

Hydraulic design of a grinding wheel with an internal cooling lubricant supply

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Abstract This paper presents a grinding wheel with an internal cooling lubricant supply (GIC). The cooling lubricant is supplied through channels inside the grinding wheel instead of conventional supply using external nozzles. With the GIC, direct coolant supply into the contact zone is possible. The hydraulic design of the GIC is similar to that of a centrifugal pump impeller. The design process is supported by simulation of fluid dynamics. First tests show that the GIC is able to supply the coolant at different flow rates with cutting speeds up to 60 m/s.

Keywords Grinding · Internal cooling · Fluid dynamics · Tool development · Coolant saving · Simulation

1 Introduction

Grinding requires the most energy to remove a specific volume of material compared to machining operations with defined cutting edges like for example milling, drilling or turning. The majority of this energy is converted into heat because of the internal and external friction. For this reason, thermal damage of the workpiece, i.e. alterations of

the surface layer, can occur. Grinding is often used as a finishing operation, so thermal damage can lead to very high costs. To avoid thermal damage, the resulting heat of the grinding process is dissipated with the help of cooling lubricant [1–5].

Commonly the cooling lubricant is supplied by external nozzles. These nozzles are operated with very high flow rates and pressures because of the difficult to reach contact zone between the grinding wheel and the workpiece. Moreover, the penetration of the surrounding air layer of the grinding wheel requires high flow rates and pressures. This air layer deflects the fluid jet away from the contact zone, especially at high wheel speeds. For this reason, these flow rates are higher than the ones actually needed to cool the contact zone, thus savings are possible [6–8]. This resulted in the development of alternative cooling lubricant supply methods like shoe nozzles, accounting for good cooling performance at low flow rates. Disadvantages of shoe nozzles are high efforts to adjust the nozzle and the limitation of possible depth of cut [2].

Another alternative is a cooling lubricant supply through channels inside the grinding wheel. This concept allows a direct coolant supply into the contact zone. The main advantage is a continuous supply of the entire contact zone with cooling lubricant even for high contact lengths. Moreover, the cooling lubricant supply is not influenced by the surrounding air layer. Due to the efficient coolant supply higher possible removal rates or a reduction of the flow rates can be assumed.

This paper presents the development and flow analyses of a grinding wheel with an internal cooling lubricant supply (GIC). In contrast to existing grinding wheels with internal cooling lubricant supply [2, 9–12], the channels in the grinding wheel are designed similar to those of a centrifugal pump impeller. For the designing process of the

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GIC, first the hydraulic design is analytically determined. Afterwards an analysis of the fluid flow is done by computational fluid dynamic (CFD) simulation. This approach facilitates constant fluid flow with reduced cavitation or stall even for high cutting speeds.

2 Previous work

In [13], an aluminum prototype wheel was introduced that provides channels inside the basic body to facilitate an internal cooling lubricant supply. The channel's hydraulic design is similar to those of a centrifugal pump impeller and was evaluated by computational fluid dynamic simulations. The coolant discharges through all channels at the entire circumference of the grinding wheel. In experimental investigations it was shown that the concept of a grinding wheel with an internal cooling lubricant supply at high wheel speeds (up to 50 m/s) is possible when considering fluid dynamics [13]. This aluminum prototype wheel was designed for flow analysis only and thus was not equipped with an abrasive layer. Based on these investigations using the aluminum prototype wheel, a GIC, that is equipped with an abrasive layer, was developed. The design process of the GIC and flow analyses are presented in this paper.

3 Design of the grinding wheel with an internal cooling lubricant supply (GIC)

3.1 Setup of the GIC

The GIC is an assembly of a base plate (Fig. 1 a) and an outer cover plate (Fig. 1 b) that are fixed together and are equipped with an abrasive layer (electroplated single layer cubic boron nitride). Once these two components are equipped with the abrasive layer, they cannot be disassembled without destroying the abrasive layer. These two components are mounted on the machine spindle and fixed with screws. The next component is an impeller that covers the mounting-screws and assures an efficient supply of the cooling lubricant to the channels (Fig. 1c). The final component is an inner cover plate that covers the impeller and is equipped with an inlet for the coolant supply (Fig. 1d). In the following chapter the design process of the GIC will be described.

3.2 Adaption of the geometry

The design of the developed GIC is based on the design of the aluminum prototype wheel presented in [13]. The aluminum prototype wheel, with a total width of 30 mm,

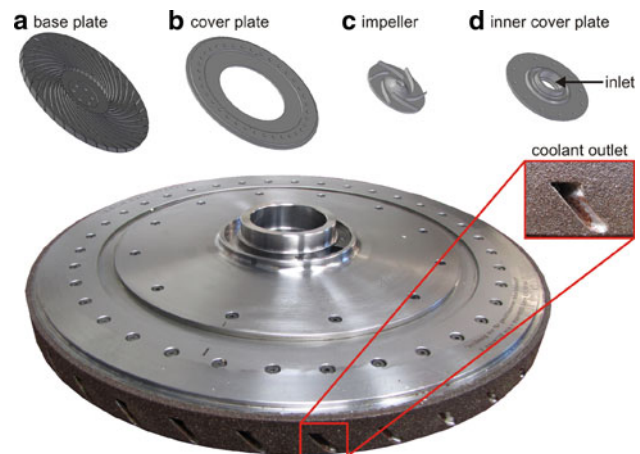


Fig. 1 Setup of the grinding wheel: parts and assembled

provides channels with a height of 10 mm. The outlets of the cooling channels cover 23% of the complete circumferential surface and 70% of the circumferential surface related to the height of the channels respectively (see Fig. 2). Therefore, only 30% of the circumferential surface can be used for the abrasive layer (related to the height of the channels).

In the area of the emersions, with only 30% of abrasive layer, high wheel wear can be expected, resulting in low tool life. To increase the tool life of the GIC, the area for the abrasive layer had to be enlarged. Therefore, the geometry of the outlets had to be adapted. It was necessary to create a transition from the channel geometry inside the grinding wheel (inner geometry) to the geometry at the circumferential of the grinding wheel (outer geometry) (see Fig. 3).

The coolant outlets are rotated by a setting angle relating to the centerline of the wheel. High setting angles result in a better supply of the contact zone: they increase the areas that are covered by outlets and thus by direct coolant supply (see Fig. 4). However, high setting angles are difficult to machine because of the undercuts in the transition area of the outer and inner channel geometry. As a compromise between good coolant supply and feasible undercuts, a setting angle of 45° degree has been specified. To

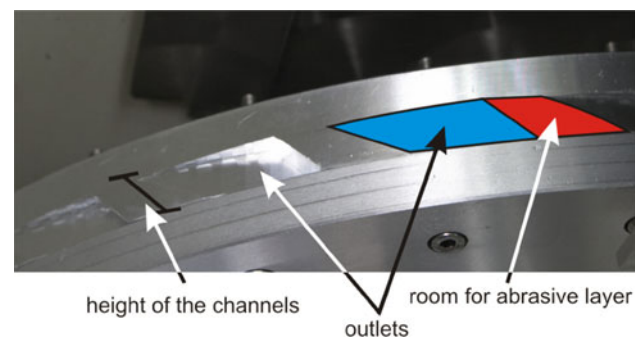


Fig. 2 Geometry of the aluminum prototype wheel

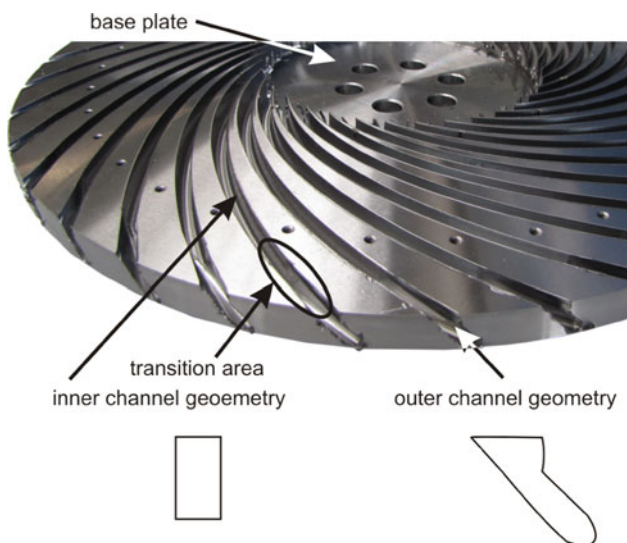


Fig. 3 Geometry of the channels

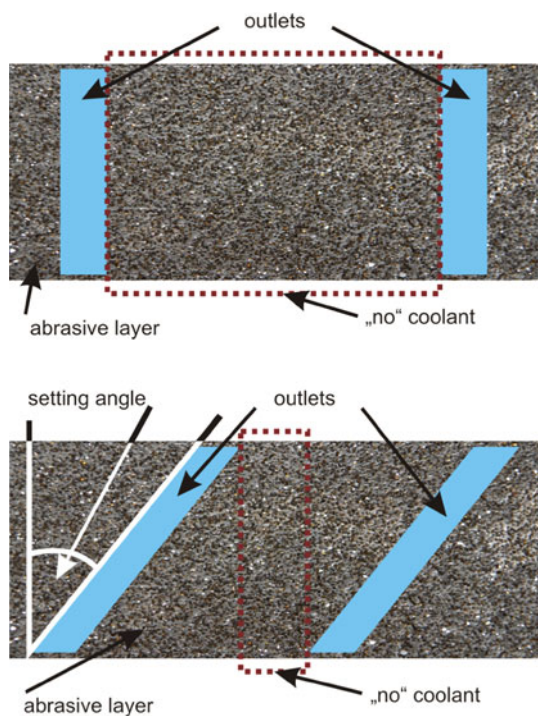


Fig. 4 Cooling supply with and without setting angle

guarantee the machinability of the undercuts, a minimal diameter of the milling tool of 3 mm is required. The resulting width of the coolant outlet is approximately 4 mm. Finally, the cross-sectional areas of the outlets and in the transition areas were held constant to assure constant flow velocities.

Grinding wheels with an internal cooling lubricant supply must consist of a base plate and a cover plate to cover the channels (see Fig. 5). In this case, the channels

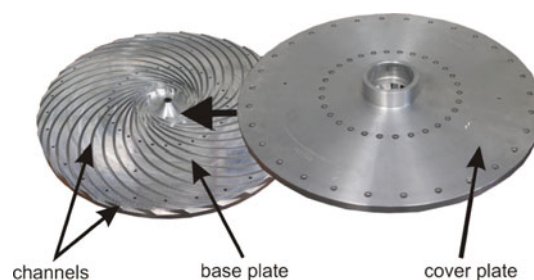


Fig. 5 Principle of the setup of a grinding wheel with an internal cooling lubricant supply

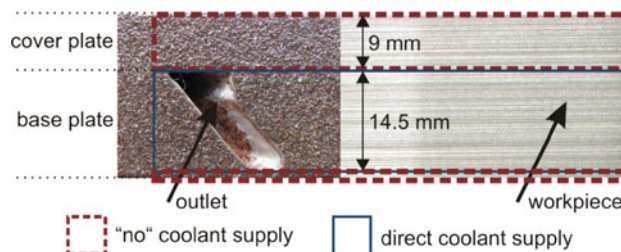


Fig. 6 Coolant supply regions

and the outlets of the channels are restricted to the base plate of the grinding wheel. This means there are no channels inside the cover plate.

The channels inside the base plate decrease the stability of the grinding wheel. For this reason, the width of the base plate is set to 15 mm, which assures sufficient stability and stiffness in high speed grinding (this was checked by stability calculations made by a grinding wheel manufacturer). The height of each coolant outlet is 14.5 mm, resulting in a cross-sectional area of about 70 mm². The remaining circumferential surface of the GIC (i.e. the areas that are not covered by the coolant outlets) is covered by the abrasive layer. For workpieces thicker than 14.5 mm, only 14.5 mm of the workpiece width will be directly supplied with coolant through the outlets (see Fig. 6).

The number of channels was set to 36 due to space limitations. Larger numbers of channels result in shorter channels and hence in a worse cooling lubricant supply. The cooling lubricant discharges at the entire circumference of the grinding wheel. For this reason, only 1/36 of the entire coolant is supplied through each channel. At a total flow rate of 300 L/min, about 8.3 L/min of cooling lubricant are supplied at each channel. The width of the cover plate is 9 mm, resulting in a total width of the GIC of 24 mm. Considering the setting angle of 45° and the width of the channels of about 4 mm, the coolant outlets take approximately 14% of the circumferential surface of the GIC, related to the height of the channels. Consequently, 86 % of the circumferential surface of the GIC consists of abrasive layer, related to the height of the channels.

Based on the specified channel geometry and the dimensions of the GIC, the hydraulic design is accomplished. The hydraulic design and its verification by using computational fluid dynamics simulations are presented in chapter 3.3.

3.3 Hydraulic design of the GIC

The design of the internal cooling channels of the grinding wheel is similar to the design of channels of a centrifugal pump impeller. Thus an enhanced hydraulic efficiency can be achieved. For the designing process of the GIC, first the hydraulic design is analytically determined. Afterwards an analysis of the fluid flow is done by CFD simulation.

3.3.1 Determination of the hydraulic design

For the determination of the hydraulic design, the fluid flow is regarded via integral average values, i.e. one-dimensional. Three-dimensional effects like secondary flow or non-uniform velocity profiles are neglected and will be examined in the CFD simulations.

Due to different requirements of a centrifugal pump impeller compared to the GIC, different input parameters are given for the designing process. As an initial step of the designing process of a centrifugal pump impeller, the delivery head and the flow rate are defined. In a next step the impeller type (e.g. radial or axial flow pump) has to be chosen and the impeller's dimensions, as well as the rotational speed has to be determined [14, 15].

In contrast to that, the input parameters for the hydraulic design of the GIC (i.e. the curvature of the channels) are the wheel diameter, the cross-sectional area of the coolant outlets (as presented in chapter 3.2), and the desired grinding wheel speed (here: up to 100 m/s).

The criterion used for the curvature of the channels is the so called “shock-free entry” [14]. This means that the angle of the flow velocity at the beginning of the channels (β_1) has to be the same as the angle of the turbine blade at the beginning of the channels (see Fig. 7). In this case, the turbine blades are the walls of the channels. The angle of the flow velocity depends on the flow rate, the cross-sectional area of the channels and the rotational speed (in this case: the cutting speed). The angle of the turbine blade at the end of the channels (i.e. at the circumference of the wheel) determines the possible delivery head. While common centrifugal pump impellers provide angles of $\beta_2 = 20\text{--}25^\circ$, for the GIC an angle of $\beta_2 = 45^\circ$ was set. First of all, this results in easier to manufacture undercuts in the transition area of the channels. Furthermore it results in less fluid friction inside the channels and higher fluid pressures at the outlets.

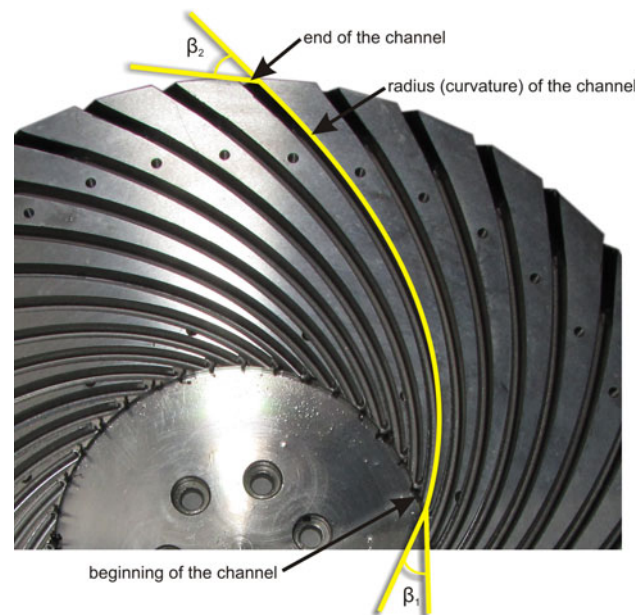


Fig. 7 Hydraulic design of the GIC

With the angles β_1 and β_2 , the curvature of the channels can be determined. This can be done analytically or with the use of computer-aided design (CAD). For the determination via CAD, the position of the beginning and the end of a channel within the GIC is set. Then the angle β_1 at the beginning of the channel and the angle β_2 at the end of the channel are set. With these geometrical conditions, only one possible radius for the curvature of the channel exists (see Fig. 7) [16]. The same procedure was done for the impeller that covers the mounting-screws to assure a shock-free entry of the cooling lubricant to the channels (see chapter 3.2).

3.3.2 Computational fluid dynamics (CFD) simulations

After the analytical specification of the hydraulic design is determined, computational fluid dynamics (CFD) simulations were used to gain detailed information about the fluid flow through the cooling channels and hence the hydraulic efficiency.

Computational fluid dynamics, which is based on Navier–Stokes-equations, simulates pressure and flow velocities for discrete points. The Navier–Stokes-equations represent a coupled system of non-linear differential equations that describe the fluid flow of Newtonian fluids. Due to the coupling and non-linearity of the system of equations, simplifications are needed to enable simulation results in a reasonable time. In this case, the Reynolds Averaged Navier–Stokes (RANS) approach is used, which represents an approximate averaged solution of the Navier–Stokes equations. Furthermore, the SST- $k\omega$ turbulence

model is utilized, which offers good approximations for both areas inside the flow and boundary areas (e.g. pipe wall).

The pump characteristic curve of the GIC (i.e. the total head versus the flow rate) is simulated by CFD. To evaluate the pumping action of the GIC, it is simulated with the same boundary conditions as common centrifugal pump impellers. For this reason, a volute casing is employed to deliver the fluid away from the coolant outlets. This volute casing delivers the fluid to an outlet branch (Fig. 8) and avoids boundary effects on the fluid flow inside the grinding wheel.

The simulation results are the delivery head and the flow rate. In addition to the delivery head, also the torque can be simulated and hence the driving power of the grinding wheel can be calculated. The driving power corresponds to the spindle power and can be compared to experimental results. In addition to integral values, like delivery head and power, local values (pressure and fluid velocity at discrete points) can be simulated. Local values help to localize regions of low pressure where gas evolution or even cavitation can occur.

The pump characteristic curves can be converted for different grinding wheel speeds by the affinity laws (Table 1). The degree of efficiency remains approximately constant, so the values only depend on the grinding wheel speed (which is proportional to the rotational speed n).

In Fig. 9 the pump characteristic curves of the grinding wheel at grinding wheel speeds of 40, 60 and 100 m/s are depicted. The pump characteristic curve for 100 m/s is simulated. The pump characteristic curves for 60 and 40 m/s are derived from the simulated curve for 100 m/s by the affinity laws. For verification of the extrapolated curves, two values for 60 and 40 m/s were simulated.

The shape of the pump characteristic curves is similar to those of common centrifugal pump impellers. The plots show that the total head rises rapidly with increasing speed. The non-linear shape of the curves result from friction effects, turbulences and impact losses.

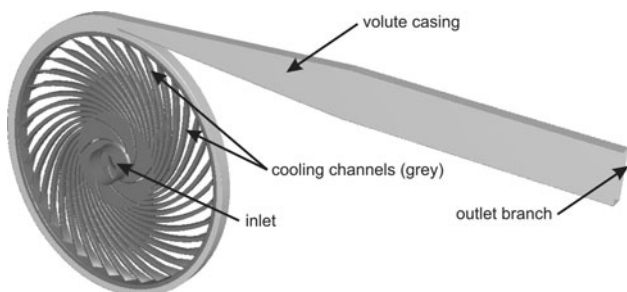


Fig. 8 Negative volume of the grinding wheel and the volute casing (outer surfaces of the flow area)

Table 1 Affinity laws (simplified)

Flow-rate Q	$(Q_1/Q_2) = (n_1/n_2)$
Total head H	$(H_1/H_2) = (n_1^2/n_2^2)$
Drive power P	$(P_1/P_2) = (n_1^3/n_2^3)$

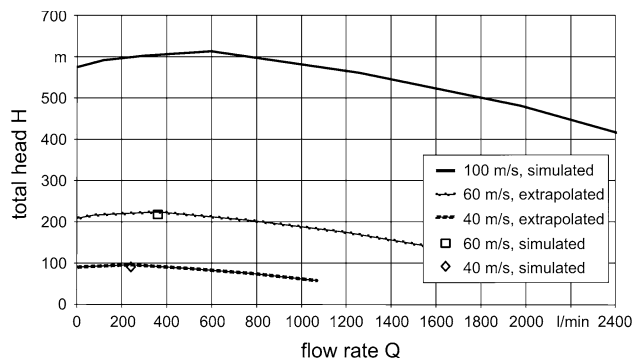


Fig. 9 Pump characteristic curve at 100, 60 and 40 m/s for the GIC

The designing process of the GIC was similar to that of a centrifugal pump impeller. Despite differing input parameters for the GIC compared to common centrifugal pump impellers, a sufficient hydraulic efficiency can be achieved. However, the experimental boundary conditions of the GIC differ to centrifugal pump impellers. Whereas centrifugal pump impellers are operated within a volute casing (similar to the simulations), the GIC pumps the fluid free into the atmosphere. For this reason experimental investigations have to proof if the GIC is able to deliver fluid.

4 Experimental investigations of the fluid flow of the GIC

4.1 Experimental setup

The performance tests are conducted on a high efficiency surface grinding machine with high stiffness. The flow rate of the cooling lubricant can be adjusted infinitely variable. For the experiments with the GIC the maximum flow rate was 390 L/min. The cooling lubricant used is mineral oil (Petrofer Isocut R 10-H). The experimentally measured spindle power (measured by a non-contact hall effect probe) is compared to the calculated spindle power (based on the CFD simulation results).

4.1.1 Test conditions

For investigations of the pump capacity of the GIC, different grinding wheel speeds were examined using different flow rates. In the first series of tests a maximum grinding wheel speed of 60 m/s was used. The parameter

Table 2 Test conditions

	Parameters
Grinding wheel speeds in m/s	30–40–60
Flow rates in L/min	100–200–300–390

combinations of grinding wheel speed and flow rate are shown in Table 2. Additionally very low grinding wheel speeds with a flow rate of 200 L/min have been conducted to provide an optical impression of the fluid flow of the GIC.

5 Results

As mentioned above, low circumferential speeds enable an optical impression of the fluid flow of the GIC (Fig. 10). When using higher grinding wheel speeds, the atomization of the fluid jet increases and the observation of the coolant flow is no longer possible. This can be seen when comparing circumferential speeds of 1 and 5 m/s (Fig. 10).

Similar to previously presented results [13], only marginal amounts of cooling lubricant discharge at the inlet. The low leakage confirms the good hydraulic design with low flow resistance resulting in a high pump capacity of the GIC.

For examination of the cooling lubricant supply at high grinding wheel speeds and flow rates, the effective spindle power was measured and compared to the simulation results. Concerning experimental investigations, the spindle power caused by acceleration of the grinding wheel itself has been removed from the measurements. Therefore, the spindle power at zero flow rate, i.e. the no-load power, was measured and used as an offset. Based on the CFD simulations, only the required power to accelerate the fluid, not the grinding wheel, is calculated.

In Fig. 11 the simulated and the experimentally measured spindle power for a grinding wheel speed of 30 m/s is plotted. The results for grinding wheel speeds of 40 and 60 m/s are displayed in Fig. 12.

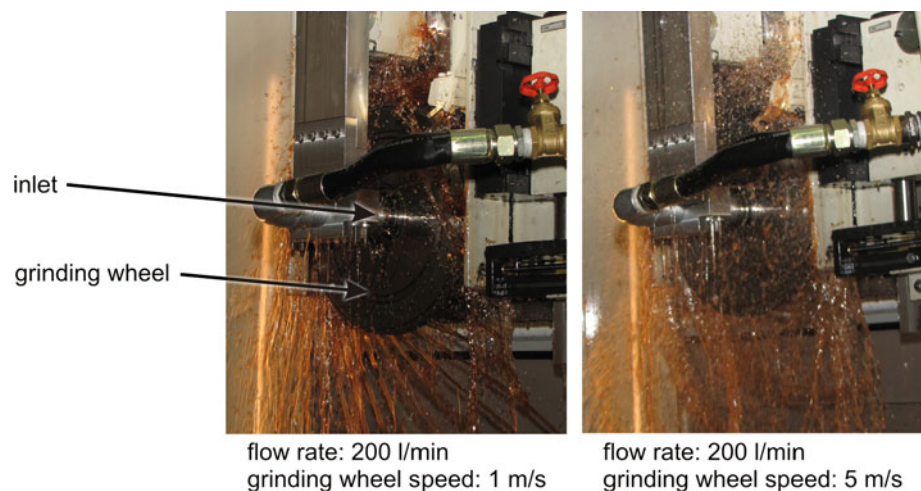
The qualitative shape of the experimentally measured and simulated curves is comparable. Both curves show a linear behavior over the flow rate and decline at about 300–400 L/min. Independently of the observed flow rate the simulated spindle power is about two times greater than the experimentally measured.

The differences result from different boundary conditions of simulation and experiment. In the CFD simulations, boundary conditions used were comparable to simulations of common centrifugal pump impellers i.e. a volute casing was modeled. In the experimental investigations the GIC is not covered by a volute casing and the fluid discharges free into the atmosphere. This also results in less friction at the coolant outlets and thus in smaller spindle powers.

The main difference of the measured and simulated power can be attributed to ambient air sucked in at the coolant inlet of the GIC. The density of the ambient air is nearly 700 times smaller than that of the cooling lubricant. For this reason already few amounts result in smaller spindle power to transport the medium. For the CFD simulation a single-phase flow (only cooling lubricant) is assumed. Therefore the calculated spindle power based on CFD simulations is higher compared to the experiment. Modeling and simulation of gases in fluid flows are difficult and additionally depend on the centrifugal pump impeller type [17, 18]. In summary, the pump characteristic curves show good correlation despite different boundary conditions for experiment and simulation.

For higher grinding wheel speeds the fluid friction inside the channels of the GIC and its pump capacity increases (Fig. 12). Despite higher pump capacities a constant amount of cooling lubricant is supplied. For this reason a higher amount of ambient air in the fluid flow can be

Fig. 10 Optical impression of the fluid flow



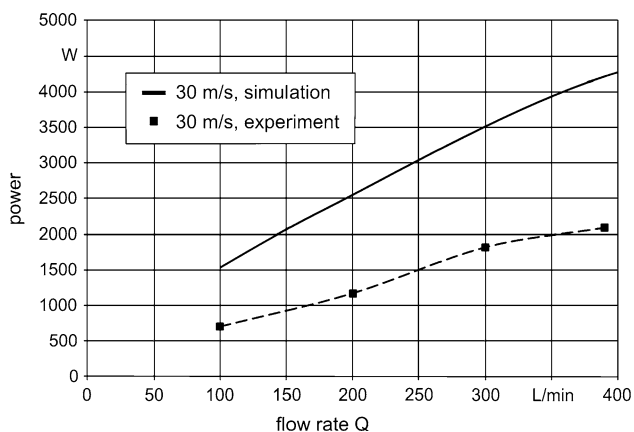


Fig. 11 Simulated and experimentally measured spindle power at a grinding wheel speed of 30 m/s

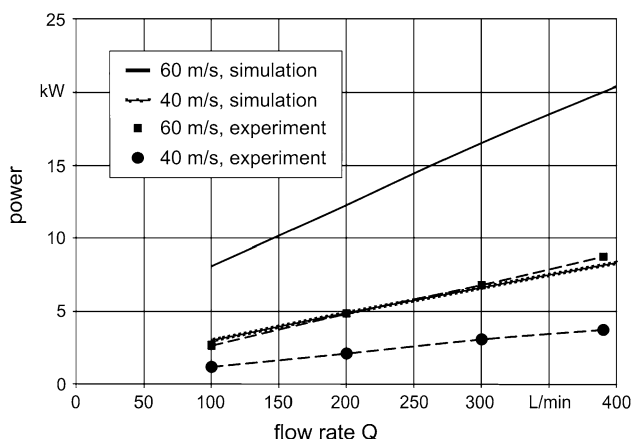


Fig. 12 Simulated and experimentally measured spindle power at a grinding wheel speed of 40 and 60 m/s

assumed. Therefore, the experimentally measured spindle power is not increasing as strong as the calculated spindle power based on the simulation results, where no ambient air was assumed in the fluid flow. In further studies the simulation’s boundary conditions have to be further adapted to the experimental setup.

6 Conclusions

A grinding wheel with an internal cooling lubricant supply (GIC) that allows a direct coolant supply into the contact zone was presented in this paper. When using the GIC the entire contact zone is constantly supplied with cooling lubricant even for high contact lengths. Moreover, the cooling lubricant supply is not influenced by the surrounding air layer.

In contrast to existing grinding wheels with an internal cooling lubricant supply, the channels in the grinding

wheel were designed similar to those of a centrifugal pump impeller. The design of the channels is supported by computational fluid dynamic (CFD) simulations.

After the hydraulic design was analytically determined and confirmed by CFD simulations, the GIC was manufactured. For examination of the cooling lubricant supply grinding wheel speeds of 30, 40 and 60 m/s and flow rates of up to 390 L/min were tested. The effective spindle power was measured for the GIC and compared to the simulation results. The simulated shape of the pump characteristic curves of the GIC were similar to those of common centrifugal pump impellers.

However, the simulated spindle powers are higher than the experimentally measured spindle powers. The differences result from different boundary conditions between simulation and experiment. In the CFD simulations, boundary conditions used were comparable to simulations of common centrifugal pump impellers i.e. a volute casing is applied. In the experimental investigations the GIC is not covered by a volute casing and the fluid discharges free into the atmosphere. The main differences between simulation and experiment can be attributed to the unknown amount of ambient air sucked in by the GIC during the experiments. In future studies the simulation’s boundary conditions have to be further adapted to the experimental setup.

In further experimental works the performance of the grinding wheel when grinding hardened steel will be investigated. The abrasive layer used for the GIC is an electroplated cubic boron nitride layer which enables high removal rates. The performance of the GIC will be compared to a reference process with a conventional electroplated CBN grinding wheel with coolant supply by an external nozzle.

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References

1. Grof E (1977) Beitrag zur Klärung des Trennvorganges beim Schleifen von Metallen. Dissertation. TU München 1977
2. Brinksmeier E, Heinzl C, Wittmann M (1999) Friction, cooling and lubrication in grinding. *Annal CIRP* 48(2):581–598
3. Brinksmeier E, Wittmann M, Klocke F, Beck T (2000) Strategien zur Kühlschmierstoff-Zuführung beim Schleifen. *Schleiftechnisches Kolloquium, Universität Bremen Paper Nr. 10*
4. Brinksmeier E, Wilke T (2004) Erfassung und Vermeidung thermischer Randzonenbeeinflussung beim Schleifen. Stuttgart, Konferenzbericht Moderne Schleiftechnologie und Feinstbearbeitung
5. Meyer L, Heinzl C, Brinksmeier E (2005) Analysis and optimization of coolant supply conditions in grinding. *Product Eng XII(2):27–30*

6. Treffert C (1995) Hochgeschwindigkeitsschleifen mit galvanisch gebundenen CBN Schleifscheiben. Dissertation. Fakultät für Maschinenwesen RWTH Aachen
7. Klocke F, Beck T (1997) Zufuhrsysteme für den anforderungsgerechten Kühlschmierstoffeinsatz. Schleiftechnisches Kolloquium, VDI-Verlag
8. Wittmann M (2007) Bedarfsgerechte Kühlschmierung beim Schleifen. Dissertation. Universität Bremen
9. Saljé E, Riefenstahl J (1982) Kühlmittelzufuhr durch die schleifscheibe beim innenrundsleifen. *Industrie-Anzeiger* 43(104):39–40
10. Klocke F (1998) Gleiche Oberflächenqualität mit Innenkühlung. *Industrieanzeiger* 37/98:68
11. Martens R, Maier R, Bettingen A (1998) Innenkühlung beim CBN-Schleifen. *VDI-Z* 140 Nr. 1/2 1998:42-44
12. Nguyen T, Zhang L (2008) Performance of a new segmented grinding wheel system. *Int J Mach Tool Manuf* 49(1–3):291–296
13. Aurich JC, Sudermann H, Hellmann D-H, Roclawski H (2006) Interne Schleifölfzufuhr beim Hochleistungsschleifen. *wt Werkstatttechnik online* 96-11/12:810–813
14. Gülich JF (2004) *Kreiselpumpen*. Springer
15. Sprengler H (1976) *Technisches Handbuch Pumpen*. VEB
16. Pfeleiderer C (1961) *Die Kreiselpumpen für Flüssigkeiten und Gase*. Springer
17. Tillak P (1998) *Förderverhalten von Kreiselpumpen bei viskosen, gasbeladenen Flüssigkeiten*. Dissertation. TU Kaiserslautern
18. Sauer M (2002) *Einfluß der Zuströmung auf das Förderverhalten von Kreiselpumpen radialer Bauart bei Flüssigkeits-/Gasförderung*. Dissertation. TU Kaiserslautern