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Numerical Investigation of Coolant Distribution in Gear Skiving

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ABSTRACT

Epicyclic gear trains are a promising solution for increasing bypass ratios in modern turbofan jet engines. Gear skiving is a highly productive machining process for manufacturing the internal gears used in such gearboxes. It involves a rotating cutting tool and a rotating workpiece whose axes are oriented at a crossing angle, thereby generating gear teeth through continuous relative motion. Optimizing the skiving process enhances accuracy, tool life, and surface quality. However, the three-dimensional kinematics and interrupted cutting pose significant challenges for reliable coolant delivery into the cutting zone. The coolant is injected through a nozzle towards the cutting zone. Due to its interaction with the tool and the workpiece, the coolant jet breaks up and forms intricate flow structures and ligaments. This study presents an implicit incompressible Smoothed Particle Hydrodynamics (SPH) approach to simulate the free surface flow of the coolant distribution in gear skiving. The methodology enables a detailed insight into complex coolant flow structures. The commercial SPH solver PreonLab® is employed. A particle spacing of 50 μm was identified as suitable based on a resolution study. The numerical results are validated using high-speed imaging and subsequently applied in a parametric study investigating both nozzle position and coolant flow rate effects on tool wetting. A higher wetted fraction of the tool tip improves cooling performance and may reduce friction during chip formation. The results reveal that a nozzle position not directly aimed at the cutting zone leads to improved tool wetting compared to industry-wide established alignment. Additionally, increased coolant flow rates enhance wetting levels, while reduced flow rates show diminishing effects. These findings highlight the importance of investigating coolant flow in gear skiving and provide guidance for the development of optimized coolant supply strategies.

INTRODUCTION

The continuous pursuit of improved fuel efficiency is driving the development of advanced turbofan engine concepts. Increasing the bypass ratio improves propulsive efficiency by accelerating a larger massflow of air with a smaller jet velocity, thereby reducing specific fuel consumption (Cumpsty and Heyes, 2015; Mattingly, 2002; Farokhi, 2025). In conventional direct-drive turbofans, the fan and low-pressure turbine operate at identical rotational speeds, leading to unfavorable design compromises when increasing the bypass ratio (Kurzke, 2009). The geared turbofan architecture overcomes these constraints by placing an epicyclic reduction gearbox between the fan and low-pressure spool (Kurzke and Halliwell, 2018; Riegler and Bichlmaier, 2007). This enables optimized operating conditions for both components, improving overall engine efficiency and significantly reducing fuel consumption. For example, Rolls-Royce's UltraFan demonstrator engine is designed to achieve a 10 % fuel consumption reduction compared to the Trent XWB engine (Rolls-Royce, 2022). The planetary gearbox in this engine transmits over 50 MW of power. Even assuming a transmission efficiency of 99 %, approximately 500 kW would be dissipated as heat, highlighting the critical importance of effective thermal management in such high-power-density gearboxes. These demanding operating conditions impose stringent requirements on gear geometry precision and surface quality to ensure reliable long-term operation. Manufacturing these precision internal gears demands

advanced machining processes that can meet tight tolerances while maintaining high productivity. Gear skiving represents a key manufacturing technology for high-precision internal gear production, with growing importance in the aeronautics sector where demand for lightweight, efficient transmission systems continues to rise. Consequently, process optimization is essential to enhance manufacturing precision, extend tool life, and improve surface quality.

Gear skiving is characterized by complex three-dimensional kinematics involving synchronized rotations and axial feed motion (Radzevich, 2024). However, the intersecting axis geometry and interrupted cutting conditions create significant challenges for effective coolant delivery to the cutting zone. The coolant jet must penetrate the confined gap between rotating tool and workpiece while avoiding deflection (Sauer et al., 2023). Insufficient coolant penetration leads to elevated tool temperatures, accelerated wear, and deteriorated surface quality (Klocke et al., 2016). Despite its importance, coolant flow behavior in gear skiving remains poorly understood, with industrial manufacturing relying on empirical nozzle positioning.

This study employs Smoothed Particle Hydrodynamics (SPH) simulations to analyze coolant distribution in gear skiving. The objectives are to validate the numerical approach against high-speed imaging, quantify tool wetting for different nozzle positions, and provide guidance for optimized coolant supply strategies.

METHODOLOGY

In this section, the fundamentals of the SPH method are briefly introduced and the ideas behind the Implicit Incompressible SPH (I-ISPH) formulation and the single phase approach are presented. The simulations are conducted using the commercial SPH solver PreonLab®, which was recently applied to similar technical applications (García et al., 2025; Aguirre Bermudez et al., 2024). The suitability of the SPH method for the present investigation was demonstrated by Keller et al. (2019) and Haber et al. (2023). Furthermore, the geometry and kinematics of the gear skiving process are described.

SPH Formulation

SPH was originally developed for astrophysical simulations by Gingold and Monaghan (1977) and Lucy (1977), and was subsequently extended to free-surface flow problems by Monaghan (1994). As a meshless Lagrangian method, SPH discretizes the continuous fluid domain into a finite set of sampling points, represented by particles, which move through the computational domain. Each particle carries the local physical field quantities such as density, pressure, velocity, and potentially more thermodynamic properties. The temporal evolution of the flow field is obtained by solving the governing equations in Lagrangian form, whereby the particles are advected along their respective trajectories, inherently capturing the motion of discrete fluid elements without requiring an underlying mesh structure. The Lagrangian formulation provides inherent advantages for applications involving free-surface flows with large deformations, high-density ratio multiphase flows, and complex multibody interactions (Keller et al., 2016). Since coolant distribution in gear skiving exhibits these characteristics, SPH represents an appropriate numerical framework for the present investigation. The SPH formulation is founded upon the discretization of spatial field quantities and differential operators through application of the Dirac delta identity, which asserts that the convolution integral of an arbitrary continuous, compactly supported function $f(\vec{x})$ with the Dirac delta distribution $\delta(\vec{x} - \vec{x}')$ reproduces the function identically:

$$f(\vec{x}) = \int_{-\infty}^{\infty} f(\vec{x}') \delta(\vec{x} - \vec{x}') d\vec{x}' \quad (1)$$

In the SPH formulation, the exact integral representation is approximated through a two-stage discretization procedure. First, the Dirac- δ distribution is replaced by a smooth kernel function $W(\vec{x} - \vec{x}', h)$, yielding the continuous kernel approximation:

$$f(\vec{x}) \approx \int_{\Omega} f(\vec{x}') W(\vec{x} - \vec{x}', h) d\vec{x}' \quad (2)$$

Equation (2) represents the field quantity f at location $\vec{x}$ as a weighted integral over the neighboring spatial domain Ω , where the kernel function W provides distance-based weighting according to the distance vector $\vec{x} - \vec{x}'$. To ensure computational efficiency, the kernel is truncated at a finite radius of influence $r_{\max}$, beyond which the kernel value is zero. The smoothing length h controls the spatial resolution of the approximation. $r_{\max}$ is typically defined as a kernel-dependent multiple of h . The second stage of discretization involves replacing the continuous integral (2) with a summation over discrete particles

$$f(\vec{x}_i) \approx \sum_j f(\vec{x}_j) W(\vec{x}_i - \vec{x}_j, h) V_j \quad (3)$$

where the index i denotes the central particle at which the field quantity is evaluated, and the index j iterates over neighboring particles within the kernel support. The term V_j represents the volume associated with particle j . Spatial derivatives are approximated using the kernel gradient:

$$\vec{\nabla} f(\vec{x}_i) \approx \sum_j f(\vec{x}_j) \vec{\nabla} W(\vec{x}_i - \vec{x}_j, h) V_j \quad (4)$$

This formulation simplifies the computation of spatial derivatives, as only the analytically available kernel gradient is required rather than the gradient of the discrete field quantities (Eq. (4)). For isothermal incompressible flows the set of governing equations consists of the continuity and momentum equation, which are discretized in their Lagrangian form

$$\frac{D\rho}{Dt} = -\rho \vec{\nabla} \cdot \vec{v} \quad (5)$$

and

$$\frac{D\vec{v}}{Dt} = -\frac{1}{\rho} \vec{\nabla} p + \frac{1}{\rho} \vec{\nabla} \cdot \boldsymbol{\tau} + \vec{f}_\sigma + \vec{f}_b \quad (6)$$

where ρ is the density, $\vec{v}$ is the velocity, p is the pressure, $\boldsymbol{\tau}$ is the viscous stress tensor, $\vec{f}_\sigma$ represents surface tension forces, and $\vec{f}_b$ denotes body forces such as gravity.

The simulations employ the I-ISPH solver implemented in PreonLab®. In this formulation, the pressure acts as a Lagrange multiplier enforcing the incompressibility constraint of Eq. (5). The I-ISPH method solves the Navier-Stokes equations through a prediction-correction strategy analogous to the projection method (Ihmsen et al., 2014). The momentum equation (Eq. (6)) is decomposed into pressure acceleration $\vec{a}_p = -\vec{\nabla} p / \rho$ and advective acceleration $\vec{a}_{adv}$, comprising viscous, surface tension, and body force terms.

In the prediction step, an intermediate velocity $\vec{v}^*$ is computed by advancing Eq. (6) without the pressure acceleration term:

$$\vec{v}_i^* = \vec{v}_i(t) + \Delta t \vec{a}_{adv,i}(t) \quad (7)$$

The corresponding predicted density ρ^* is then obtained by applying the SPH discretization of Eq. (5):

$$\rho_i^* \approx \rho_i(t) - \Delta t \sum_j m_j (\vec{v}_i^* - \vec{v}_j^*) \cdot \vec{\nabla} W(\vec{x}_i - \vec{x}_j, h) \quad (8)$$

where m_j denotes the mass of particle j . In general the predicted density ρ^* is neither constant nor equal to the reference density ρ_0 , which violates the incompressibility constraint. Therefore, in the correction step, the pressure field is determined by solving a Pressure Poisson Equation

$$\nabla^2 p_i = \frac{\rho_0 - \rho_i^*}{\Delta t^2} = \frac{\rho_i}{\Delta t} \vec{\nabla} \cdot \vec{v}_i^*, \quad (9)$$

which ensures that the velocity field resulting from the pressure acceleration $\vec{a}_p$ matches the predicted density change rate $\frac{\rho_0 - \rho^*}{\Delta t}$.

The Laplacian operator in the pressure Poisson equation is discretized using the SPH summation (Ihmsen et al., 2014; Aguirre Bermudez et al., 2024), yielding a linear system of equations for the pressure field at all particle positions. This sparse linear system is solved using iterative methods, with various solution schemes discussed in Koschier et al. (2020). The resulting pressure field is then used to compute the pressure acceleration

$$\vec{a}_{p,i} = -\frac{\vec{\nabla} p_i}{\rho_i} \approx -\sum_j m_j \left(\frac{p_i}{\rho_i^2} + \frac{p_j}{\rho_j^2} \right) \vec{\nabla} W(x_i - x_j, h) \quad (10)$$

and correct the predicted velocity to obtain the final velocity at the new time step. The viscous term in Eq. (6) is discretized using a combination of the viscosity models from Morris et al. (1997) and Monaghan (2005), accounting for both shear and bulk viscosity contributions. Following Morris et al. (1997), the viscous acceleration is approximated as:

$$\left[\left(\frac{1}{\rho} \vec{\nabla} \mu \vec{\nabla} \right) \vec{v} \right]_i = \sum_j \frac{m_j (\mu_i + \mu_j)}{\rho_i \rho_j} \frac{\vec{x}_{ij} \cdot \vec{\nabla} W(x_i - x_j, h)}{\vec{x}_{ij}^2 + 0.01 h^2} \vec{v}_{ij} \quad (11)$$

where $\vec{v}_{ij} = \vec{v}_i - \vec{v}_j$ is the relative velocity, $\vec{x}_{ij} = \vec{x}_i - \vec{x}_j$ is the relative position vector, and μ denotes the dynamic viscosity.

Surface tension forces are modeled following Akinici et al. (2012), combining molecular cohesion and surface curvature contributions. The total surface tension force acting on particle i is expressed as:

$$\vec{f}_{\sigma,i} = \sum_j K_{ij} \left(\vec{f}_{coh} + \vec{f}_{curv} \right) \quad (12)$$

where K_{ij} is a symmetrized correction factor accounting for the continuum representation of the fluid, preventing unphysical particle isolation:

$$K_{ij} = \frac{2\rho_0}{\rho_i + \rho_j} \quad (13)$$

Here, $\vec{f}_{coh}$ represents cohesive forces and $\vec{f}_{curv}$ accounts for surface area minimization.

The cohesive force contribution is computed as:

$$\vec{f}_{\text{coh},i} = \sum_j -\sigma m_i m_j C(|\vec{x}_{ij}|) \frac{\vec{x}_{ij}}{|\vec{x}_{ij}|} \quad (14)$$

where σ is the surface tension coefficient and $C(x)$ is a cohesion kernel function for 3D-SPH simulations (Tartakovsky and Meakin, 2005). To ensure surface area minimization, a curvature-driven force is added (Akinci et al., 2012):

$$\vec{f}_{\text{curv},i} = -\sigma m_i (\vec{n}_i - \vec{n}_j) \quad (15)$$

where the surface normal $\vec{n}_i$ is obtained from the gradient of the smoothed color field:

$$\vec{n}_i = h \sum_j \frac{m_j}{\rho_j} \vec{\nabla} W(x_i - x_j, h) \quad (16)$$

With the numerical framework established, the following subsection details the geometry and kinematics of the gear skiving process investigated in this study.

Geometry and Kinematics

As depicted in Fig. 1, the gear skiving setup consists of two main components: the cutting tool and the workpiece made of AISI 4140 steel. The chuck that clamps the workpiece as well as the housing of the machine are neglected in the further considerations. The tool has the geometry of an involute gear, while the workpiece is progressively machined from a plain ring into a finished internal gear. The cutting tool and the workpiece rotate simultaneously, with their axes oriented at a crossing angle. The relative motion between the tool and the workpiece generates the gear tooth gap through an interrupted cutting process. The process parameters are summarized in Tab. 1. The cutting tool rotates with rotational speed ω_T about

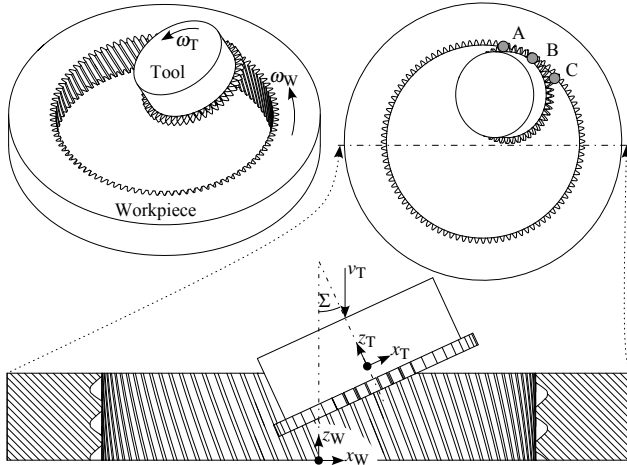


Figure 1 Geometry and Kinematics of Gear Skiving

Table 1 Process Parameters for Gear Skiving

Parameter	Value	Units
Workpiece		
Number of Teeth	96	
Normal Module	1.34	mm
Pitch Circle Diameter	128.64	mm
Height	4.0	cm
Rotational Speed ω_W	715.785	min^{-1}
Tool		
Number of Teeth	48	
Normal Module	1.34	mm
Pitch Circle Diameter	64.32	mm
Rotational Speed ω_T	1431.000	min^{-1}
Feed Velocity $v_{T,z}$	-0.301	mm min^{-1}
Axis Crossing Angle Σ	24.75	$^\circ$
Number of Cuts	10	

the tool axis z_T , while the workpiece rotates with rotational speed ω_W about the workpiece axis z_W . The angle between these two rotation axes defines the axis crossing angle Σ . Superimposed on the rotational motion, the tool undergoes a translational feed motion with velocity $v_{T,z}$ oriented parallel to the z_W direction. The complete tooth gap geometry is not generated in a single pass. Instead, the cutting process employs a multi-cut strategy, wherein the described kinematic cycle is repeated ten times with an incremental radial infeed applied before each cutting pass. The detailed modeling of chip formation lies beyond the scope of this investigation, which focuses on global coolant distribution. Consequently, a geometric simplification is implemented: chip material is not explicitly represented in the computational domain. This simplification is justified by the small chip dimensions and the small geometry change during a single cutting pass, which is on the order of the uncut chip thickness and therefore negligible compared to the overall geometry extents. Instead, the workpiece geometry corresponds to the final machined configuration following the complete cutting sequence. Hence, the cutting tool does not geometrically intersect the workpiece material, but establishes contact along the machined tooth flanks. Three nozzle configurations (A, B, C) are examined for coolant delivery to the cutting zone, as illustrated in Fig. 1. The nozzle position is fixed in the machine reference frame, with its central axis oriented parallel to the z_W -axis in all configurations. Configuration B aligns the nozzle axis directly with the tool-workpiece engagement zone, where the tool teeth mesh with the workpiece tooth gap. Configurations A and C are obtained by rotating the nozzle $\pm 20^\circ$ about the machine z_W -axis relative to configuration B. A volumetric coolant flow rate of $7.0 \text{ dm}^3 \text{ min}^{-1}$ is injected through a circular

nozzle with a diameter of 4 mm, positioned 30 mm above the workpiece upper surface. Additionally, to investigate the influence of coolant flow rate, two additional simulations are conducted for the reference nozzle position B with flow rates of $5.25 \text{ dm}^3 \text{ min}^{-1}$ (75% of reference) and $10.5 \text{ dm}^3 \text{ min}^{-1}$ (150% of reference). The physical properties of the coolant fluid employed in this investigation are summarized in Tab. 2. The fluid is a mineral oil-based coolant (Fuchs Ecocut 715 LE). Surface tension and dynamic contact angles (advancing and receding) were characterized using a Wilhelmy plate tensiometer (KRÜSS GmbH) according to the method described by [Wilhelmy \(1863\)](#). Contact angle measurements were conducted for the material pairing of AISI 4140 steel and Ecocut 715 LE at ambient temperature.

Table 2 Physical Properties of Fuchs Ecocut 715 LE at 20 °C

Property	Density	Kin. Viscosity	Surface Tension	Adv. Contact Angle	Rec. Contact Angle
Unit	kg m^{-3}	$\text{mm}^2 \text{ s}^{-1}$	mN m^{-1}	°	°
Value	890	42	29.0	35.2	27.2

Numerical Setup

The validity of the single-phase assumption is assessed through dimensionless analysis of two distinct flow regimes: the incoming jet and the oil film on rotating surfaces. For the jet-in-crossflow configuration, the oil jet exits the nozzle at approximately 9.3 m s^{-1} , while the crossflow air velocity induced by the rotating components is estimated at 4.8 m s^{-1} . This corresponds to a momentum flux ratio $J \approx 2.8 \times 10^3$, indicating that liquid jet inertia dominates over aerodynamic effects. The aerodynamic Weber number $We_g \approx 3.8$ remains below typical atomization thresholds, suggesting only minor surface deformation without droplet formation. Consequently, these parameters confirm weak aerodynamic coupling for the jet trajectory. On the rotating tooth flanks, the oil film exhibits a film Reynolds number of $Re_f \approx 0.8$, confirming viscous-dominated laminar flow. The capillary number $Ca \approx 1.3 \times 10^{-3}$ demonstrates that surface tension stabilizes the interface against deformation. To evaluate aerodynamic influence, the characteristic air velocity is estimated from the circumferential speed at the pitch diameter, yielding a gas-phase Weber number $We_g \approx 0.3$. This indicates that aerodynamic stresses remain insufficient to cause film breakup or stripping. Additionally, the density ratio $\rho_{\text{air}}/\rho_{\text{oil}} \approx 1.3 \times 10^{-3}$ confirms negligible gas-phase inertial effects. These scaling arguments establish that the oil film dynamics are governed by viscous and capillary forces, with aerodynamic coupling being weak. The single-phase free-surface SPH formulation is therefore justified. Fluid-solid interaction is modeled using a particle-based boundary treatment method. Both the cutting tool and workpiece are discretized with solid boundary particles, whose kinematics are prescribed according to the process parameters in Table 1. When a fluid particle enters the influence radius of a solid boundary, its pressure is mirrored on the solid surface. The applied no-slip boundary condition follows the approach proposed by [Hu and Adams \(2006\)](#), wherein virtual particles are generated inside the solid domain by mirroring fluid particles across the wall surface. These virtual particles inherit the volume, pressure, and viscosity of their corresponding fluid particles, ensuring consistent momentum exchange at the fluid-solid interface. The adhesion factor for setting the contact angle is determined based on [Aguirre Bermudez et al. \(2024\)](#) and [Wieth \(2018\)](#).

Coolant particles are injected through a circular inlet boundary with a parabolic velocity profile, consistent with the nozzle geometry and flow rate specified in the experiment. To limit computational cost, a cylindrical outlet boundary encloses the entire computational domain at a minimum radial distance of 15 cm from all solid bodies. This outlet removes particles that exit the region of interest, maintaining a manageable particle count throughout the simulation.

The initial condition is an empty domain, with coolant injection starting at $t = 0 \text{ s}$, because any apriori prescribed coolant film distribution would be arbitrary and not representative of the actual process. The simulation is advanced in time until a quasi-steady state of coolant distribution is reached. The time step is adaptively determined based on the Courant-Friedrichs-Lewy condition. Based on [DeWinter and Blok \(1974\)](#) the analytical expected average film thickness on the tool teeth is approximately $340 \text{ }\mu\text{m}$. A resolution study is conducted to identify a suitable particle spacing that can capture the key features of the coolant distribution while maintaining computational efficiency. A particle spacing of $50 \text{ }\mu\text{m}$ is selected, which provides sufficient resolution to resolve the film thickness and flow structures without excessive computational cost.

RESULTS AND DISCUSSION

This section presents the simulation results of coolant distribution in gear skiving using the SPH methodology described previously. First, a detailed description of the coolant flow patterns and temporal evolution is provided for the reference nozzle configuration B. Subsequently, the numerical predictions are validated against experimental high-speed imaging to establish the reliability of the computational approach. Finally, a parametric study comparing three nozzle positions (A, B, C) and three coolant flow rates is performed, with quantitative analysis of tool wetting to identify optimal coolant delivery strategies.

Description of Coolant Distribution

For the sake of comparability, the dimensionless physical time $t^+ = t \cdot \omega_T$ is introduced, which represents the number of tool rotations. Fig. 2 shows the temporal evolution of the coolant distribution for nozzle position B. The coolant jet impinges into the cutting zone at $t^+ \approx 0.1$, and starts to spread along the tool teeth and the workpiece tooth flanks (cf. Fig. 2a). The coolant is captured and distributed on the upper side of the workpiece ring, in its tooth gaps and on the tool teeth. Small amounts of the coolant jet are splashed off immediately in this early impingement phase, but the majority of the coolant is captured. As the bodies continue to rotate (cf. Fig. 2b, 2c), the wetting mechanism stays the same, but the wetted tool teeth start to fling off the coolant due to centrifugal forces, forming long stretched ligaments originating from the tool tips. Particularly, these ligaments are splashed onto the workpiece surface. The rest does not interact any further and is flung off into the surrounding domain. At $t^+ = 2.0$ (cf. Fig. 2d), the workpiece completes one full rotation, as it rotates at half the speed of the tool. Already wetted areas of the workpiece, especially the upper surface, are wetted again. The film on the upper surface of the workpiece is continuously stripped off because of the centrifugal forces. At $t^+ = 3.0$ (cf. Fig. 2f), first ligaments start to separate from the upper surface of the workpiece. An increasing amount of coolant is progressively flung off from both the tool and the workpiece into the surrounding domain. Around $t^+ = 5.0$ (cf. Fig. 2i), a quasi-steady state between coolant pickup and flinging off is reached. Even though the coolant distribution is locally transient, the overall distribution pattern remains constant. Compared with the timescale of the whole gear skiving process, which typically involves many revolutions of the tool and workpiece, the coolant distribution reaches a quasi-steady state relatively quickly. This suggests that the transient phase of coolant distribution is short-lived, and the system stabilizes to a consistent pattern of coolant coverage on the tool and workpiece surfaces after a few rotations. With the chosen particle spacing of $50 \mu\text{m}$, the total number of particles in the simulation domain stabilizes at around 1.4×10^8 for $t^+ \geq 5.0$.

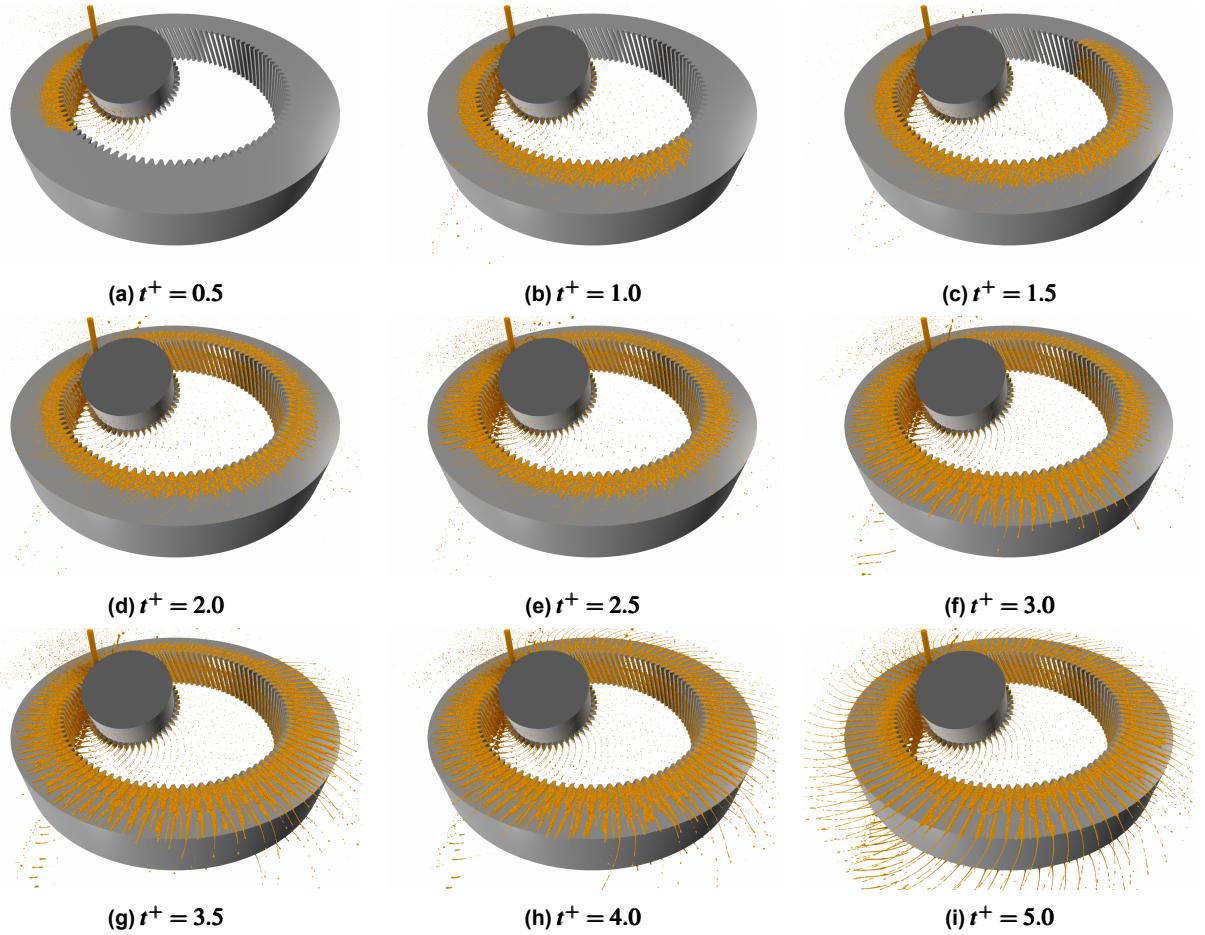


Figure 2 Temporal Evolution of Coolant Distribution for Nozzle Position B

Fig. 3 provides a detailed view of the coolant distribution at $t^+ = 6$, which is representative of the quasi-steady state. On the righthand side of the figure, the wetted areas of the tool teeth top surface are indicated. The tool rotates in counter-clockwise direction. Every tooth is wetted periodically as it passes through the meshing zone, where the tool teeth engage with the workpiece tooth gaps. The coolant covers nearly the entire surface of the tooth. During the rotation, the coolant film on the tooth is stripped off by centrifugal forces. Some parts of the tooth in proximity to its root are dried out. The tip of the teeth stay wetted for the whole rotation. This mechanism of periodic wetting, followed by centrifugal stripping

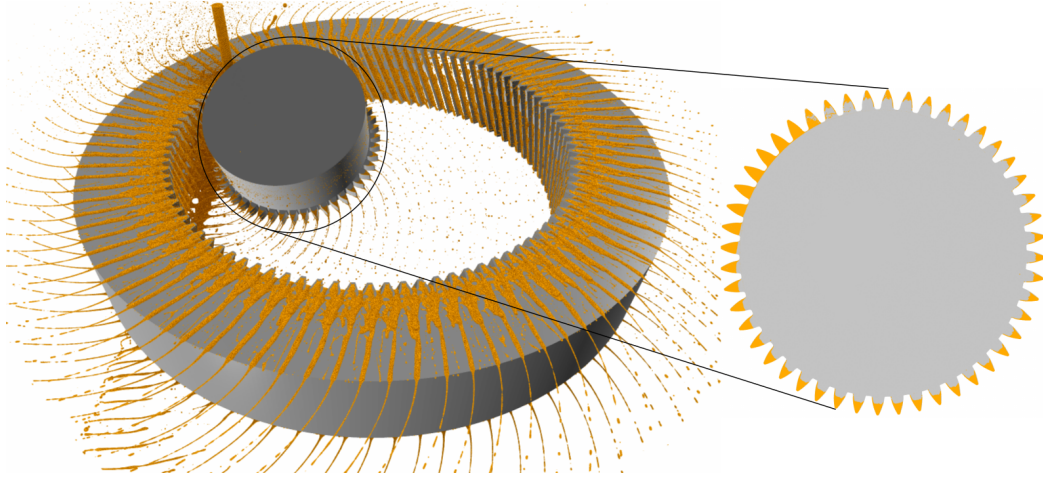
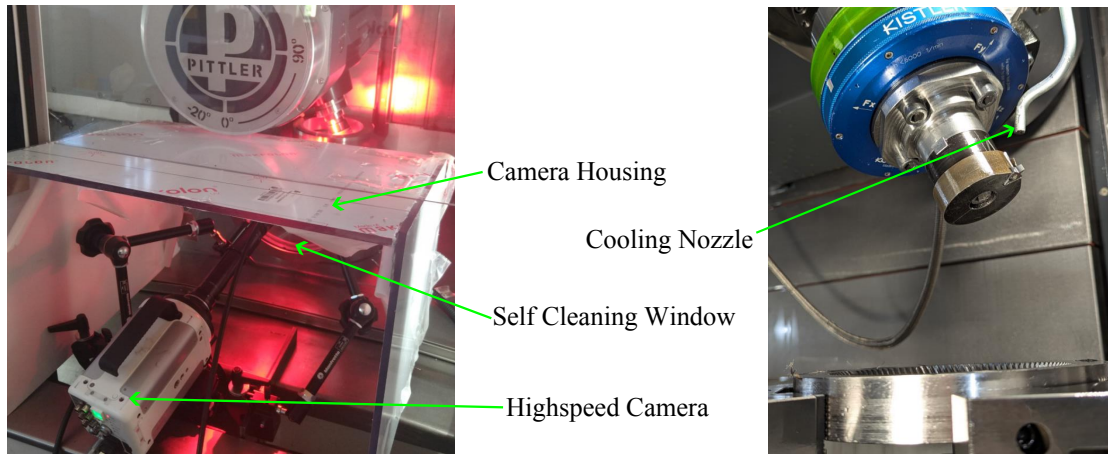


Figure 3 Coolant Distribution at $t^+ = 6$ for Nozzle Position B

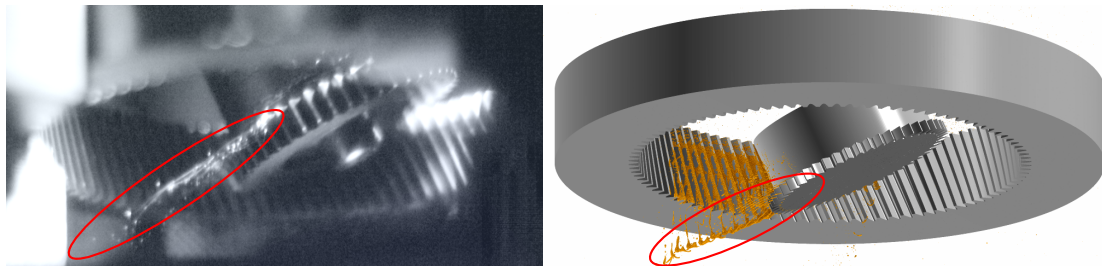
and flinging off, is described as fling off cooling of gears by [Kromer et al. \(2019\)](#) and contributes to the majority of its convective heat transfer. Consequently, to improve cooling performance, it is crucial to ensure that the coolant effectively wets the tooth surfaces.

Validation

For validation, the simulated coolant distribution for nozzle position B is compared with high-speed camera images from the experimental setup described by [García et al. \(2025\)](#). The presented simulations of the single tooth gear skiving showed good agreement with the experimental results. As shown in Fig. 4a, the optical access in the experiment is limited because the camera has to be protected from splashing coolant with its own housing. Also the space above the tool is too small for optimal imaging hence it is occupied by the shaft of the tool and cutting force measurement equipment. Consequently, the camera was placed under workpiece and the field of view is interrupted periodically by the three jaws of the chuck. Fig. 4b



(a) Experimental Setup ([García et al., 2025](#))



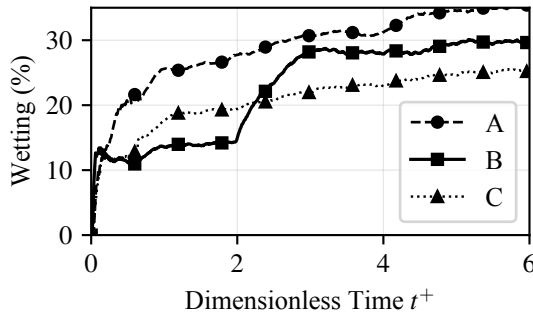
(b) Comparison of SPH Simulation and High-Speed Imaging at $t^+ = 0.5$

Figure 4 Validation of Coolant Distribution for Nozzle Configuration B

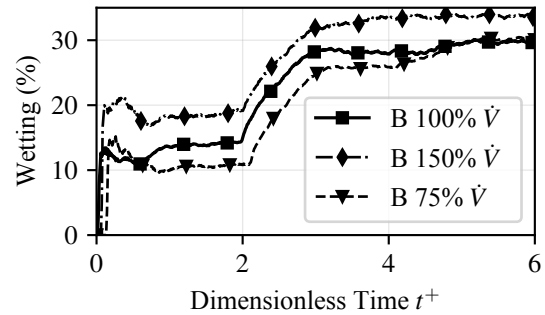
shows experimental setup and the coolant distribution at a dimensionless time of $t^+ = 0.5$, chosen because later stages are obscured by coolant flung off the rotating components. The image on the left highlights the pattern of coolant stripped off the tool teeth due to centrifugal forces, which is similarly captured in the simulation shown on the right. The trajectory angle of the coolant ligaments and the distance show good agreement. While the simulation clearly shows coolant on the workpiece surface, experimental observation of this region is limited by shadowing. Previous single-tooth gear skiving experiments [García et al. \(2025\)](#) confirmed the reliability of the simulation setup. High-speed images from later stages of the process suffer from increasingly severe shadowing and splashing, which limits the validation to early stages of the process. In contrast, the single tooth gear skiving setup of [García et al. \(2025\)](#) provided longer optical observation time because the amount of splashing coolant is much lower compared to the investigated setup. However, the good agreement in this initial phase provides confidence in the numerical approach for capturing the key features of coolant distribution in gear skiving.

Parametric Study

According to Fig. 1 three different nozzle positions (A, B, C) are investigated. As the surface wetting of the tool teeth is crucial for cooling performance, the wetting of the tool teeth over time is analyzed for the three nozzle positions. Fig. 5a shows the percentage of wetted tool teeth area over time for the three nozzle positions. The wetted area increases with time for all nozzle positions until it reaches a steady state. The results show that the reference nozzle position B, which is aligned with the tool-workpiece engagement zone does not necessarily lead to the best wetting of the tool teeth. Instead nozzle position A, which is rotated by 20° in the direction of workpiece rotation, leads to a more than 5 % higher wetting than position B. Position C, which is rotated by 20° against the direction of workpiece rotation, leads to a 3 % lower wetting than the reference position B. In absolute terms, the steady-state wetting levels are 35 % for position A, 30 % for the reference position B, and 27 % for position C. An interesting finding is that the pickup of coolant by the tool teeth from the filled tooth gaps of the workpiece during the meshing contributes significantly to the wetting of the tool teeth. Thus it can also be beneficial to bring more coolant onto the workpiece, as it happens in the case of nozzle position A. This observation holds as long as the majority of the coolant flowrate is captured by either the workpiece or the tool. If the nozzle is rotated too far, the coolant jet will partially miss the cutting zone and the overall wetting will decrease.



(a) Wetting of the Tool Teeth over Time for Different Nozzle Positions



(b) Wetting of the Tool Teeth over Time for Different Coolant Flow Rates

Figure 5 Wetting of the Tool Teeth over Time

In addition to nozzle position, the influence of coolant flow rate on tool wetting is investigated. For the reference nozzle position B, three flow rates are examined: the reference flow rate of $7.0 \text{ dm}^3 \text{ min}^{-1}$, a reduced flow rate of $5.25 \text{ dm}^3 \text{ min}^{-1}$ (75% of reference), and an increased flow rate of $10.5 \text{ dm}^3 \text{ min}^{-1}$ (150% of reference). The temporal evolution of tool wetting exhibits similar characteristics for all flow rates (cf. Figure 5b), with the wetted area increasing over time until reaching a quasi-steady state after approximately five tool rotations. However, the absolute wetting levels differ significantly. The steady-state wetting values are 30 % for the reference flow rate, 34 % for the increased flow rate (150 %), and 30 % for the reduced flow rate (75 %). The increased flow rate leads to substantially higher tool wetting throughout the entire time span, demonstrating that higher coolant flow rate enhances the ability to wet the tool surfaces. Conversely, the reduced flow rate shows no significant difference compared to the reference configuration after $t^+ > 5$.

Combining the findings from both parametric studies, it can be concluded, based on [Kromer et al. \(2019\)](#), that higher wetting leads to better cooling performance. Therefore, it is recommended to position the nozzle at position A in order to reach similar wetting results to those of an increased volume flow rate, while maintaining lower coolant consumption. The exact nozzle location, influence of different angles, and interaction with chip transport require further investigation in future work.

CONCLUSIONS

This study presents the first detailed numerical investigation of coolant distribution in gear skiving using an implicit incompressible Smoothed Particle Hydrodynamics approach. The SPH method successfully captures the complex three-dimensional flow structures arising from the interaction between the coolant jet and the rotating cutting tool and workpiece.

A particle spacing of 50 μm was identified as suitable through resolution studies, enabling accurate representation of the coolant film dynamics on the tooth surfaces. The numerical results show good agreement with high-speed imaging experiments, validating the capability of the SPH approach to predict coolant distribution patterns and ligament formation due to centrifugal forces.

The parametric study of three nozzle positions reveals a counterintuitive finding: the industry-standard alignment of the nozzle directly at the cutting zone (position B) does not provide optimal tool wetting. Instead, rotating the nozzle 20° in the direction of workpiece rotation (position A) increases tool tooth wetting by more than 5 % compared to the reference configuration. This improvement is attributed to enhanced coolant pickup from the workpiece tooth gaps during meshing, which contributes significantly to tool wetting. Conversely, rotating the nozzle against the workpiece rotation direction (position C) reduces wetting by 3 %.

The investigation of coolant flow rate effects for the reference nozzle position B demonstrates that increased flow rates (150% of reference) lead to higher tool wetting, indicating that higher coolant flow rate improves the ability to wet kinematically hidden areas of the tool. However, reduced flow rates (75% of reference) show no significant difference compared to the reference configuration after the quasi-steady state is reached.

The quasi-steady state analysis demonstrates that the coolant distribution stabilizes after approximately five tool rotations for all investigated configurations. The wetting mechanism follows a periodic pattern: initial impingement and spreading, followed by centrifugal stripping and ligament formation. Tool tips remain wetted throughout the rotation cycle, which is crucial for effective fling-off cooling.

These findings highlight the importance of investigating both nozzle positioning and flow rate effects in gear skiving coolant delivery. Improved tool wetting enhances cooling performance and may reduce friction during chip formation, ultimately contributing to extended tool life, improved surface quality, and higher manufacturing efficiency for precision internal gears used in geared turbofan engines. The results provide guidance for developing optimized coolant supply strategies that consider both geometric alignment and flow rate parameters.

NOMENCLATURE

Greek Symbols

δ	Dirac delta distribution
Δt	Time step, s
μ	Dynamic viscosity, Pa s
ρ	Density, kg m^{-3}
ρ_0	Reference density, kg m^{-3}
σ	Surface tension coefficient, N m^{-1}
Σ	Axis crossing angle, °
τ	Viscous stress tensor, Pa
ω	Rotational speed, min^{-1}
Ω	Spatial domain

Latin Symbols

$\vec{a}$	Acceleration vector, m s^{-2}
C	Cohesion kernel function
D/Dt	Material derivative
$\vec{f}$	Force vector, N
f	Field quantity/function
h	Smoothing length, m
K	Correction factor
m	Mass, kg
$\vec{n}$	Surface normal vector
p	Pressure, Pa
$r_{\max}$	Maximum radius of influence, m
t	Time, s
t^+	Dimensionless time (number of tool rotations)
$\vec{v}$	Velocity vector, m s^{-1}
V	Volume, m^3
W	Kernel function
$\vec{x}$	Position vector, m

Subscripts

0	Reference value
adv	Advective
b	Body forces
coh	Cohesion
curv	Curvature
i, j	Particle indices
ij	Relative quantity between particles i and j
max	Maximum
p	Pressure
T	Tool
W	Workpiece
z	z -direction
σ	Surface tension

Superscripts

+	Dimensionless quantity
*	Predicted/intermediate value

Dimensionless Numbers

Ca	Capillary number
J	Momentum flux ratio
Re	Reynolds number
We	Weber number

Abbreviations

AISI	American Iron and Steel Institute
I-ISPH	Implicit Incompressible Smoothed Particle Hydrodynamics
SPH	Smoothed Particle Hydrodynamics

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