Commissioning new applications on processing machines: Part I - process modelling

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Abstract
The subject of this splitted article is the commissioning of a new application that may be part of a processing machine. Considering the example of the intermittent transport of small-sized goods, for example, chocolate bars, ideas for increasing the maximum performance are discussed. Starting from an analysis, disadvantages of a conventional motion approach are discussed, and thus, a new motion approach is presented. For realising this new motion approach, a virtual process model has to be built, which is the subject of this article. Therefore, the real process has to be abstracted, so only the main elements take attention in the modelling process. Following, important model parameters are determined and verified using virtual experiments. This finally leads to the possibility to calculate useful operating speed–dependent trajectories using the process model.

Keywords
Processing machines, trajectory planning, multibody simulation, discrete element method, process design

Introduction
Modern processing machines are often characterised by the use of classical linkages in the combination of modern servo drives. For illustration purpose, an often used example of such an assembly is shown in Figure 1 which is a five-bar linkage with 2 degrees of freedom (2-DoF). The task of this linkage is to convey small-sized goods, for example, chocolate bars, with an intermittent motion. Thereby the goods are conveyed along a horizontal path with a rise to dwell motion. After the goods are transported, they stand still and the working tool of the linkage (the comb) realises a return stroke.

In the industry of producing consumer goods, a high-performance $p$ (produced products per unit of time) of processing machines is intended and calculated by

$$p = q \cdot f$$  \hspace{1cm} (1)\hfill

whereby $f$ is the operating speed in Hertz and $q$ is the amount of produced products in one cycle of the machine.\footnote{Technische Universität Dresden, Dresden, Germany} This aim is decisive not only for economic but also for ecological and sustainable reasons. However, it is a wrong conclusion that by simply increasing the operating speed $f$, the performance $p$ increases equivalently (see Figure 2). Instead, the effective performance decreases after the operating speed exceeds an optimal operating speed. As a consequence, the performance of a machine has a maximum value and is thus limited. The reason is product damage growing with the operating speed which is caused by

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different characteristics of the machine. To investigate these dependencies, different aspects of the machine and process behaviour as well as their interaction have to be considered. In Figure 3, several reasons for a limited maximum performance of a processing machine are displayed.

As shown, the maximum performance is, among others, limited by the control and drive behaviour of the machine. With increasing operating speed, the position error of the working tool rises and consequently the motion accuracy subsides. This results in less positioning accuracy of the product and may induce instabilities in the process behaviour. Another aspect is the increasing vibration of the mechanical structure which is caused by the progressive oscillating inertial forces in the structure. This leads to a less positioning accuracy of the working tool and rising noise, which can also be a limiting factor to the maximum performance. Eventually, the product behaviour has to be considered. With increasing operating speed, the velocity of the comb rises. This results in a higher stress for the product surface (e.g. a chocolate bar), which may breaks and so limits the maximum performance.

Thus, the maximum performance of the machine could be increased, if the motion of the working tool is changed in such a way that the product's stress subsides. This results in less positioning accuracy of the product and may induce instabilities in the process behaviour. Another aspect is the increasing vibration of the mechanical structure which is caused by the progressive oscillating inertial forces in the structure. This leads to a less positioning accuracy of the working tool and rising noise, which can also be a limiting factor to the maximum performance. Eventually, the product behaviour has to be considered. With increasing operating speed, the velocity of the comb rises. This results in a higher stress for the product surface (e.g. a chocolate bar), which may breaks and so limits the maximum performance.

Current state of technology

The conventional way of conveying pieced goods intermittently is to shove them over a slide surface using a multi-chamber transport comb, while a cam follower mechanism is used for setting the motion of the five-bar linkage. The disadvantage of this approach is that a cam only carries the information for one specific motion, which means that the motion of the working tool is principally the same for each operating speed, but always scaled to the targeted operating speed. The comb motion realised with the aid of a cam consists out of two parts. First, the product is moved from the start to the end position, while the comb rests at the beginning and the ending. Second, the comb moves back to the starting position performing a return stroke. These two parts build one motion cycle (see Figure 4(a)). Following, only the first part of the motion is considered: the transport phase. The easiest possibility for realising the transport phase is a rise to dwell motion. The corresponding basic velocity profile of the working tool and the product is shown in Figure 5(a).

First, the boundary conditions of this realisation have to be considered. Because of product tolerances, it is not possible to realise a form closure transport. This would induce damage on the surface of the product. Thus, the gap between the two tines is greater than the product width. Therefrom, a constraint concerning the completely new motion approaches as described in Döring and Majschak. To reveal that this model-based approach is expedient in practical manner, the simulated motions have to be tested on a test rig.

In the first part of this article, the process model building and verification is treated. An algorithm is discussed which generates process considering and operating speed–dependent motions for the working tool. The motions thus generated are experimentally investigated on a test rig in the second part of this paper, whereby necessary steps to realise the new synthesised motions with modern servo drives instead of cam follower mechanisms are presented.
maximum reachable performance can be derived: the limitation of this motion approach is reasoned by the maximum deceleration of the comb \( \min \{a_{\text{comb}}\} \). If the absolute deceleration of the comb is higher than the deceleration caused by friction between the product and the slide surface, a detachment of the product from the comb occurs. This leads to a free unbound sliding of the product, which is a disadvantage of this motion approach. As a consequence, the product is not well positioned and so the comb may collide with the product at the return stroke. This would lead to damaged products, and thus, a lower operating speed has to be set. To quantify the consequences resulting from this effect, a quality criterion has to be defined. This helps to identify the operating speed for which a stable process on a maximum performance can be guaranteed. In this case, it seems to be constructive to use the positioning error of the product at the end of the transport.
phase which is the distance between the intended and the real positions of the product. The positioning error should be less than 0.5 mm to realise a stable process. Following from that, a qualitative trend of the positioning error over the operating speed can be discussed (see Figure 6(a)). The product is well positioned for low operating speed because the maximum deceleration of the comb is always smaller than the friction deceleration of the slide surface. But with increase in the operating speed, the maximum deceleration of the comb gets larger than the friction-induced deceleration, and so, a positioning error of the product is expected. This positioning error increases with the operating speed quadratically. It is obvious that the positioning error gets greater than the allowed error at a critical operating speed $f_{cr}$, which leads to the maximum reachable performance with respect to equation (1).

In practice, a higher maximum performance can be achieved by shifting the critical operating speed to a higher value. This can be realised by suppressing the presented detachment effect by additional friction forces on the product, for example, by increasing the friction forces with additional external loads. But this results in some disadvantages concerning the product load, for example, a higher compressive stress. Therefore, the question is whether a new motion approach is imaginable which allows a higher maximum performance with simultaneously lower product load compared to the conventional approach.

**New model-based motion approach**

A starting point for realising a new motion approach is the apparent disadvantage of the conventional motion approach: the unbounded free sliding of the product. The product detachment was identified as the main reason for the resulting positioning errors on higher operating speeds. But it seems to be a good idea to provoke this detachment of the product from the working tool and to use the products’ proper motion. The corresponding principle is shown in Figure 4(b). The product and the working tool are in rest and in contact at the beginning of the transport phase. Then, the product is accelerated by the comb. At the moment the comb decelerates, the product detaches from the comb and slides freely over the ground. Finally, the comb accelerates again and the second tine catches the product and decelerates it to the final position. For realising this motion approach, the basic velocity profile of the comb and the expected velocity profile of the product are shown in Figure 5(b).

Basically, two benefits of this new motion approach can be named: because of the free sliding phase of the product, the comb has to perform a smaller stroke with a constant transport distance of the process. This results from the additional gap between the two tines of the comb. The larger the gap, the lower the stroke. Other benefits are the smaller product loads. In theory, the product is only strained by the acceleration forces of the comb and the friction. Therefore, it is important that the second tine catches the product with the appropriate velocity and acceleration in the right moment. But if this transition conditions are not fulfilled, a
positioning error occurs because of an impact between the product and the working tool. Thus, it is easy to see that because of the free sliding phase, the corresponding start and end velocities of this phase are not scalable to specific operating speeds. Hence, another positioning error over operating speed trend is expected than it was discussed for the current state of technology (see Figure 6(b)). If a known optimal motion for a specific operating speed is assumed, it is obvious that by scaling it to other operating speeds (regardless of whether higher or lower), a positioning error will occur.

Assuming a tolerable positioning error, there may be a range of operating speeds from \( f_{kr} \) to \( f_{u.cr} \) (acceptance region) that fulfill the requirement. To apply the new motion approach for operating speeds below the lower critical \( f_{kr} \) and above the upper critical \( f_{u.cr} \) operating speed, other motion profiles have to be used, which fulfill the criteria in these areas. Combining more than one profile leads to overlapping acceptance regions. This effect can be used to increase the maximum performance of the machine. Such an approach cannot be realised with classical cam follower mechanisms because they involve only one motion profile. In contrast to this, modern servo drives basically offer the possibility for the execution of more than one motion profile. In the current state of technology, commercial servo drives do not offer the required functionality for the definition and execution of more than one motion profile. In Holowenko et al., a possibility for realising this new functionality was presented.

In theory, with this motion approach, the maximum performance can be increased with no limitation by the process. But there are still limits imposed by the machine, such as maximum drive torques. However, there is a lower limit for the feasible performance, since the new motion approach only works if the product detaches from the working tool. This detachment effect only occurs if the deceleration of the comb is larger than the friction-induced deceleration. For each applied motion profile, there is a specific operating speed \( f_{de} \) (see Figure 6(a)) above which the detachment occurs, because the critical absolute deceleration \( |a_{cr}| \) is exceeded. The new approach only works above operating speeds for which the motion exceeds the critical absolute deceleration.

Summarising, the new motion approach enables one to achieve a higher maximum performance because the detachment problem of the conventional motion approach is avoided. Finally, the product’s load significantly subsides since the product is caught by the comb without any impacts. To realise this new motion approach, two steps are necessary: First of all, an algorithm has to be designed which allows the calculation of an optimal motion for a given operating speed. Because of the multiple process parameters, which influence the yet unknown optimal trajectory, a process model has to be built which will be used in a simulation. The results of the simulation will be an important part within the algorithm. Therein, the model is used to calculate different optimal motions for the working tool. Finally, it will be proven that this new process does not work with a single motion profile. Hence, this new approach is not suitable for traditional cam follower mechanisms, but for modern servo drives it allows the execution of speed-dependent motions.

**Building the process model**

In order to create the process model, an abstraction of the process is first necessary, so only the main effects and elements will take attention in the simulation. Therefore, the main interacting elements of the process are chosen (see Figure 7). The system to be simulated consists of the product, two slide surfaces and a quarter of the comb. Therefore, the abstracted comb is represented by \( 2 \times 2 = 4 \) tines which are necessary to simulate one motion cycle. For implementing the model in a multi-body simulation, the joints between the local frames of the different elements (bodies) regarding the world frame \( O_w \) have to be set. Here, three different types of joints are used. The frames of the slide surfaces \( O_{GS} \) are fixedly connected to the world frame. The local frame of the comb \( O_C \) is connected with a prismatic joint to the world frame, whereby the corresponding DoF are set with specific motions. Finally, the product frame \( O_P \) is connected to the world frame \( O_W \) with a 6-DoF joint. Thus, the product can move freely in the simulated space.

For realisation of the simulation, the model is implemented in MATLAB/Simulink and the Multibody Toolbox. This allows the easy implementation of three-dimensional (3D) bodies including their mass and inertia definitions. To initialise the model, the toolbox provides an import function for computer-aided design...
(CAD) parts with their specific rigid frames. By linking the body frames as described, as well as setting the initial positions, the model is fully built. While the simulation is running, the multibody equations are automatically solved, so the motions can be integrated accordingly. To realise the contact interaction, the **Contact Forces Library** is used. It allows the contact detection and calculation with the help of a *Voigt–Kelvin* model (see Figure 8) as it is used in the discrete element method (DEM). By the calculation of elastic impacts as well as frictions, it becomes possible to simulate rigid body interactions. The method was originally used to simulate molecular systems. Therefore, spheres are still the means of choice because their contact is easy to detect by the calculated intersection \( \delta \) of the two according radii. Another basic contact case is the intersection of a plane and a sphere. Here, the calculated trajectories of interacting rigid bodies are very decisive for the success of the new motion approach. Therefore, the force-based approach of the DEM is used instead of the impetus-based approach, as described in Baraff, to describe realistic material behaviour.

If a contact is detected \( (\delta \geq 0) \), the spring damper element helps to calculate a repulsive normal force \( F_n \). For calculating this force, a variety of laws are available for supply. The most common law is the linear spring-dashpot model (least significant difference (LSD), see equation (2)), where \( c_n \) describes the spring and \( d_n \) the damping parameter

\[
F_{n, LSD} = \begin{cases} 
  c_n \cdot \delta + d_n \cdot \dot{\delta}, & \delta > 0, \dot{\delta} > 0 \\
  c_n \cdot \delta, & \delta > 0, \dot{\delta} < 0 \\
  0, & \delta < 0, \dot{\delta} < 0 
\end{cases}
\]

(2)

This law can be used for simple applications, for example, contacts with small relative velocities. The disadvantage of this law is the fact that the coefficient of restitution \( e_n \) (see equation (3)) is constant over all relative contact velocities. This coefficient describes the amount of kinetic energy which is available for the contact partners and is thus a measurement for the velocity after the impact, whereby \( v'_2 \) describes the velocity before and \( v'_1 \) describes the velocity after the impact

\[
e_n = \frac{v'_2 - v'_1}{v'_1 - v'_2}
\]

(3)

In practice, the coefficient of restitution is not constant because impacts are mostly related with plastic deformation, so with increasing relative velocity, the ratio of free kinetic energy to inner deformation energy changes. To describe this contact behaviour, the **Hertzian Spring-Dashpot model** (HSD) was constructed and named after Hertz (see equation (4)). Because of the parameters \( \alpha \) and \( \beta \), it is possible to realise a variety of velocity-dependent coefficients of restitutions

\[
F_{n, HSD} = \begin{cases} 
  c_n \cdot \delta^\alpha + d_n \cdot \text{step}(\delta) \cdot \delta^\beta \cdot \dot{\delta}, & \delta > 0, \dot{\delta} > 0 \\
  c_n \cdot \delta^\alpha, & \delta > 0, \dot{\delta} < 0 \\
  0, & \delta < 0, \dot{\delta} < 0 
\end{cases}
\]

(4)

Besides the calculation of normal forces \( F_n \), additional tangential forces \( F_t \) in form of friction forces occur in case of an impact. The easiest way to simulate such friction forces is Coulomb’s Law which says that the friction force is proportional to the normal force, whereby \( \mu \) describes the coefficient of friction

\[
F_t \leq \mu \cdot F_n
\]

(5)

There are multiple ways to define the coefficient of friction \( \mu \) with respect to the relative velocity \( v_{rel} \) according to the desired model behaviour. Figure 9 shows two different definitions, whereby case A shows the most simplest definition of \( \mu \) over \( v_{rel} \) and case B shows the regularisation form of case A. Because the regularisation form is associated with the distinction of dynamic \( \mu_D \) and static \( \mu_S \), friction, it has the advantage that numerical oscillation for relative velocities of \( v_{rel} \approx 0 \) is disabled. This is realised with a small threshold velocity \( v_{th} \), which defines a steep edge around the zero-crossing.

In the **Contact Forces Library**, case B is provided. A further advantage of this definition is the possibility to redefine the function \( \mu(v_{rel}) \) to a desired form; for example, velocity-dependent friction, as described in Popov, can be simulated.

To implement the process model, all bodies are prepared in an external CAD system, so the import can be realised by simply setting the initial position as well as the density, hence the mass and the inertia tensor can be derived. For realising the body interactions, simple contact cases of the **Contact Forces Library** are used. As seen in Figure 10, the contact between the product and the sliding surface is implemented as a sphere-to-plane contact. The corresponding contact law can only be set after appropriate tests at the real active unit. For setting the size of the sphere radii \( R \), Miller names as an approximate value

\[
R_{min} = 0.05 \text{ min\{length, width\}}
\]

(6)
After these steps, the model is set up, and so, the first step of the simulation is done. While implementing the model, the necessary parameters were discussed, especially the contact parameters are decisive for the significance of the model. Therefore, the model has to be verified regarding these parameters.

**Model verification**

As described in Bossel and Sargent, the aim of the model verification is to check whether the model has been built right and thus works physically correct or not. One possibility to realise this is to check whether the model parameters induce the expected physical behaviour of the system. By comparing the expected (theoretical) model behaviour with the simulated behaviour, a quantification of the model goodness can be determined. Therefore, the main model parameters have to be selected and checked by executing virtual experiments. Following from the model building and the simulation task, the parameters that must be checked are the ones which are related to the active unit. So, the coefficients of restitution, static friction and kinetic friction are important. After a successful verification, the model goodness concerning the physical correctness can be claimed. In Malone and Xu, experiments for the validation of a DEM model are described which can be used virtually to verify the model.

To evaluate the coefficient of static friction, the experiment of the inclined plane is consulted (Figure 11(a)). For a given coefficient of static friction $\mu_S$, the plane can be tilted up to a certain angle $\alpha_0$ before the product starts to slide. The relationship between these two quantities can be derived from a force equilibrium and is given by

$$\mu_S = \tan(\alpha_0) \quad (7)$$

For verification, the model parameter $\mu_S$ is set to $\mu_S = \tan(12^\circ)$. At the beginning, the product is in rest, while the plane starts to tilt with a slow angular velocity. The result of the virtual experiment is shown in Figure 11(b). It is obvious that the product remains in its position till the angle reaches $\alpha_0$. This behaviour proves that the coefficient is well realised within the model.

To verify the coefficient of kinetic friction, an impact experiment is implemented. The product is in rest on a slide surface and gets an impact with a specific collision velocity $v_{col}$. The product slides for a specific path with the length of $\Delta s$ till it stops, and the whole kinetic energy is converted into friction energy. The relationship between these quantities can also be derived from a force equilibrium and is given by

$$\mu_D = \frac{v_{col}^2}{2 \cdot \Delta s \cdot g} \quad (8)$$

whereby $g$ describes the gravitational acceleration. The model parameter is set to $\mu_D = 0.15$ and the virtual experiment is executed for different collision velocities. The theoretically expected (the given parameter $\mu_D$) and the simulated results are shown in Figure 12. The relative error of maximum of 0.16% is sufficient, and thus, the verification of this model parameter is successful.
The coefficient of restitution is a parameter which cannot be set directly in the model. Thus, the elastic behaviour of a contact, which results out of the coefficient of restitution, is determined by the coefficients of the contact law $c_n$ and $d_n$. For the LSD model, an analytical conjunction between these parameters and the coefficient of restitution is given by Luding\textsuperscript{17}

$$d_n = \frac{-\sqrt{5} \cdot \ln(e_n) \cdot \sqrt{m_{\text{eff}} \cdot e_n}}{\sqrt{\ln(e_n)^2 + \pi^2}}$$

whereby $m_{\text{eff}}$ is the effective mass of the contact partners (in this case, the mass of the product). This relationship results from an integration of the contact differential equation, which is only possible because of their linearity. First, the LSD model is tested with an assumed coefficient of restitution of $e_n = 0.4$. To test this quantity, an impact experiment is executed. Thereby, the working tool hits the resting product with a constant collision velocity $v_{\text{col}}$. The expected product velocity after the impact is given by

$$v'' = (1 + e_n) \cdot v_{\text{col}}$$

In Figure 13, the results of this experiment for different collision velocities are shown. Because of the assumed constant restitution behaviour, the expected product velocity is linear with respect to the collision velocity. As can be seen, the relative error of the simulated results related to the theoretical value (the given parameter $e_n$) is nearly constant at 1% and consequently independent of the collision velocity. Therefore, error of 1% has to be assumed for the model goodness. There is no analytical conjunction between the HSD model equation (4) and the coefficient of restitution, because the associated differential equation is almost entirely solvable numerically. In Antypov and Elliott\textsuperscript{18} an analytical solution is shown which only works for a specific choice of $\alpha$ and $\beta$. Thus, the model cannot be verified with respect to the coefficient of restitution. Instead, it seems to be expedient to adjust the parameters of the HSD model equation to the results of some experimental data. So, the model precision states exemplarily to the specific use case.
Eventually, the verification of the model was successful. All parameters were tested in virtual experiments, while the results are satisfying. It is expected that the calculated errors resulting from the model are smaller than the inaccuracy of measurements which will occur while setting the real model parameters. Thus, the model can now be tested concerning the usability for trajectory planning within the new motion approach.

**Model-based trajectory planning**

For realising the new model-based motion approach, the necessary trajectory has to be planned. Therefore, a distinction of the path and the related velocity profile seems to be expedient. Because of the horizontal two-dimensional (2D) comb motion, the path is given by a line segment with the length of the comb distance. Much more interesting is the associated velocity profile of the motion. To fulfil the new motion approach, the profile has to have a certain shape. Therefore, the schematic profile of the velocity profile is shown in Figure 14.

For each considered operating speed, the transportation time $t_{tr}$ and the event times $t_1$, $t_2$ and $t_3$ have to be defined. They represent the following specific events of the transport phase for which useful velocity and acceleration transition conditions have to be found:

- $t = 0$: at the beginning of the motion, the comb and the product are in rest, so the velocity and the acceleration are zero.
- $t = t_1$: this time is characterised by the beginning of the free sliding phase of the product. So, the acceleration $a_1$ must be negative (to overcome the friction) at a specific velocity value $v_1$.
- $t = t_2$: at this time, the comb has to do a reversal acceleration motion, so it stops slowing down and starts to accelerate at a specific velocity value $v_2$. The deceleration between $t_1$ and $t_3$ is necessary to give the product the possibility to overcome the comb gap between the two tines.
- $t = t_3$: after the product slides freely between the comb gap, it has to get caught by the next tine. Thus, the comb has to have the same velocity and acceleration as the product to avoid an impact and realises a smooth catching.
- $t = t_{tr}$: at the end of the transport phase, the comb and the product are in rest.

To connect the specific events of the velocity profile, it seems to be useful to use a quintic function,\textsuperscript{19} which has six free adjustable parameters, so all velocity and acceleration constraints are included and also a jerk continuity can be reached

$$v_i(t) = \sum_{i=0}^{5} a_i \cdot t^i$$ \hspace{1cm} (11)

For the synthesis of the complete velocity profile, the adjustable velocity and acceleration parameters for the three events have to be set. Therefore, an iterative algorithm was designed, which functionality can be seen in Figure 15. Starting with the initial values and the fixed values given by the reached operating speed, the four velocity segments are calculated and merged to the complete velocity profile. To meet the condition that the path ends in the moment $t_{tr}$ at the intended ending position, the velocity profile is scaled. Afterwards, the simulation is executed and the motion of the product is analysed. Since a useful stopping criterion for the algorithm and a gradient for updating the searched values is wanted, residues $r$ from the simulation are calculated and compared to a wanted precision $\varepsilon$. The residues include the final position and velocity of the product and also the impact velocity and acceleration at $t_3$. If the calculated residues are smaller than the required precision, the algorithm stops.
The algorithm represents an iteratively working solver for a nonlinear system of equations. It is implemented with the gradient-based trust region algorithm\(^\text{20}\) which minimises the residues. Therefore, it is important to choose valid initial values, so converge can be reached. With the help of the algorithm and the product model, it is possible to create an optimal velocity profile for each given operating speed. If at least two profiles are given, a characteristic map can be generated by interpolating the velocity profiles between them. For a free (but useful) choice of parameters, an exemplarily characteristic map is shown in Figure 16.

As it can be seen, the shape of the velocity profiles for each operating speed is different. This proves the necessary specification of different optimal motions for different given operating speeds \(f\). Therefrom, it is obvious that a classical cam follower mechanism cannot fulfill the challenges given by this new motion approach. Thus, only a modern servo drive with the appropriate control, for example, presented in Holowenko et al.,\(^\text{7}\) has the opportunity to realise such an application.

**Summary**

In this article, the maximum performance-limiting factors of a processing machine were discussed. As an example, the intermittent transport of pieced goods was observed and the disadvantages were illustrated. Therefrom, a new motion approach for the process was derived, which theoretically helps to increase the maximum machine performance. To realise this new approach, a process model with the help of a 3D multi-body simulation in addition with the appropriate contact realisation was built and verified. This process model was then used in an algorithm which calculates operating speed-dependent velocity profiles. These profiles can be assembled to a characteristic map, which is necessary to successfully realise the process for different operating speeds.

In the second part of this article, the model will be validated. This means that the model parameters are adjusted to the real active unit behaviour. This allows the calculation of optimal trajectories for the new motion approach. In addition, the success of these optimal motions is evaluated on a test rig, whereby the implementation of the speed dependency will be a main part. Furthermore, expedient possibilities of process observation and control will be discussed.

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