Simulation-based twist drill design and geometry optimization

E. Abele¹ (1), M. Fujara¹

Institute of Production Management, Technology and Machine Tools
¹Technische Universität Darmstadt, Germany

Abstract

Designing a high-performance twist drill is difficult due to the complex relationship between drill geometry and numerous and conflicting design goals. Earlier approaches of computer-aided twist drill design are limited to only few design aspects. This article presents a new holistic method of using computing power for twist drill design and optimization. A complete drill geometry model is used to obtain drill performance characteristics and to ensure functional capability. Numerical simulation models calculate structural stiffness and stability, torque and thrust force, coolant flow resistance, chip evacuation capability and chip flute grindability. A multi-objective geometry optimization is realized by implementing metaheuristic optimization algorithms. As a result, a numerical overall optimization of twist drill performance is possible.

Keywords: Drilling, Tool Geometry, Optimization

1 Introduction

Even today, more that 200 years from the date of invention, the design of an efficient and powerful twist drill is still a very difficult task. The complex geometry compared to other types of cutting tools incorporates numerous conflicts of design objectives such as low cutting forces, wear resistance, torsional and axial stability, chip evacuation capability and more. Handling those interdependencies requires a high level of experience from the tool designer. Most approaches of simulation-based drill design focus on a limited number of geometrical aspects such as optimization of cross-section geometry or optimization of drill point grinding parameters. Chen et al. and Paul et al. investigated the relationship of a parametric twist drill point geometry and mechanical loads with the aim of cutting force reduction but without consideration of drill stability and chip evacuation [1, 2, 3]. Audy developed a method for computer-assisted drill point design with consideration of cutting forces for arbitrary cutting edge geometries [4]. Treanor et al. applied the finite element method (FEM) to twist drill stress analysis not for design but verification [5]. These approaches are limited to specific design aspects and do not propose new drill geometries automatically. Due to new twist drill applications, such as deep hole drilling, micro machining, high-speed and dry machining, an automated holistic drill design and optimization strategy becomes desirable. Jang et al., Chen et al., Schulz et al. and Abele et al. developed design and optimization strategies for multiple design aspects including a twist drill FEM analysis [6, 7, 8, 9, 10, 11]. These studies have limitations because of their simplified drill geometry models or too few optimization criteria. An effective computer-assisted twist drill design requires multiple conflicting optimization criteria to be incorporated simultaneously. Furthermore a drill geometry model is required to represent important design features of modern twist drills, such as reverse web taper and internal cooling channels which are absent in most previous approaches. All pre-existing drill design approaches lack of an integrated grindability verification, which is considered essential for cross-section design. In contrast to earlier approaches, this article describes a new approach of a holistic simulation-based method of twist drill design using multiple design objectives. A drill geometry model, capable of representing modern deep hole drilling tools with reverse web taper and multi-faced point grinding is used to determine the most relevant drill performance characteristics. For the determination of optimal drill geometry for given requirements, multi-objective optimization algorithms are implemented.

2 Methodology

2.1 Geometry Optimization Concept

The optimization architecture is designed for the application of evolutionary optimization algorithms (Fig. 1). A predefined drill geometry solution set, the start generation, is evaluated by all fitness functions and constraint functions. For the start generation, measured drill geometries from existing drills and new designs are used. It is important, that the start generation has a large variety of drill geometries. Various simulation models are used as fitness functions, such as finite element analysis of twist drill stiffness and stability, semi-empirc cutting force model for drilling torque and thrust prediction, analytical fluid dynamics for calculation of coolant flow resistance and estimation of chip evacuation capability. Constraint functions verify the validity and functional capability of drill design solutions, such as a chip flute grindability verification and compliance with size, orientation and location tolerances. While a fitness function returns a quantitative fitness value, a constraint function returns validity or invalidity of the drill solution. An overall fitness value is calculated from the return values by an evaluation scheme, for example with weighted sum method. Depending on the complexity of optimization task, other evaluation schemes such as Pareto-dominance-based rankings may become necessary.

![Fig. 1. Twist drill geometry optimization flowchart](image1)

Hereafter the optimization algorithm generates new solutions. Therefore, a Genetic Algorithm (GA) is used, but other evolutionary algorithms have also been proven suitable, for example evolution strategy algorithm or particle swarm optimization algorithm. In the next iteration loop step, the new solution set is evaluated again by all fitness functions and constraint functions. The iteration loop repeats until a stop criterion is fulfilled, for example the achievement of a specified fitness value.

2.2 Twist Drill Geometry Representation

For the efficiency of the optimization approach, the drill geometry representation is of great importance. All design attributes relevant for twist drill performance should be represented as complete and precise as possible. For the creation of all new drill geometries, a very versatile geometry model is preferable. Conversely, for the application of optimization heuristics to the drill geometry problem, a low number of design parameters is preferable.

![Fig. 2. Parametric model of twist drill cross-section profile](image2)

For the geometry model (Fig. 2), drill margin, body diameter and the transition between them are modeled as radiuses. The flute from...
body to margin or protection chamfer is a B-Spline of third order with 6 control points. Due to the parametric description, a change of few parameters does not compromise profile integrity. Only 5 parameters do a complete description from margin to heel. The flute can represent virtually any design by adjusting the location of point 4 and 5, optionally 3. Point 1 is always at heel, point 6 at margin. Point 2 is near 1 on a virtual extension of drill body diameter to cause the Spline to start tangential to drill body diameter. To model a web taper, a second and third cross-section model with some modification to the first but with same layout is implemented to represent the cross-section with tapered web close to shank. Very few parameters define the differences to the first cross-section (points 3-5 + drill body width). Points 1 and 2 are coupled to heel location. A facet point grading with 3 flank faces per lip is used. The faces are described as grinding plane normal vectors (Fig. 3). The third flank face additionally has a point thinning angle.

For improved dominance rankings and euclidean distances as a reference for selection probability, a Pareto-based multi-criteria optimization of the drill geometry is possible. As clamping constraint, all nodes at shank side are fixed.

For torsion load, two tangential forces equivalent to 

\[
M_{\text{f,torz}} = \frac{F \cdot l}{2}
\]

are applied at drill corners. For the bending load case, nodal forces are applied to drill point, equivalent to \( F_B = 1 \, N \). After computation of static response, nodal displacements and maximum principal stresses are saved to a text file for data import to Matlab.

For \( K_B \), nodal displacements at margin in two reference planes located with distance to loads and constraints are evaluated (Fig. 5). \( K_B \) is calculated from nodal displacement difference \( u_k - u_l \), margin radius \( r_{\text{margin}} \) and reference plane distance \( l_{\text{ref}} \).

\[
K_B = \frac{M_{\text{f,torz}}}{r_{\text{margin}}} = \frac{M_{\text{f,torz}}}{r_{\text{margin}}} = \frac{M_{\text{f,torz}}}{r_{\text{margin}}}
\]

The total torsional stiffness \( \tau = K_B^{-1} \text{drill} \) depends on drill length \( l_{\text{drill}} \). This includes misinterpretation cause by excessive stresses due to nodal force application or notching effects. The maximal principal stress \( \sigma_{\text{calc}} \) between reference planes divided by given torque \( M_{\text{calc}} \) is evaluated as torsional stress per torque \( K_{\sigma} \). The nominal drilling torque limit \( M_{\text{lim,calc}} \) can be calculated with knowledge of carbides tensile strength \( \sigma_{\text{max}} \).

\[
M_{\text{lim,calc}} = \sigma_{\text{max}} = \frac{M_{\text{lim,calc}}}{K_{\sigma}}
\]

The bending stiffness \( K_B \) is calculated from bending force \( F_B \) and maximal overall displacement \( \max(u) \).

\[
K_B = \frac{F_B}{\max(u)}
\]

It should be noted that because of the non-uniformity of twist drill cross-section and because of cross-section warping, analytical calculation methods using moments of inertia are not applicable for torsional stiffness and stress analysis. Exemplary calculations have proven that there is not even a reasonable correlation of second moment of inertia and torsional stiffness [5]. For the calculation of \( K_{\sigma}, K_B \) and \( K_{\text{lim,calc}} \) an automated twist drill FEM modeling and computation is built using Matlab and Ansys. First, an APDL script (Ansys parametric design language) is generated from Matlab scripts, containing modeling and computation commands as well as drill profile in polygons (variable along drills longitudinal axis). For the drill modeling in Ansys, polygon points are approximated using splines and joined by ruled surfaces. The drill body geometry is meshed using tetrahedron elements. As clamping constraint, all nodes at shank side are fixed.

For most parameters, additive mutation is used with exponentially distributed summands and parameter specific expectation values. For some parameters, flank face normal vectors for instance, adapted mutation operators with normal distribution within limits are used. Selection operator uses a roulette wheel algorithm. For multi-criteria optimization, a weighted sum evaluation scheme is used to calculate an overall fitness from single fitness values. This is easy to adjust and appropriate for optimization scenarios with few fitness functions. Scenarios with more than three fitness functions show a problematic behavior. A gain in one fitness function overcompensates losses in other fitness function, resulting insufficient controllability of the optimization. For example an excessive gain in torsional stiffness due to flute size reduction overcompensates a loss in chip evacuation capability. For improved multi-criteria optimization, Pareto-dominance-based advancements of the GA, MOGA and NSGA-II are implemented. Using non-dominance rankings and euclidean distances as a reference for selection probability, a Pareto-based multi-criteria optimization of the drill geometry is possible. As a result, the optimization is not seeking for a global optimum, but is developing a pareto-optimized solution set for various requirement profiles.

### 3 Fitness and Constraint functions

#### 3.1 Finite Element Analysis of Drill Stiffness and Stability

The drill body takes torque and thrust force loads. A high torsional stiffness \( K_B \) reduces both dynamic load and torsional chatter vibration. High bending stiffness \( K_B \) improves drill centering ability and avoids self-excited unbalance vibration of extra-long drills. The torsional stress per torque \( K_{\sigma} \) in conjunction with carbide tensile strength determines the maximal torque the drill may take. Some investigations show a correlation of tool life and torsional stiffness [5]. For the calculation of \( K_B, K_{\sigma} \) and \( K_B \), an automated twist drill FEM modeling and computation is built using Matlab and Ansys. First, an APDL script (Ansys parametric design language) is generated from Matlab scripts, containing modeling and computation commands as well as drill profile in polygons (variable along drills longitudinal axis). For the drill modeling in Ansys, polygon points are approximated using splines and joined by ruled surfaces. The drill body geometry is meshed using tetrahedron elements. As clamping constraint, all nodes at shank side are fixed.

For torsion load, two tangential forces equivalent to \( M_{f,torz} \) are applied at drill corners. For the bending load case, nodal forces are applied to drill point, equivalent to \( F_B = 1 \, N \). After computation of static response, nodal displacements and maximum principal stresses are saved to a text file for data import to Matlab.

For \( K_B \), nodal displacements at margin in two reference planes located with distance to loads and constraints are evaluated (Fig. 5). \( K_B \) is calculated from nodal displacement difference \( u_k - u_l \), margin radius \( r_{\text{margin}} \) and reference plane distance \( l_{\text{ref}} \).

\[
K_B = \frac{M_{\text{f,torz}}}{r_{\text{margin}}} = \frac{M_{\text{f,torz}}}{r_{\text{margin}}} = \frac{M_{\text{f,torz}}}{r_{\text{margin}}}
\]

The total torsional stiffness \( \tau = K_B^{-1} \text{drill} \) depends on drill length \( l_{\text{drill}} \). This excludes misinterpretation cause by excessive stresses due to nodal force application or notching effects. The maximal principal stress \( \sigma_{\text{calc}} \) between reference planes divided by given torque \( M_{\text{calc}} \) is evaluated as torsional stress per torque \( K_{\sigma} \). The nominal drilling torque limit \( M_{\text{lim,calc}} \) can be calculated with knowledge of carbides tensile strength \( \sigma_{\text{max}} \).

\[
M_{\text{lim,calc}} = \sigma_{\text{max}} = \frac{M_{\text{lim,calc}}}{K_{\sigma}}
\]

The bending stiffness \( K_B \) is calculated from bending force \( F_B \) and maximal overall displacement \( \max(u) \).

\[
K_B = \frac{F_B}{\max(u)}
\]

It should be noted that because of the non-uniformity of twist drill cross-section and because of cross-section warping, analytical calculation methods using moments of inertia are not applicable for torsional stiffness and stress analysis. Exemplary calculations have proven that there is not even a reasonable correlation of second moment of inertia and torsional stiffness [5]. For the calculation of \( K_B, K_{\sigma} \) and \( K_B \), an automated twist drill FEM modeling and computation is built using Matlab and Ansys. First, an APDL script (Ansys parametric design language) is generated from Matlab scripts, containing modeling and computation commands as well as drill profile in polygons (variable along drills longitudinal axis). For the drill modeling in Ansys, polygon points are approximated using splines and joined by ruled surfaces. The drill body geometry is meshed using tetrahedron elements. As clamping constraint, all nodes at shank side are fixed. For torsion load, two tangential forces equivalent to \( M_{f,torz} \) are applied at drill corners. For the bending load case, nodal forces are applied to drill point, equivalent to \( F_B = 1 \, N \). After computation of static response, nodal displacements and maximum principal stresses are saved to a text file for data import to Matlab.

The total torsional stiffness \( \tau = K_B^{-1} \text{drill} \) depends on drill length \( l_{\text{drill}} \). This excludes misinterpretation cause by excessive stresses due to nodal force application or notching effects. The maximal principal stress \( \sigma_{\text{calc}} \) between reference planes divided by given torque \( M_{\text{calc}} \) is evaluated as torsional stress per torque \( K_{\sigma} \). The nominal drilling torque limit \( M_{\text{lim,calc}} \) can be calculated with knowledge of carbides tensile strength \( \sigma_{\text{max}} \).

\[
M_{\text{lim,calc}} = \sigma_{\text{max}} = \frac{M_{\text{lim,calc}}}{K_{\sigma}}
\]

The bending stiffness \( K_B \) is calculated from bending force \( F_B \) and maximal overall displacement \( \max(u) \).

\[
K_B = \frac{F_B}{\max(u)}
\]
cutting edge is split into primary and secondary cutting edge. The secondary also incorporates the indentation zone around the dead center. The large variation of cutting conditions requires a discretization of cutting edges into ECTs (Fig. 6). For this modeling technique it is assumed, that all forces acting on the cutting edge can be expressed by two components, friction force $F_f$ in the rake face plane and normal force $F_n$ normal to the rake face.

Drill cutting edges are discretized by a number of ECTs (elemental cutting tools).

![Fig. 6. Elemental cutting tool (ECT), cutting angles](Image 66x623 to 301x733)

Friction force $F_f$ and normal force $F_n$ are assumed as proportional to the chip cross-section area $A_c$ of the ECT.

$$F_f = K_f \cdot A_c$$

$$F_n = K_n \cdot A_c$$

The specific normal force $K_n$ and specific friction force $K_f$ are different on each ECT. A power law models the relation of specific normal and friction force and cutting-related parameters.

$$K_n = a_n \cdot t_r^{a_n} \cdot v_{c}^{a_n} \cdot (1 - a_n)$$

$$K_f = b_v \cdot t_r^{b_v} \cdot \beta_v^{b_v} \cdot (1 - a_n)$$

The coefficients $a_n, a_v, b_v$ and $b_v$ are specific for tool and workpiece material and determined from calibration experiments with multivariate regression. For the calibration, spot drilling experiments are used with different cutting speeds and feed rates and full factorial design. When the drill engages the workpiece, the width of cut continuously increases and increments from thrust and torque signal are assigned to each ECT. $F_f$ and $F_n$ are calculated for each ECT by transformation. In a reverse direction, $F_f$ and $M_z$ can be calculated for arbitrary drill geometries from geometric data and coefficients $a_0, a_3$ and $b_0, b_3$. Verification experiments show that prediction accuracy is increased if two separate models for primary and secondary cutting edge are used. Calibration experiments were carried out with six different twist drill geometries. The variance of the coefficients $a_0, a_3$ and $b_0, b_3$ from these experiments was very low. This qualifies the cutting force model for prediction of $F_f$ and $M_z$ of new twist drill geometries. A weak point is the lack of a reliable cutting force model for thrust force from the indentation zone. Currently, this force is included in the secondary cutting edge force model. A significant extension of the indentation zone would result in thrust force estimation errors. To prevent this, a separate constraint function limits the indentation zone size.

### 3.3 Chip Evacuation Capability

An appropriate chip evacuation capability (CEC) is essential for twist drills. CEC is considered as good if chip removal does not lead to a significant torque increase while drilling depth increases. In practice, multiple parameters are known to influence CEC. Systematic investigations are difficult because of parameter interactions. Some parameters, such as exact chip forming, are still not predictable with adequate accuracy. A simple method for CEC prediction is the evaluation of flute cross-sectional area [8, 9, 10, 11]. Hands-on experience does acknowledge its relevance on CEC. However, this criterion neglects the influence of the flute shape. A more sophisticated method is the calculation of the maximum inscribed ellipse size in the flute cross-section [14]. An ellipse rather than a circle is used as spiral chip representation because of the drill twist. The cross-sectional shape of spiral chips is roughly circular in a projection normal to chip flow direction. Due to the helix angle of chip flow, it becomes an ellipse in a drill cross-sectional view (Fig. 7).

![Fig. 7. Iterative best-fit of the maximum inscribed ellipse to chip flute cross-section](Image 320x315 to 561x428)

In this assumption, the ellipses minor axis is aligned with the drill center point and contacts the hole wall at one and the flute at two points. Both criteria seem eligible for computer-aided geometry optimization and do well in test trials. However, they do only a very inaccurate and abstract CEC estimation [11]. Experimental investigations have been carried out to find a universal relation of CEC and a model-based criterion. Even though the general tendency that very small flutes are prone to chip clogging can be acknowledged, no general relation between the flute size and CEC could be found. As a result, the flute enlargement through reverse web tap has the most significant effect on CEC. Spiral chips, formed in a flute region close to the drill point are in a state of radial compression. A comparing of the spiral chip diameter with the best-fit ellipse diameter shows this phenomenon. A flute enlargement reduces mechanical transmission due to reduced friction between chips and the hole wall. Thus, it reduces torque increase. Even though also chip formation and chip breakage influence chip clogging, the chip clogging tendency is noticeably reduced. For an improved CEC criterion, chip expansibility due to flute enlargement is taken into account. In experiments short breaking spiral chips from thick web twist drills deviate from a circular shape. A better chip shape prediction was attained with a new chip forming model (Fig. 8). The actual chip shape depends onchip flute shape in chip flow direction. It is partially a copy of the chip flute curvature but has also smaller radiiueses.

![Fig. 8. Chip shape and expansibility in chip flute estimation](Image 320x694 to 561x806)

In step 1 two ellipses with an empirically determined radius are fitted to the upper and lower boundary. The drill center aligns with the ellipse’s minor axis. The ellipse’s sections from the flute to the hole wall contact point are considered as part of predicted chip shape. In step 2, the missing segments of the chip shape are determined from hole wall and chip flute. In step 3, the chip expansibility is determined as the maximum possible scale-up of the spiral chip shape. The chip shape prediction from step 1 and 2 is experimentally verified with 9 twist drills with different flute sizes and shapes. The maximal diameter prediction error was 11%. A qualitative CEC value is derived from the maximal possible scale-up percentage while an increase is valued within certain limits. Additionally, a minimum chip size threshold is set to avoid unacceptable small chip flutes. Typical flute enlargements are made by various grinding techniques. In a test series with different web tapers, those drills with a chip flute enlargement of at least 8% over 100 mm had a proper CEC, regardless of flute size.

### 3.4 Coolant Flow Resistance

An effective supply of aqueous cooling lubricants (abbreviated coolant) reduces thermal and mechanical load to cutting tools. Additionally, the coolant volume flow supports chip evacuation. Particularly for extra-long twist drills, adequate coolant flow resistance of twist drills coolant channels is critical for sufficient coolant supply. The common method to overcome flow resistance by high supply pump pressure is limited. Therefore, geometry optimization to reduce flow resistance is desirable. This fitness
function predicts the coolant flow $V$ for a given supply pressure $p_1$, with respect to drill length $l_{\text{Drill}}$, coolant channel diameter $d_{\text{CC}}$, distance from drill center to coolant channel center $r_{\text{CC}}$, twist length $l_{\text{twist}}$ and gap width from coolant channel outlet to hole bottom $w_c$. The flow speed is gauged by catching and weighing the fluid. Coolant density $\rho$ is determined by weighing a measuring pitcher. The comparison of flow speeds from water and coolant through a capillary tube determines the dynamic viscosity $\eta$. For the first investigation, the drill is considered to blow off freely without flow resistance from workpiece. The coolant channel flow is assumed as pipe flow with additional intake and outlet pressure drops. A pressure drop caused by channel warping is negligible low. Measurements on twist drills with $D=3.20 \text{ mm}$ and $L/D=3.30$ result in Reynolds numbers of 16,000 to 148,000, far beyond laminar flow threshold of 2,300. Therefore, the type of flow is turbulent. Roughness measurements show that the channel surfaces are hydraulic smooth without extra resistance from surface roughness. According to [15] the pipe flow coefficient $\lambda$ is

$$\lambda = \frac{0.3164}{\sqrt{Re}}$$

Bernoulli’s equation applied to the twist drill states (Fig. 9):

$$p_1 + \frac{\rho \cdot u_1^2}{2} = p_2 + \frac{\rho \cdot u_2^2}{2} + \Delta p_{\text{drill}}$$

$u_1$, $u_2$ refer to the flow speeds before intake, in the channel and after outlet (Fig. 9). Because $u_2^2 \gg u_1^2$, it is considered that $u_1^2 = 0$. For the flow speed after blow-off, a reference distance of $1 \cdot d_{\text{CC}}$ is used. Here $p_2$ equals the (negligible) atmospheric pressure and $u_2 = 0.625 \cdot u$ [16]. $\Delta p_{\text{drill}}$ is the total pressure drop.

Fig. 9. Flow resistance model

Accordingly, the simplified Bernoulli’s equation is

$$p_1 = \frac{\rho \cdot u^2}{2} + \Delta p_{\text{drill}}$$

$\Delta p_{\text{drill}}$ consists of the following pressure drop terms.

$$\Delta p_{\text{drill}} = \Delta p_{\text{inlet}} + \Delta p_{\text{CC}} + \Delta p_{\text{outlet}} = \zeta_{\text{inlet}} \cdot \frac{\rho \cdot u^2}{2} + \zeta_{\text{CC}} \cdot \frac{\rho \cdot u^2}{2} + \zeta_{\text{outlet}} \cdot \frac{\rho \cdot u^2}{2}$$

The intake pressure drop $\Delta p_{\text{inlet}}$ is described by a tube entrance with coefficient of $\zeta_{\text{inlet}} = 0.5$ [17]. The ratio of channel length $l_{\text{CC}}$ to channel diameter $d_{\text{CC}}$ determines the turbulent pipe flow pressure drop (Darcy’s law).

$$\Delta p_{\text{CC}} = \frac{\lambda \cdot l_{\text{CC}} \cdot \rho \cdot u^2}{2 \cdot d_{\text{CC}}} \text{ with } \zeta_{\text{CC}} = \frac{\lambda \cdot l_{\text{CC}}}{d_{\text{CC}}}$$

Assuming the outlet as a pipe enlargement the outlet pressure drop according to [18] is

$$\Delta p_{\text{Outlet}} = \frac{\rho \cdot u^2}{2} \cdot \left(1 - \frac{d_{\text{Outlet}}^2}{R^2}\right) \text{ with } A_2 = \infty : \Delta p_{\text{Outlet}} = \frac{\rho \cdot u^2}{2}$$

So the total pressure drop of the twist drill becomes

$$p_1(u) = \frac{u^2}{2} \left(\frac{1}{d_{\text{CC}}} + \frac{\lambda \cdot l_{\text{CC}}}{d_{\text{CC}}} \right)$$

The twisted coolant channel length $l_{\text{CC}}$ is

$$l_{\text{CC}} = \sqrt{\frac{2 \cdot \pi \cdot r_{\text{CC}} \cdot l_{\text{twist}}}{r_{\text{twist}}} + l_{\text{twist}}^2}$$

In summary, the required supply pressure $p_1$ for a given coolant flow $V$ is

$$p_1(u) = \frac{8 \cdot \rho \cdot V^2}{\pi^2 \cdot d_{\text{CC}}^2} \left(\frac{1}{64} + \frac{\left(\frac{2 \cdot \pi \cdot r_{\text{CC}} \cdot l_{\text{twist}}^2}{r_{\text{twist}}} + l_{\text{twist}}^2\right)^2}{d_{\text{CC}}^2} \cdot \frac{3.164}{\sqrt{R}}\right)$$

In validation experiments, the prediction error of flow rate was about 3.27%. In the actual drilling process, coolant outflow passes a gap-shaped narrowing between flank face and hole bottom. The additional flow resistance is of importance for tool design. Further calculations use a simplified cylindrical model of the outlet gap shape (Fig. 10).

Fig. 10. Coolant channel outlet gap, shape calculation and simplified model

A common formula for tube denting pressure losses determines the pressure drop $\Delta p_{\text{gap}}$.

$$\Delta p_{\text{gap}} = \frac{\rho \cdot u^2}{2} \cdot \left(\frac{\zeta_{\text{gap}}}{A_{\text{gap}}} - 1\right)^2$$

Because of an insufficient correlation with experiments, a new, empiric function is set up from experimental data.

$$\Delta p_{\text{gap}} = \zeta_{\text{gap}} \cdot \frac{\rho \cdot u^2}{2} \text{ with } \zeta_{\text{gap}} = f\left(\frac{A_{\text{gap}}}{A_{\text{CC}}}\right)$$

For this purpose, a face-grinded carbide round stock with coolant channels is positioned with gap distances over a baffle, measuring the flow rate. This gives a characteristic curve of flow resistance and gap width. It was determined that for $\frac{A_{\text{gap}}}{A_{\text{CC}}} > 2$ the pressure drop is negligible low (this equates $w_c = 0.5 \cdot d_{\text{CC}}$). Two ranges apply for values below this ratio.

For $0.2 \leq \frac{A_{\text{gap}}}{A_{\text{CC}}} \leq 0.6$: $\zeta_{\text{gap}} = -1052.9 \cdot \frac{A_{\text{gap}}}{A_{\text{CC}}} + 706.56$

For $0.6 \leq \frac{A_{\text{gap}}}{A_{\text{CC}}} \leq 2$: $\zeta_{\text{gap}} = -51.986 \cdot \frac{A_{\text{gap}}}{A_{\text{CC}}} + 404.36 \cdot \frac{A_{\text{gap}}^4}{A_{\text{CC}}^4} - 1,240.3 \cdot \frac{A_{\text{gap}}^3}{A_{\text{CC}}^3} + 1,877.8 \cdot \frac{A_{\text{gap}}^2}{A_{\text{CC}}^2} - 1,408.5 \cdot \frac{A_{\text{gap}}}{A_{\text{CC}}} + 422.66$

The gap shape is calculated and averaged with a geometry model in Matlab (Fig. 11). The neglect of a non-linearity regarding the gap width averaging leads to a pessimistic estimation of the flow rate.

Fig. 11. Actual outlet gap width calculation and linear averaging

It is important that in the drilling process the hole bottom is not a cone. The feed rate $f$ determines its obliqueness. The following equation describes the hole bottom shape in the drilling process:

$$f = \frac{\cos\alpha - \sin\alpha \cdot (\cos\alpha \cdot a + \sin\alpha \cdot b)}{\cos\alpha \cdot b + \sin\alpha \cdot a}$$

$\Delta p_{\text{gap}}$ is experimentally verified with a non-rotating drill in a predrilled blind hole. Because the finished hole bottom is a cone, shape deviations cause increased gap widths $w_c$ and hence decreased pressure drops $\Delta p_{\text{gap}}$. Further mechanisms such as centrifugal forces and two-phase flow in chip flute are very difficult to model and assumed to be of minor relevance for twist drill design. The final relation of $p_1$ and $V$ ends in this equation:

$$p_1(V) = \frac{8 \cdot \rho \cdot V^2}{\pi^2 \cdot d_{\text{CC}}^2} \left(\frac{1}{64} + \frac{\lambda \cdot l_{\text{CC}}}{d_{\text{CC}}} \cdot \frac{\zeta_{\text{gap}}}{2}\right)$$

The result is a coolant flow prediction with sufficient accuracy for twist drill design.

3.5 Chip Flute Grindability

A verification of manufacturability is essential for twist drill geometry optimization. The most critical part is the chip flute grinding in just one grinding wheel engage. Therefore, a constraint function is built, incorporating the approaches of Lin et al. [19, 20] and Hsieh [21]. It solves the “inverse problem”, using a polygon-based drill profile
Grindability is considered if the flute can be grinded with one engage of grinding wheel with respect to setting angle $\lambda_0$, inclination angle $\alpha_0$, grinding wheel radius $r_0$ and maximum grinding wheel width $w_{gr,\text{max}}$ (Fig 12).

For the calculation, grinding wheel is divided to a number of “discs” with infinitesimal small width, according to the basic idea of Lin et al. [20]. These discs contact the chip flute in one contact point (CP). In this ideal-theoretic view, the surface normal of the chip flute intersects the grinding wheel rotation axis at every CP (Fig. 12). Due to the discretization of the calculation, it misses with some skew distance. Therefore, an alternative criterion is used for CP determination. Sectional planes intersect twist drill profiles to make intersection curves (Fig. 13).

Each intersection curve contacts the grinding disc at the CP (Fig. 14). The CP is determined by the least distance to the grinding wheel axis plane. Any point on grinding wheel that concurs with the CP on the chip flute does not cause unintentional material removal somewhere else in that sectional plane.

The contact curve (connection of the CPs) is not coplanar and proceeds from heel to margin (Fig. 15). Using a fine discretization, the contact curve gets continuous with tolerable small jumps (Euclidean distance of two CPs). If the CP distance exceeds a certain threshold, the discretization resolution is refined. If the distance remains beyond the threshold, even if discretization is refined by a factor of $2^n$, an intersection problem is identified [19, 20, 22]. The result is an incomplete material removal, unable to grind the chip flute contour without contour violation somewhere else (Fig. 15). Hence, the chip flute is not grindable for given parameters.

For verification and fine-tuning purposes, the CPs are transformed into a drill cross-section plane. A comparison with an ideal twist drill profile shows little contour errors in regions with large CP distance due to discretization (Fig. 16). A model verification is conducted by Boolean operations with CAD models of the drill and grinding wheel (Fig. 17).

Therefore, a search strategy implementation for fast grindability checks five initial search points at first (Fig. 19).
Chip evacuation, torsional stiffness and stability, torque and coolant flow. The optimization algorithm is the classic Genetic Algorithm due to chip flute expansion as a chip evacuation capability to calculate the drill stiffness and stability. For an estimation of chip quality criteria for drill geometry solutions are implemented as gain in all criteria. Thus, the user is required to pick a solution that best fulfills the optimization goal. The optimization progress and an exemplary comparison of drill geometries before and after optimization is illustrated in Figure 20.

4 Exemplary Optimization Task

A twist drill geometry is optimized regarding 5 design objectives: Chip evacuation, torsional stiffness and stability, torque and coolant flow. The optimization algorithm is the classic Genetic Algorithm with a weighted sum scheme and iteratively adjusted weights. The optimization progress and an exemplary comparison of drill geometries before and after optimization is illustrated in Figure 20.

3.6 Further Constraint Functions

- Limitation of the effective (dynamic) clearance angle \( \eta \approx \tan^{-1} \frac{f}{2 \cdot r_{ECT} + \pi} \)

\( (\eta_s: \) static clearance angle, \( f: \) feed rate, \( r_{ECT}: \) ECT radius). Within the indentation zone, \( \eta_s \) is negative. To avoid excessive thrust force, the indentation zone size is limited by a \( \eta_s \) limitation.
- Flank face to hole bottom contact avoidance (in drilling process).
- Transition angle limitation from primary to secondary cutting edge, relevant for a predetermined chip breaking point formation.
- Chip flute profile diameter limitation (heel and web).
- Coolant channel position check (minimum distance to drill profile).

5 Conclusion and Future Research Activities

The application of Pareto-dominance-based multi-objective optimization algorithms such as MOGA and NSGA-II does not require criterion weights but does improve the initial solution set in various directions. For drill optimization, Pareto-based algorithms require much more computing time to achieve equivalent results as the weighted sum scheme. For a comparison of optimization efficiency, multi-objective particle swarm and simulated annealing algorithms shall be implemented.

- The twist drill geometry model that was used is a good trade-off between freedom of design and required number of design parameters (decision variables). However, such high shape variability requires a large number of solution evaluations for optimization convergence. A reduction in the number of decision variables may noticeably reduce computing time.
- Although fitness functions are validated separately, an experimental verification of the entire twist drill geometry optimization is pending. Therefore, commercially available twist drills will be optimized and tested against each other.

6 References


Fig. 19. Two-stage meander-shaped search path for valid grinding parameters

Fig. 20. Optimization progress and exemplary fitness comparison of drill solutions