Numerical Model to Predict Kickback for Angle Grinders

Gérard Fleury

INRS (Institut National de Recherche et de Sécurité), Nancy, France

Abstract

Angle grinders are a type of powertool used widely by companies and craftsmen for grinding and cutting applications. The choice of an appropriate rotating tool is essential to ensure the safety of the operator. Nevertheless, some working conditions can cause the rotating disc to be either pinched or snagged. As the rotational kinetic energy of the disc decreases, a fraction of the energy is lost due to frictional warming of the disc and workpiece, but the remainder of the variation in energy may be transmitted as kinetic energy to the grinder, leading to it being ejected. This sudden reaction of the powertool, known as “kickback”, can, in rare cases, cause serious or even fatal injuries to the worker. INRS, the French Research and Safety Institute for the Prevention of Occupational Accidents and Diseases, is currently conducting a study to investigate kickback for angle grinders and the resulting increase in the risk of injury.

Preliminary research has led to the development of a prototype test bench dedicated to reproducing kickback events in the laboratory. The test consists of pinching the rotating wheel of a grinder and observing the resulting ejection of the powertool. A modelling approach was initiated to support the further development of the test bench and to increase our understanding of the mechanical phenomena leading to the ejection of the grinder. The input parameters are the properties of the rotating wheel (mass, diameter, moment of inertia and initial angular velocity) and the features that characterize the angle grinder (mass, moment of inertia and position of the centre of gravity). The magnitude of the pinching force applied to the rotating wheel, the initial position of its application point and the coefficient of friction between the wheel and the brake device are also taken into account in the numerical simulation. The kinetic energy of the grinder at the time of ejection and the direction of ejection are calculated in order to assess the kickback event.

The model is implemented using LS-DYNA and the Python programming language to solve the differential equations that govern the phenomena leading to the ejection. In this paper, we present the model, its implementation and the results. It illustrates how the strengths of both LS-DYNA and Python are exploited and combined to achieve our goal. In particular, the effect of the position of the initial contact point on the magnitude of the kickback and its direction is investigated and maps are presented and discussed.

1 Introduction

Angle grinders are a type of powertool used widely by companies and craftsmen for grinding and cutting applications. The operator must comply with the correct operating procedures and conditions, including the choice of an appropriate tool (cutting disc, grinding wheel, backing pad or brush), to ensure his safety. Even so, some working conditions can cause the rotating tool to be either pinched or snagged. As the rotational kinetic energy of the wheel decreases, a fraction of the energy is lost due to friction between the wheel and the workpiece. However, if sliding occurs, kinetic energy may be transmitted to the grinder leading to it being ejected from the contact area between the disc and the workpiece. This sudden reaction of the powertool, known as “kickback” and defined in detail in IEC/EN standard 60745 [1], can, in rare cases, cause serious or even fatal injuries to the worker. Kickback cannot be avoided but some manufacturers have designed their own technical solutions to reduce the risk of kickbacks and have commercialized angle grinders that are equipped with kickback reducing systems [2, 3].

A few studies dealing with kickback related to hand-held powertools have been published, but most of these focused on chain saws [4, 5], whose performance in terms of reducing the risk of kickback is assessed by tests described in the ISO-9518 standard [6]. Bolay [7] investigated kickback and developed a test bench allowing the rotating disc of a grinder to be pinched. The appropriate sensors were used to measure the physical quantities that should be considered in the development of kickback-reducing equipment. However, no method was proposed for checking and quantifying the effectiveness of the newly developed safety features.

The aim of the on-going study conducted by INRS is to develop a comprehensive physical understanding of kickback that is applicable to all electric angle grinders and to eventually propose a new method to assess the performance of kickback-reducing features embedded in electric angle grinders.

Preliminary work led to the development of a prototype test bench (see Fig. 1) dedicated to reproducing kickback events in the laboratory. The procedure selected consists of pinching the rotating wheel of a grinder and observing the resulting ejection.

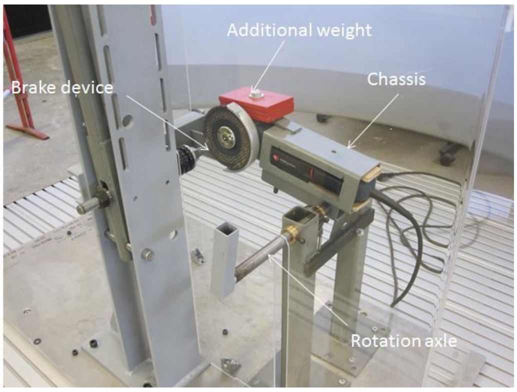

Fig.1: Prototype test bench for generating kickback

Kickbacks were observed using the prototype test bench, but further developments are still required. For the current test procedure, the grinder is clamped to a steel tube which can only rotate around a fixed axle. However, assuming that the ejection lasts less than the human response time (i.e. that the operator cannot interact with the grinder as kickback occurs), less restriction to chassis movement is necessary. In addition, the pinching conditions selected should aim to transfer the highest energy to the grinder in order to achieve the most powerful ejection. If the pinching force is too low, it is assumed that the grinder will be ejected with low energy; however if the pinching force is too high, the disc will remain snagged in the brake device. Therefore, there is an optimum level of force at which the ejection is most powerful. Finally, relevant physical quantities and objective indicators must be identified in order to equip the test bench with appropriate sensors.

Because of the technical issues described above, as well as the importance of maintaining test safety during the development of this procedure, a modelling approach was developed in parallel. In this paper, we present the modelling approach, the models developed and the first results obtained in a parametric study.

2 The modelling approach

The aim of the model was to predict the kickback of an angle grinder as its rotating wheel is suddenly pinched. The model is based on fundamental assumptions that define which phenomena need to be taken into account. This allowed identification of the set of the input parameters for the model, as described in Section 2.1. In a second step, the model was implemented using the capabilities of LS-DYNA. Simulations were conducted by varying the values of certain input parameters and results were analysed using LS-PrePost. This work provided valuable information that allowed us to improve our understanding of the mechanical phenomena involved. This in turn helped us to express the differential equations that govern the motion of the grinder and to define the indicators required to quantify kickback. These differential equations were finally solved by means of Python code, allowing our numerical process to be optimized for simulating the response of a grinder to a large number of initial conditions.

2.1 The model

The numerical simulation of a kickback event requires the angle grinder and the brake system to be modelled in order to reproduce the interaction of the workpiece with the rotating disc. As depicted in Fig. 2, the electric angle grinder is composed of four main components: the body, including the electrical motor and the gear box, an accessory (a disk in our case), a side handle and a protective guard. The side handle and the protective guard are movable, depending on the operating conditions. The movements of the machine are described with respect to a fixed coordinate system (O,x,y,z) where the origin corresponds to the initial position of the centre of gravity of the disc. The x-axis is parallel to the longitudinal axis of the body and oriented towards its centre of gravity G_g. In the orthogonal direction, the y-axis is directed from the centre of gravity of the disc G_d towards the centre of gravity of the body G_g. Consequently, the z-axis is oriented to satisfy (x,y,z) is orthonormal direct. With respect to this coordinate system, angle grinders are designed to drive their disc with a negative angular velocity.

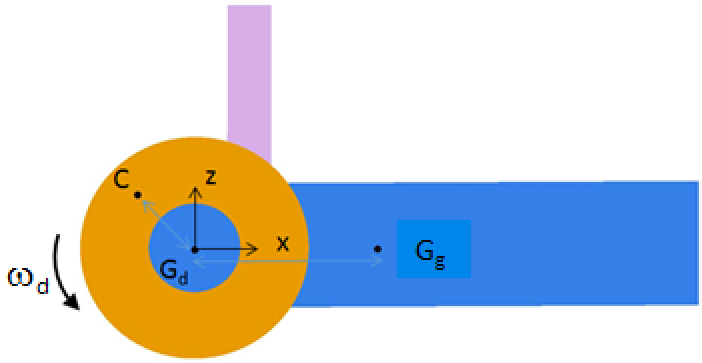

Fig.2: Model of an angle grinder

The brake system consists of two separates brake pads mounted in a brake shoe. Brake pads are small deformable cylinders oriented in the y-direction and positioned face to face on each side of the disc. The brake shoe is an assembly of rigid bodies that guide the pads towards each other and thus exert two opposite constant forces, applied simultaneously and perpendicularly on both sides of the disc. The cross section of each cylinder, defined by its radius R_{mor}, is small enough to assume a punctual interaction with the disc. The constant normal force, F_N, is therefore applied at the contact point C, defined as the centre of the brake pad and having a fixed position (x_c, z_c). Due to gliding of the disc relative to the brake pads, the physical contact point on the disc changes continuously and the set of contact points describes a curve on the disc called the ejection path. The interaction between the disc faces and the pads are taken into account by means of a constant friction coefficient µ, defined as the normal force F_N divided by the tangential force F_T. Thus, the magnitude of the tangential force is known, but its direction in the (x,z) plane varies during the phase of pinching. The wheel is assumed to be a rotating rigid disc defined by its radius R_d, its mass m_d, and its initial angular velocity ω_d. The body of the grinder is also assumed to be a rigid body modelled by the initial position of its centre of gravity d_g, its mass m_g and its moment of inertia J_g about a y-axis through its centre of mass. The effect of the power supplied by the electrical motor and transmitted to the disc is also integrated in the model. The motor is assumed to exert a continuous torque to the disc with a linear behaviour, i.e. the transmitted torque decreases linearly with the angular velocity. Two additional parameters, T_0 and T_s, are therefore required to model this effect. This reference model is therefore a 2-rigid-body model requiring the input of 13 parameters (R_{mors}, x_c, z_c, F_N, µ, d_g, m_g, J_g, m_d, R_d, ω_d, T_0, T_s).
Calculations are performed in order to simulate the ejection path up to the point where the contact between the disc and the friction pads vanishes. During this phase, no other external loads are applied to the grinder. This is realistic if the gravity can be neglected regarding the accelerations resulting from the pinching force and if the human operator has no time to react, i.e. the duration of the phase of gliding between the workpiece and the disc is shorter than the human response time (about 0.2 s).
To perform a simulation, a preliminary check is carried out to exclude any points within the area of the disc that is overlapped by the body of the machine, where pinching is practically impossible. For the initial conditions leading to the ejection of the grinder, its kinetic energy (calculated as the sum of the

translational and the rotational energies) and the direction of the ejection are selected for quantifying kickbacks.

2.2 Implementation in LS-DYNA

Geometries and meshes were built with GMSH, a three-dimensional finite element mesh generator distributed under the terms of the GNU General Public Licence [8]. Keyword files are generated by using internally developed Python scripts [9]. The grinder was modelled as a *PART_INERTIA, which is defined by its mass (TM), its centre of gravity (NODEID) and its moment of inertia IYY. The disc is meshed with thick shells (*SECTION_TSHELL) and is defined as a rigid body by filling out a *RIGID_MATERIAL card. The density (RHO) is adjusted to obtain the mass of the disc. The *PART of the disc is linked to the *PART of the grinder by means of a revolute joint in the y-axis (*CONSTRAINT_JOINT_REVOLUTE) and an initial angular velocity around the y-axis is applied to the disc by a *INITIAL_VELOCITY_GENERATION card. Both grinder and disc have constrained displacement in the y-direction as well as constrained rotations in the x and z-directions (CMO 126).

The brake shoe consists of two independent small and light cylinders meshed with solid elements and defined as rigid bodies. Each body is separately connected to the ground part by means of a cylindrical joint whose axis corresponds to the longitudinal axis of the cylinders oriented in the y-direction. Thus, each cylinder of the brake shoe can rotate and translate along its longitudinal axis. Each part of the brake shoe is connected to a brake pad by merging nodes of a common face. Pads are modelled as small light deformable cylinders with elastic behaviour and meshed with solid elements. A contact is defined between each pad and each face of the disc by defining appropriate segment sets referenced in a *CONTACT_AUTOMATIC_SURFACE_TO_SURFACE card, where the parameters FS and FD are assumed to be equal to the friction coefficient µ. Two forces with opposite signs are defined by means of a LCID curve and are applied to the rigid bodies of the brake shoe via a *LOAD_RIGID_BODY card.
Gravity is applied to the whole model by adding a *LOAD_BODY_Z card. The resistance torque exerted by the electrical motor on the rotating disc is applied by means of a *DEFINE_CURVE_FUNCTION card that defines the continuously applied torque as a function of the angular velocity of the rotating disc.

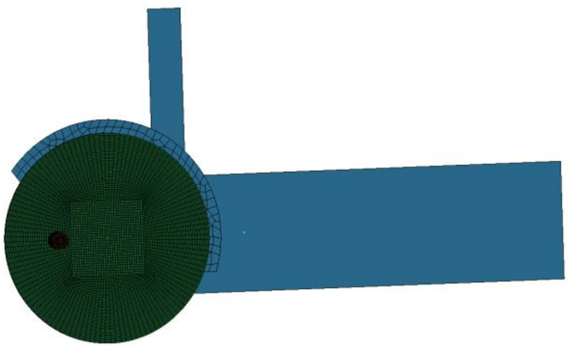

Fig.3: The LS-DYNA model

Explicit analyses with controlled constant timestep option (10^{-5} s) were launched using the smp version R810 of LS-DYNA. Trajectories of all bodies were visualized and relevant time histories were analysed using the LS-PrePost graphical interface.

2.3 Differential equations and solving with the Python code

The kinetic energy transferred to the grinder and the ejection direction were calculated by moving the application point of the pinching force on the disc, and the results were then reported on energy and angle maps. These maps are generated by automated numerical simulations and analysis of results. To achieve this, we first applied the fundamental principles of mechanics to formulate the mechanical problem to be solved by means of a set of differential equations. All quantities are defined in the coordinate system (0,x,y,z) described in the previous chapter. Assuming that all motions belong to the x,z plane, the mechanical problem can be simplified as a 2D problem involving two rigid bodies connected to each other by a revolute joint. Six unknowns, $a_{xg}, a_{zg}, \ddot{\theta}_g, a_{xd}, a_{zd}, \ddot{\theta}_d$, which describe the movements of both bodies, must be calculated by means of six independent equations required to solve the problem. Newton's second law is applied to the whole system including the grinder and the disc, leading to two relationships:

$$-2\overrightarrow{F_T} = m_g \overrightarrow{a_g} + m_d \overrightarrow{a_d} \tag{1}$$

The application of the principle of conservation of the angular momentum to the grinder gives:

$$J_g \ddot{\theta}_g \vec{y} = m_g \left(\overrightarrow{G_g G_d} \, \Lambda \, \overrightarrow{\mathrm{a_g}} \right) \tag{2}$$

Assuming that the torque exerted by the electrical motor can be neglected, the variation of the angular velocity results from the action of the friction force applied at the point C. Thus, the application of the principle of conservation of the angular momentum to the disc leads to:

$$J_d \ddot{\theta}_d \vec{y} = \overrightarrow{G_d C} \, \Lambda \, \overrightarrow{\mathrm{F_T}} \tag{3}$$

This equation requires calculation of the orientation of the friction force vector and calculation of the position of the instantaneous rotation centre C_r, defined as the point with an instantaneous velocity equal to zero. This condition enables the coordinate of C_r to be expressed as a function of position, translational and the angular velocity of the disc. Orthogonality between the tangential friction force and the vector defined by the contact point, C, and the instantaneous rotation centre, $C_{r,}$ is assumed and leads to following additional equations:

$$\overrightarrow{F_T} \cdot \overrightarrow{C_r C} = 0 \quad \text{with} \quad \overrightarrow{V_0(C_r)} = \vec{0} \tag{4}$$

Two additional relationships that link the unknowns are written by applying kinematic constraints from the hinge that connects the disc to the grinder.

Finally, the six differential equations to be solved are:

$$a_{xg} = A_{11}\,\mathrm{B}_1 + A_{12} B_2 \tag{5}$$

$$a_{zg} = A_{21}\,\mathrm{B}_1 + A_{22} B_2 \tag{6}$$

$$\ddot{\theta}_g = \frac{m_g}{J_g}\left[a_{xg}\left(z_d - z_g\right) - a_{zg}\left(x_d - x_g\right)\right] \tag{7}$$

$$\ddot{\theta}_d = \frac{-2\mu F}{J_d \,\mathrm{var}} \left\{ \dot{\theta}_d \left[\left(x_d - x_c\right)^2 + \left(z_d - z_c\right)^2 \right] - V_{xd}\left(z_d - z_c\right) + V_{zd}\left(x_d - x_c\right) \right\} \tag{8}$$

$$a_{xd} = a_{xg} + \ddot{\theta}_g (z_d - z_g) - \dot{\theta}_g^{\,2} (x_d - x_g) \tag{9}$$

$$a_{zd} = a_{zg} - \ddot{\theta}_g (x_d - x_g) - \dot{\theta}_g^{\,2} (z_d - z_g) \tag{10}$$

with:

$$\mathrm{var} = \sqrt{\left(V_{xd} - \dot{\theta}_d (z_d - z_c)\right)^2 + \left(V_{zd} + \dot{\theta}_d (x_d - x_c)\right)^2} \qquad Const = \frac{1}{(m_g + m_d)^2 + \frac{(m_g + m_d) m_g\, m_d\, d_g^{\,2}}{J_g}}$$

$$A = \frac{1}{Const} \begin{bmatrix} m_g + m_d + \frac{m_d\, m_g}{J_g}(x_d - x_g)^2 & \frac{m_d\, m_g}{J_g}(x_d - x_g)(z_d - z_g) \\ \frac{m_d\, m_g}{J_g}(x_d - x_g)(z_d - z_g) & m_g + m_d + \frac{m_d\, m_g}{J_g}(z_d - z_g)^2 \end{bmatrix}$$

$$B = \begin{bmatrix} \frac{2\mu F}{\mathrm{var}}\left(-V_{xd} + \dot{\theta}_d (z_d - z_c)\right) + m_d\, \dot{\theta}_g^{\,2} (x_d - x_g) \\ \frac{2\mu F}{\mathrm{var}}\left(-V_{zd} - \dot{\theta}_d (x_d - x_c)\right) + m_d\, \dot{\theta}_g^{\,2} (z_d - z_g) \end{bmatrix}$$

These six differential equations can be easily integrated by implementing a Euler method and by assuming initial conditions for x_d, z_d, x_c, z_c, x_g, z_g, $\dot{\theta}_g$, $\dot{\theta}_d$.

3 Results

Multiple simulations were first carried out with LS-DYNA and the Python code to check that both implementations produced exactly the same results. In this section, we present the results of a case with the following input parameters: m_d=170 g, R_d=62.5 mm, ω_d=12000 rpm, m_g=2.2 kg, d_g=85 mm, J_m=0.013 kg m^2, F_N=1000 N, µ=0.3., and the variables x_c and z_c. The values describing the grinder and the disc are based on a typical grinder powered by a 1500 W electrical motor equipped with a 125 mm diameter disc rotating without load. The drive torque of the electrical motor is not considered in this case, and a 1000 N pinching force combined with a friction coefficient of 0.3, leading to a tangential force of 300 N, is a realistic set of values. The position of the brake pads x_c and z_c will be specified for each result shown.

3.1 Results of LS-DYNA simulations

For a simulation leading to the trajectory of the grinder depicted in Fig. 4, the brake pads are positioned at a distance of 0.7 x R_d from the disc centre and at an angle of 130° relative to the x-direction. Both the initial position (transparent bodies) and the position of the grinder when losing contact with the brake pads are shown in the figure. The black point corresponds to the fixed position of the brake pads and the red line illustrates the path of the contact point when the observer is attached to the grinder. The separation between the disc and the brake shoe occurs after 19 milliseconds. Analysis of the transfer of the energy shows that only a small proportion (13.3 J) of the initial energy (262 J), i.e. the rotational kinetic energy of the disc, is transmitted to the grinder, the difference being dissipated in heat due to the friction. The dissipated heat is identical to the mechanical work of the tangential friction force F_T, which can be calculated by multiplying F_T with the length of the trace of the contact point on the disc. This line can be visualized by using LSPre-Post to display the trace of the centre of the brake pad in a frame attached to the rotating disc (Follow Part). The display of this trace shows that the disc rotates 1.5 rounds prior to the ejection. Thus, the dissipated heat is approximately 1.5 x 0.7 Rd x 2 µ F_N = 247 J, which is in agreement with the loss of energy predicted by LS-DYNA.

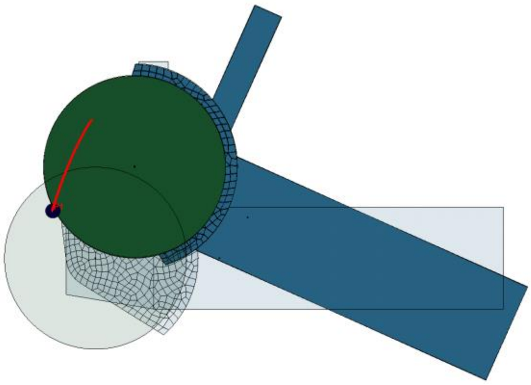

Fig.4: Trajectory followed by the grinder during its ejection

Figure 5 shows the second part of the trajectory, while the grinder is moving in the air. A yellow table and the outline of a human body have been added to create a passive environment, which is useful as a frame reference to facilitate understanding and analysis of the movements of the grinder. The position of the grinder at ejection (transparent bodies) and its final position at about 0.5 seconds are represented. The red line illustrates the path followed by the centre of the disc. During this phase, gravity is the only load applied to the model. The simulation indicates that the grinder is ejected towards the human operator with a high risk of shock, but also that the disc has stopped rotating. Figure 5 shows the complexity of the trajectory of the centre of the disc, and this example of results obtained by the simulation shows the final target of the model, i.e. an assessment of the risk of injury caused by the kickback.

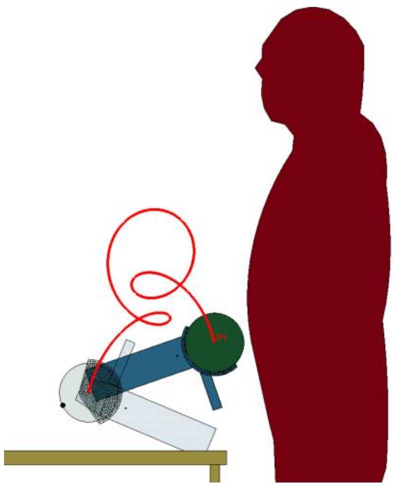

Fig.5: Trajectory followed by the grinder in the air after its ejection

3.2 Results obtained with the Python code

This section describes an example of results obtained by running the developed Python module. The conditions required by the model are the same as those used in the previous section, and numerous comparisons were performed to check that the Python code led to identical results to those obtained by LS-DYNA.

Figure 6 shows a map representing the kinetic energy transferred to the grinder when it loses contact with the brake pad depending on the location of the pinching point. Areas in grey indicate the points at which it is physically impossible to position the brake shoe and the points where ejection does not occur. Coloured areas show pinching points at which ejection occurs and the colour depends on the magnitude of the kinetic energy transmitted to the grinder. The most powerful ejection occurs at a point in the red area and is approximately referenced by the cylindrical coordinates r=0.6 R_d and θ=165°. The maximal transmitted energy is equal to 21 J.

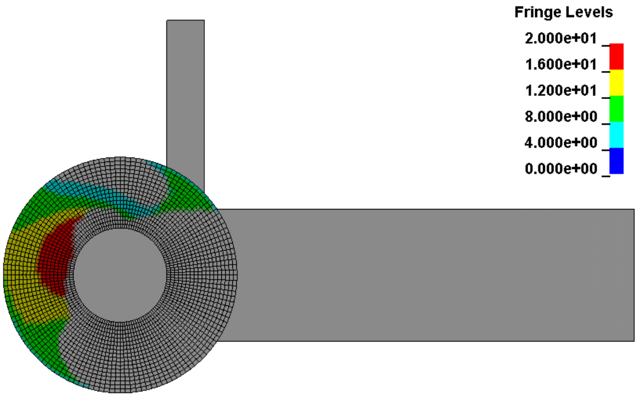

Fig.6: Map of kinetic energy in Joule transmitted to grinder at the ejection

The magnitude of the kinetic energy transmitted to the grinder is a pertinent indicator for assessing the risk of injury from kickback, but is not itself sufficient. Other indicators, such as the remaining angular velocity of the disc or the direction of the ejection, must also be assessed. Figure 7 shows the variation in the direction of ejection of the grinder depending on the location of the pinching point. As in Figure 6, the grey areas indicate the points at which it is physically impossible to position the brake shoe and the pinching points where ejection does not occur. Coloured areas show pinching points at which ejection occurs and the colour is assigned to the direction of the tangent of the trajectory of the centre of gravity of the grinder as it loses contact with the brake shoe. The coloured crown in the right upper corner of the figure shows the colour scale of the directions. The red colour, for example, represents the directions between the x and the z reference axes, as defined in Figure 2. These are oriented towards the human operator and therefore the red areas are assumed to be conditions of increased risk of injury for the worker. Figure 7 shows a large green area, indicating that the grinder will move away from the human operator, and also indicates a small red area where the grinder will be ejected towards the human operator. This is the position that was selected for the full LS-DYNA analysis, presented in the previous section.

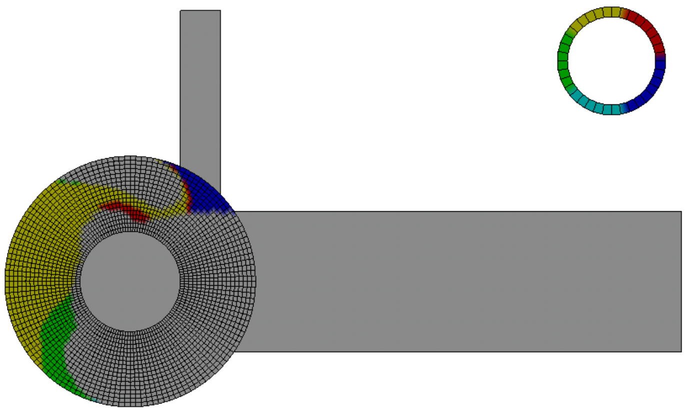

Fig.7: Map of the direction of ejection

4 Conclusion

The use of electrical angle grinders exposes workers to the risk of loss of control of their machine due to kickback, and consequently to a risk of injury. INRS has conducted preliminary research to develop a test bench for reproducing kickback in the laboratory by pinching the rotating wheel of a grinder. To improve this test bench and to increase our understanding of the mechanical phenomena leading to the ejection of the grinder, a numerical approach was initiated. The numerical approach was based on a grinder model implemented using two different methods—LS-DYNA and a Python code—to solve the differential equations that govern the phenomena leading to the ejection. Advantages were taken from both implementations. The two methods led to exactly the same results. The robustness of LS-DYNA, its ability to solve mechanical problems and the capabilities offered by LS-PrePost for the analysis of results were used for single analyses. The efficiency and direct accessibility of the Python code was used for multiple parametric analyses. We investigated in particular the effect of the position of the pinching point on the disc. The kickback magnitude and the direction of ejection were found to change significantly when the position of the pinching point was slightly modified. The knowledge acquired from the numerical simulations will be used to continue the development of our test bench. In order to check the validity of the assumptions on which the numerical model is based, our next challenge will be to improve the testing procedure and the measurements.

5 Glossary

G_d:	centre of gravity of the disc.
G_g:	centre of gravity of the grinder.
C:	contact point between the disc and the brake pads.
C_r:	instantaneous velocity centre.
m_g:	mass of the grinder.
m_d:	mass of the disc.
R_d:	radius of the disc.
ω_d:	initial angular velocity of the disc.
$C(\omega)$	drive torque of the electric motor as a function of the angular velocity
x_d, z_d:	coordinate of G_d.
x_g, z_g:	coordinate of G_g.
V_{xd}, V_{zd}:	components of the velocity vector of G_d.
V_{xg}, V_{zg}:	components of the velocity vector of G_g.
a_{xd}, a_{zd}:	components of the acceleration vector of G_d.
a_{xg}, a_{zg}:	components of the acceleration vector of G_g.
$\ddot{\theta}_g$:	angular acceleration of the grinder.
$\dot{\theta}_g$:	angular velocity of the grinder.
$\ddot{\theta}_d$:	angular acceleration of the disc.
$\dot{\theta}_d$:	angular velocity of the disc.
J_g:	moment of inertia of the grinder about a y-axis through its centre of gravity.
J_d:	moment of inertia of the disc about a y-axis through its centre of gravity.
d_g:	distance on the x-axis separating G_d and G_g.
μ:	friction coefficient between the disc and a brake pad.
F_N:	normal pinching force applied by a brake pad on the disc.
F_T:	tangential friction force in the plane of the disc.
F_{Tx}, F_{Tz}:	components of the tangential friction force.

6 Literature

[1] IEC/EN Standard 60745, Part 2-3: Hand-held Motor operated Electric Tools – Safety – Part 2-3: Particular requirements for grinders, polishers and disc-type sanders, 2006

[2] https://www.bosch-professional.com

[3] https:// www.makita.fr

[4] Dabrowski A.: Reducing Kickback of Portable Combustion Chain Saws and Related Injury Risks: Laboratory Tests and Deductions, International Journal of Occupational Safety and Ergonomics (JOSE), Vol. 18, N°. 3, 3999-417, 2012

[5] Drew A., Parmigiani J., : A Method for Detecting the Occurrence of Chainsaw Kickback, Conference ASME 2011, International Mechanic Engineering Congress and Exposition.

[6] ISO Standard N° 9518:1998, Forestry machinery – Portable chain-saws, Kickback Test.

[7] Bolay C.: Verfahren zur Erkennung unfallträchtiger Verkantungsfälle bei handgeführten Trennschleifern, IPA-IAO, Forschung und Praxis 245, ISBN-13:978-3-540-62767-8, 1997.

[8] http://gmsh.info

[9] https://www.python.org/