flow. The movement of the splash can be described with ballistic parabolic trajectories [11].

Figures 9 and 10 show the closer inspection of mean velocity field at the corner of the gear and at the corner of the pinion. Note that for figures 10c, 10f, and 10i the pinion do not touch the oil bath. The movement that is seen in the figure is due to the flow from the gear side and some fluid that is dragged by the rotating gear being transfered to the pinion and hit the fluid free-surface at the pinion side. Notice the scale of those figures. The max scale value is just 10% of to the actual pitch line velocity for each cases.

The movement of the teeth exit the oil bath creates a void. This void is then filled by oil that has been dragged by the next teeth. This pushing action creates a flow that looks entering the bath as seen in figure 9c and 10b. The drops of oil that impinge the free-surface creates a void and later on creates an air bubble which continue impinging due to momentum [12]. This is explained why the higher the pitch line velocity the more bubble are formed. Because in higher pitch line velocity more void is created because the teeth hit the surface more often and more volume flow of the oil, which is lifted up to the air, impinge the free-surface.

4. Conclusion

Some conculsion that can be drawn from this study are :

- 2D2C PIV was successfully performed for capturing the flow inside a model gearbox. The resulting images show that the flow is complex, three dimensional and involves effects of rotation, inertia, free-surface dynamics, formation of bubbles and drops, and also phenomena of laminar-turbulent transition.
- Flow visualization study reveal that bubble formation and splashes in high pitch line velocity is severe and create visibility issue when performing flow visualization and PIV.
- The distribution of oil inside the Chalmers gearbox rig depends on the depth of immersion, and mainly governed by the gear. For centreline case, the oil distribution is quite equal between the left and the right side. For oil level at $2\times$ pinion and $2\times$ gear, the oil distribution are more on the gear side.

- The flow is dominated by inertial forces, as the pitch line velocity increase.
- For centreline case, at $V_t = 0.55$, 1.1, 1.62 m/s, there is no recirculation region.
- Bubbles are created due to void break up. The void can be created because the teeth enter the oil sump or the teeth exit the oil sump or the oil that drop and impinge the free-surface.

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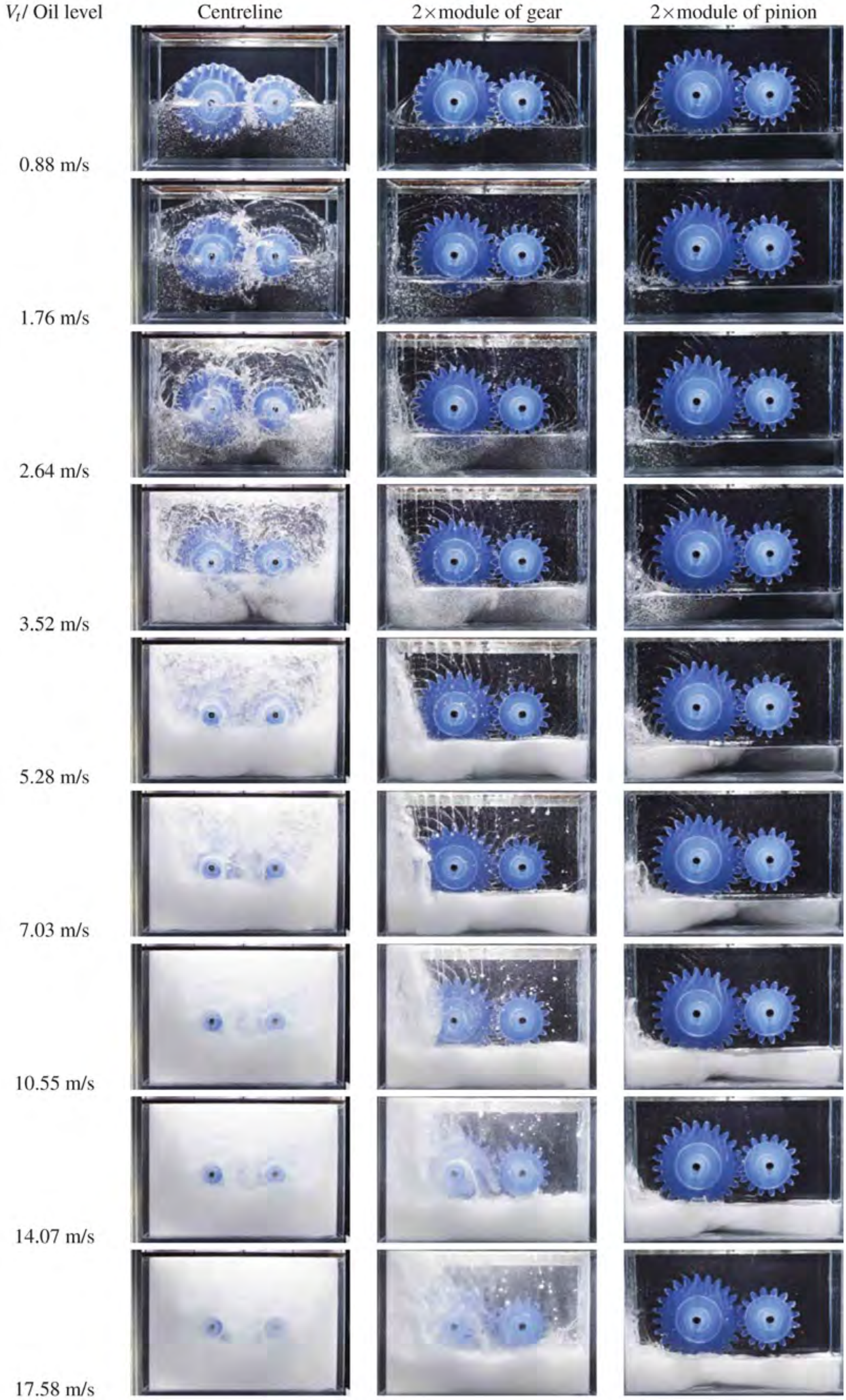
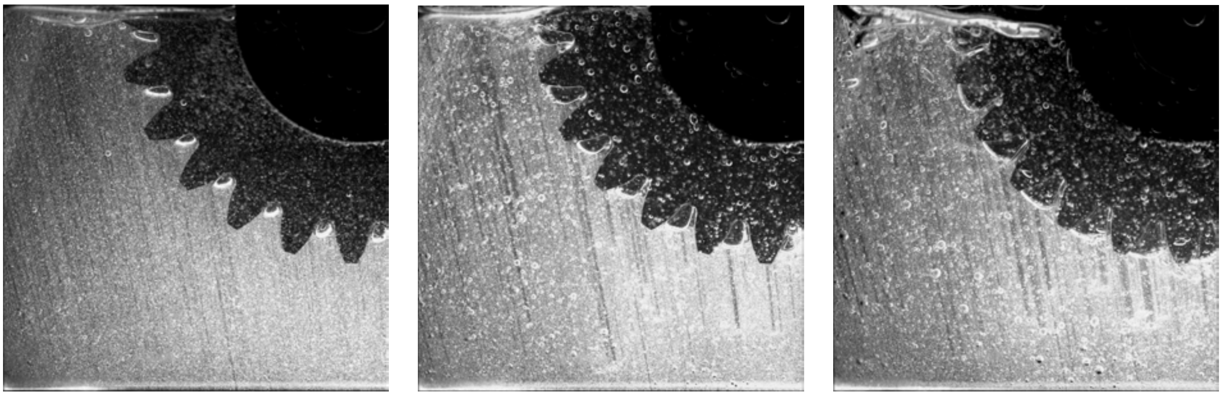
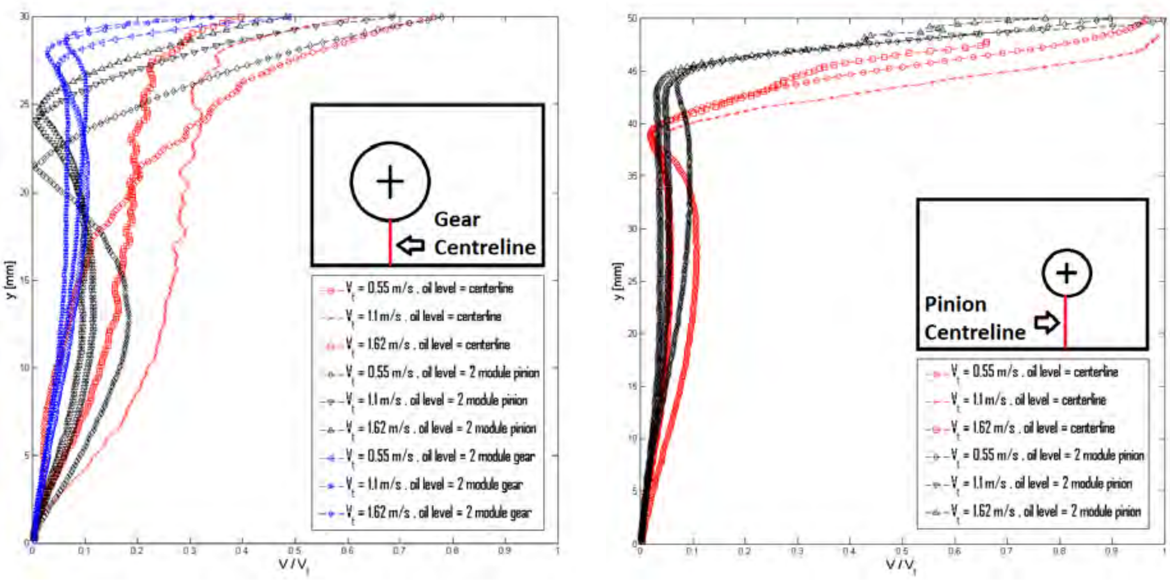


Figure 4: Collection of instantaneous image of oil flow inside a gearbox at room temperature



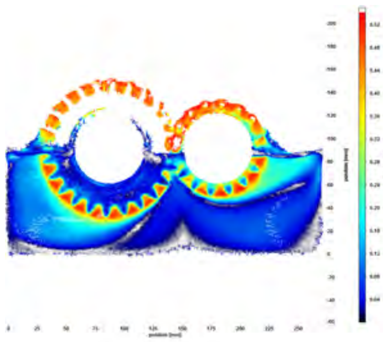
(a) $V_t = 0.55$ m/s (b) $V_t = 1.1$ m/s (c) $V_t = 1.62$ m/s

Figure 5: Comparison of bubble size and quantity around the gear. Note that for oil level 2× module gear, the pinion does not touch the oil bath.

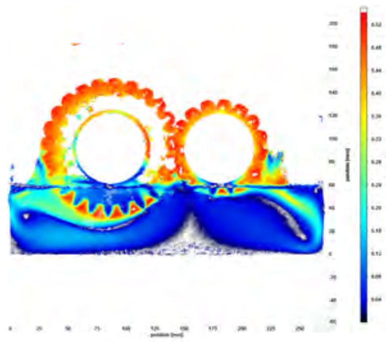


(a) Mean velocity at gear centreline (b) Mean velocity at pinion centreline

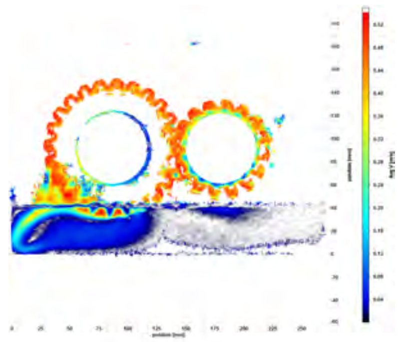
Figure 6: Comparison of absolute mean velocity at the centreline



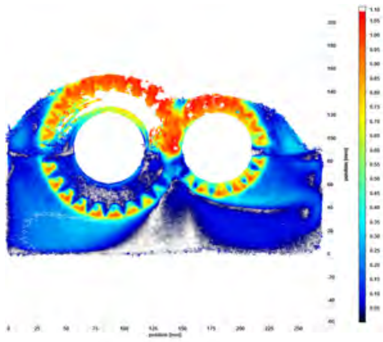
(a) $V_t = 0.55$ m/s ,
oil level = centerline



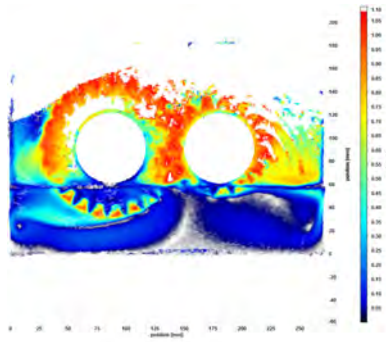
(b) $V_t = 0.55$ m/s ,
oil level = 2 $\times$ module of pinion



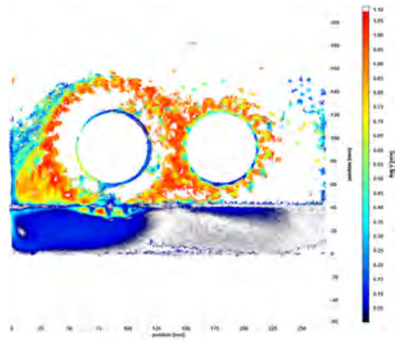
(c) $V_t = 0.55$ m/s ,
oil level = 2 $\times$ module of gear



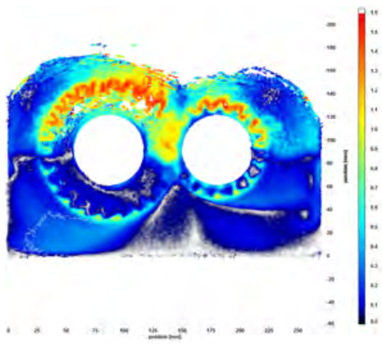
(d) $V_t = 1.1$ m/s ,
oil level = centerline



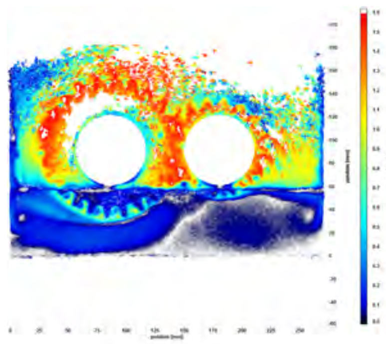
(e) $V_t = 1.1$ m/s ,
oil level = 2 $\times$ module of pinion



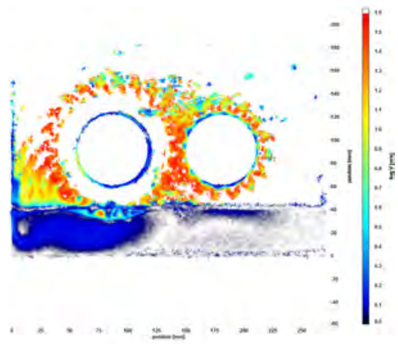
(f) $V_t = 1.1$ m/s ,
oil level = 2 $\times$ module of gear



(g) $V_t = 1.62$ m/s ,
oil level = centerline

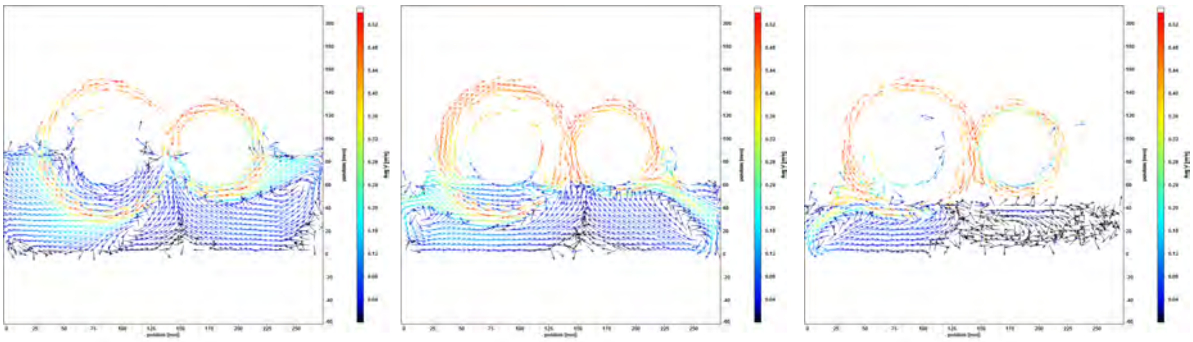


(h) $V_t = 1.62$ m/s ,
oil level = 2 $\times$ module of pinion



(i) $V_t = 1.62$ m/s ,
oil level = 2 $\times$ module of gear

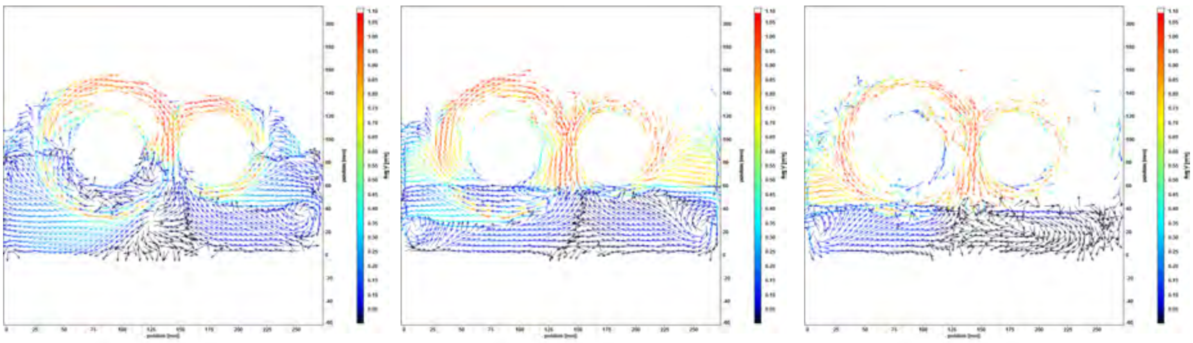
Figure 7: Overall mean velocity fields. Oil temperature at 20°C



(a) $V_t = 0.55$ m/s , oil level = centerline

(b) $V_t = 0.55$ m/s , oil level = 2 $\times$ module of pinion

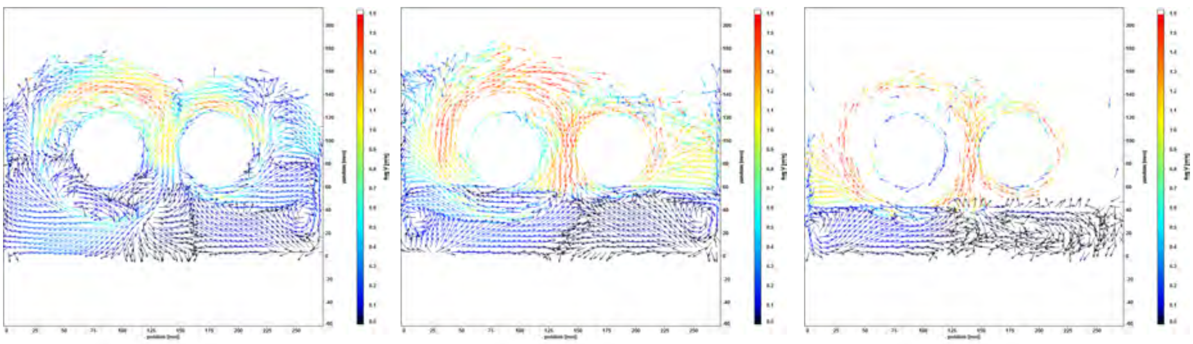
(c) $V_t = 0.55$ m/s , oil level = 2 $\times$ module of gear



(d) $V_t = 1.1$ m/s , oil level = centerline

(e) $V_t = 1.1$ m/s , oil level = 2 $\times$ module of pinion

(f) $V_t = 1.1$ m/s , oil level = 2 $\times$ module of gear

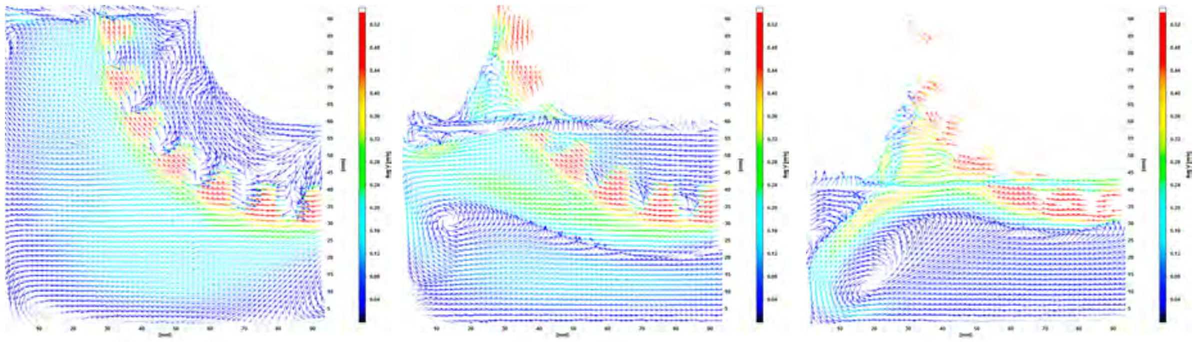


(g) $V_t = 1.62$ m/s , oil level = centerline

(h) $V_t = 1.62$ m/s , oil level = 2 $\times$ module of pinion

(i) $V_t = 1.62$ m/s , oil level = 2 $\times$ module of gear

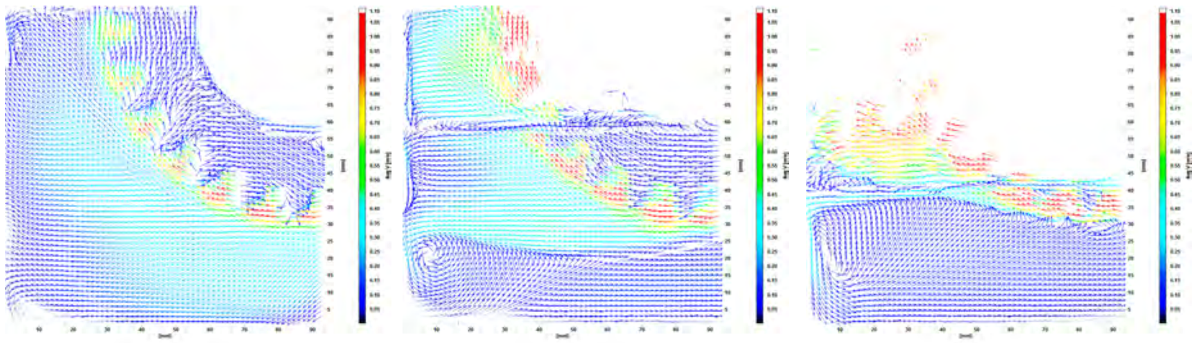
Figure 8: Overall mean velocity vector fields. Oil temperature at 20 °C



(a) $V_t = 0.55$ m/s ,
oil level = centerline

(b) $V_t = 0.55$ m/s ,
oil level = 2 × module of pinion

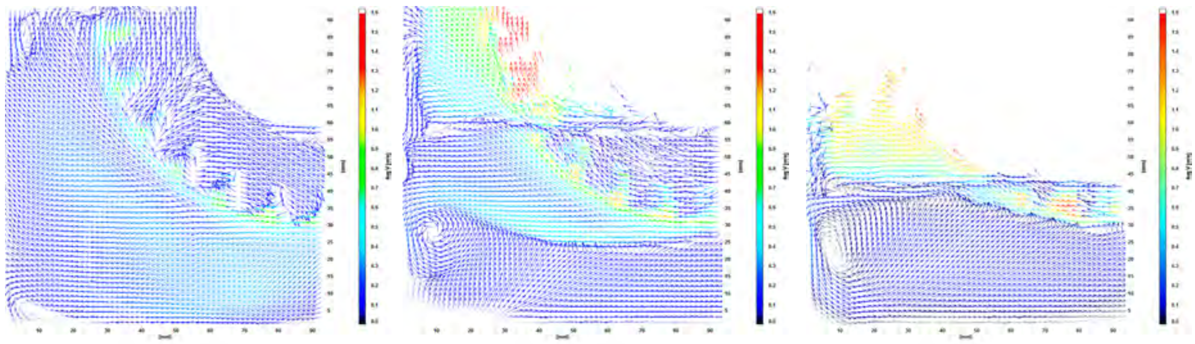
(c) $V_t = 0.55$ m/s ,
oil level = 2 × module of gear



(d) $V_t = 1.1$ m/s ,
oil level = centerline

(e) $V_t = 1.1$ m/s ,
oil level = 2 × module of pinion

(f) $V_t = 1.1$ m/s ,
oil level = 2 × module of gear

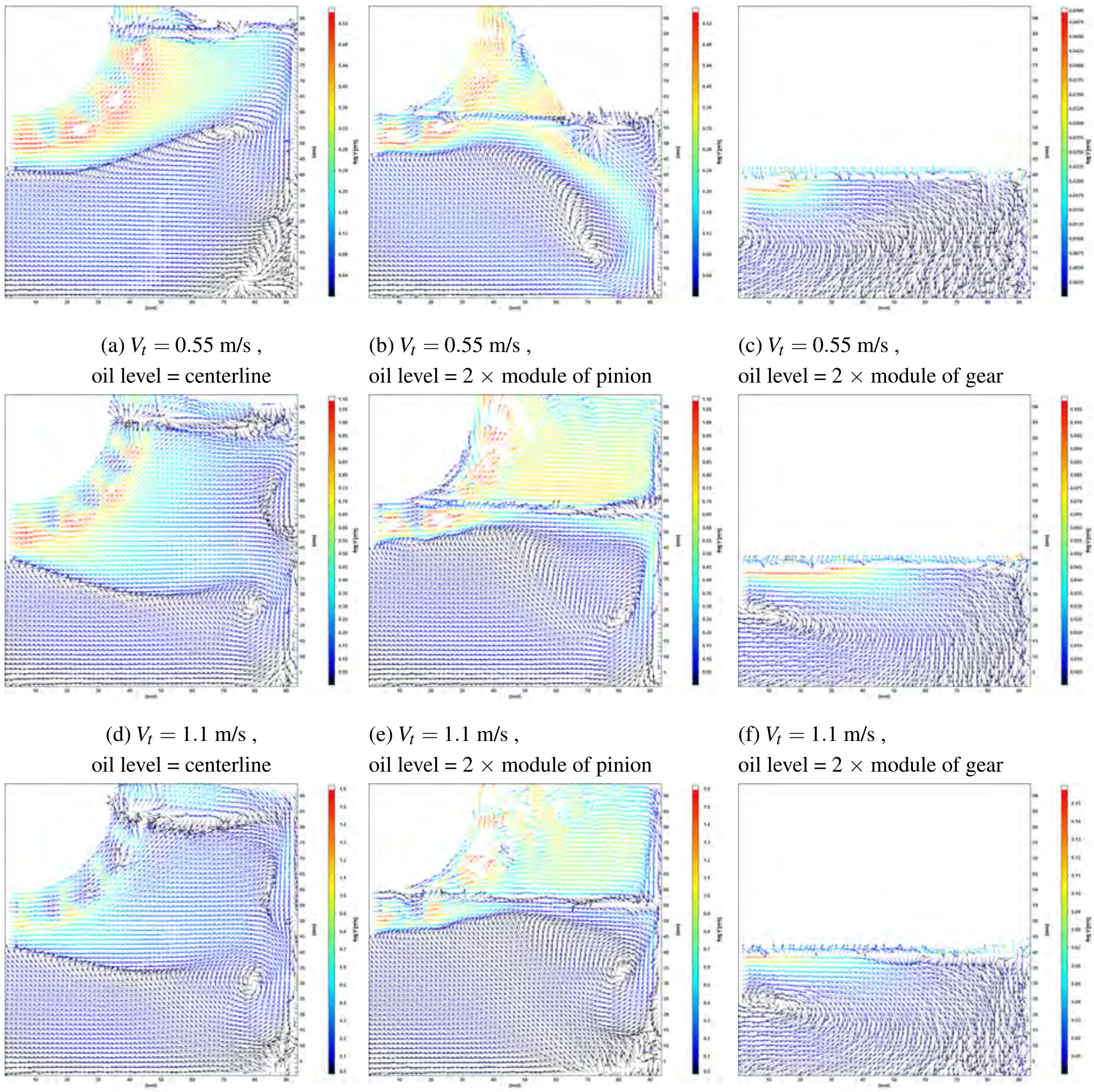


(g) $V_t = 1.62$ m/s ,
oil level = centerline

(h) $V_t = 1.62$ m/s ,
oil level = 2 × module of pinion

(i) $V_t = 1.62$ m/s ,
oil level = 2 × module of gear

Figure 9: Mean velocity vector field close to the gear. Oil temperature at 20°C



(a) $V_t = 0.55$ m/s , oil level = centerline

(b) $V_t = 0.55$ m/s , oil level = 2 $\times$ module of pinion

(c) $V_t = 0.55$ m/s , oil level = 2 $\times$ module of gear

(d) $V_t = 1.1$ m/s , oil level = centerline

(e) $V_t = 1.1$ m/s , oil level = 2 $\times$ module of pinion

(f) $V_t = 1.1$ m/s , oil level = 2 $\times$ module of gear

(g) $V_t = 1.62$ m/s , oil level = centerline

(h) $V_t = 1.62$ m/s , oil level = 2 $\times$ module of pinion

(i) $V_t = 1.62$ m/s , oil level = 2 $\times$ module of gear

Figure 10: Mean velocity vector field close to the pinion. Oil temperature at 20°C. Note that for oil level 2×module of gear, the pinion do not toucht the oil bath.