ELECTRONIC SYNCHRONOUS SHAFT FOR SWIVEL AXES DRIVEN BY COUPLED SELFLOCKING WORMGEAR DRIVES

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ABSTRACT

Coupling worm gears in large machine tool manipulator units (axes) can lead to overload damages under some operating conditions, like asymmetrical driving and emergency stop. This behaviour is caused by the self locking characteristic of worm gears with small pitch. If operated in the direction opposite to the designated one, self-locking can occur in the gears, subjecting the gears to an arbitrary torque which can be destructive. This condition can occur without counter torque on the driving side. In most gears, tooth engagement is based on a rolling motion. The slip component increases with a growing axis angle between the paired gears. In case of worm gears, the axis angle is 90°, and therefore the coupling is purely frictious. This friction coupling is important in modelling. Especially the transition between the static and dynamic friction, where the system parameters change abruptly by a factor of two, should not be neglected. Based on the evaluated model a synchronous controller has been developed. This controller gives the opportunity to drive swiveling axes with coupled selflocking gears by a standard gantry topology with an overlayed state changing master-slave topology.

KEYWORDS

coupled selflocking gears, swivel axis, synchronous controller, electronic synchron shaft, worm gear

1 INTRODUCTION

Machining of large and heavy workpieces by using machine tools with more than three axes increases steadily. Multiple axes milling, for example, requires not only motion of the tools or translation of workpieces but also rotary motions of workpieces. Rotary axes for moving workpieces are often implemented as swivel axes, rotary tables, or as combinations of both. Most of the implemented swivel axes and rotary tables nowadays are driven by using spur gear drives or worm gear drives [12]. For this research paper, dealing with coupled selflocking gear driven axes, rotary tables, and swivel axes can be treated similarly. They just differ in the stiffness between both drives. So in this research, swivel axes will be treated representatively and rotary
tables will be reduced to swivel axes with high bridge stiffness. Due to low drive torque, single drive driven axes are often not able to handle heavyweight loads with the required dynamics. They also do not allow accurate machining while in motion [6]. The elastic drive chain leads to loss of accuracy and an increased influence of the process forces on the positioning accuracy at the processing point (TCP, tool center point). Together with gear backlash, chatter can also occur, which results in chatter marks on the machined surface, as described in [8]. To reduce these problems, distributed parallel acting actuators are used, which increase the input value, improve axis stiffness, and reduce backlash. The reduction, or, ideally, the elimination of backlash increases the precision, enabling processing without first clamping the axis and thus enabling machining while the axis is moving. This slack compensation can be performed by interlocking the two actuators of an axis and is generally torque-controlled. Among axes with distributed drives, predominantly drives with non-self-locking spur-gears can be found, as there is currently no way of operating axes with self-locking drives safely and non-destructively. Pretension with self-locking, coupled drives is not feasible using torque control. Due to the high rigidity of worm gears, a combination of worm gears with the concept of distributed drives is expected to result in a remarkable gain in stiffness. Figure 1 shows the testbed used in the experiments.

2 STATIC AND DYNAMIC FRICTION IN STRAIGHT-WEDGE MECHANISMS

For illustration purposes, the worm gear in this study will be modelled as the related wedge-type mechanism. Using this model, the phenomena of the worm gear can be presented.
The two superposed wedges with a twofold inclined plane thereby replace the tooth contact with the pressure angle $n$ and the pitch angle $m$. The equivalent image reduced to two dimensions in Fig. 3 contains all the necessary parameters for modeling [11], [3]. The tooth surface normal force is calculated from a deflected spring damper unit, as shown in Fig. 5, that models the elastic tooth contact with backlash. Based on this axial force the internal friction $F_R$ can be determined based on the system states. This internal friction is responsible for the self-locking behavior of gearboxes with high ratios.
Graphically, this situation is presented by the friction cone presented in Fig. 4. If the force of the load force vector is within this cone, friction is dominant, and the system stops. Based on this graph, the difference between self-locking gears and self-braking gears can also be explained. If the force vector is found within the smaller red cone as a result of the pitch angle i.e. it is within the dynamic friction cone, friction prevails in each system state and the gear comes to a halt in the absence of the driving force $F_A$ and independent of the external load force. The system is self-braking. If the force vector is within the range outside the red and inside the blue cone, the system in a moving state does not necessarily come to a stop. However, if the system remains at rest, the system will remain in this state regardless of the external load force. In this paper, worst-case estimate is used to examine only the case of self-braking gears.

3 OPERATION OF COUPLED GEARS

If one applies the above considerations to the system shown in Fig. 5, it is clear that torque-controlled pretensioning of outgoing-side mechanically-coupled self-locking drives cannot be realized. Here, the backlash-free spring damper unit presented between the two gear units is a simplified equivalent mechanical diagram of the bridge. If one side of the pivot axis is driven, no statement can be made about the second gear, if the applied driving force results from internal friction and causes pretensioning, or if the driving force originates from an additional internal friction force. This means that the contacting flank is not uniquely identifiable. Furthermore, the friction coefficients are not known with sufficient accuracy, so that the tooth flank normal force cannot be computed based on the driving forces. In any case, information about the deflection of the spring-damper units is necessary.
By implication, this means that the pretensioning can be set by adjusting the offset of the two drives with respect to each other. Thus, it is possible to realize a control scheme modeled on the mechanical balance shaft between two drives. Instead of a simple torque pretensioning, a drive-dependent differential torque addition is realized (see Fig. 6). This concept can also be used, as opposed to the previously used control topologies such as gantry or master-slave, to compensate unbalanced operating conditions such as a drive failure or unevenly distributed loads or transmission parameters.

4 SYNCHRONOUS CONTROLLER

The implementation of the synchronous controller in the form of a difference-dependent torque overlay is shown schematically in Fig. 6 and Fig. 7 and comes up to an electronic implementation of a mechanical synchron shaft or balancer shaft which where used in the past for synchronisation problems [7] and [9].

Figure 5. Coupled straight wedge mechanism schema of coupled worm gears

Figure 6. Synchronous controller concept shown at straight wedge mechanism
The controller consists of a standard P-PI cascade controller for each drive and is operated in a gantry topology as described in [4], [10], [12] and [13]. Each drive gets the same nominal values. In addition, there is an superposed P-PI controller, which, based on the drive deviation error $e_v$, outputs a torque set point which is added to the controller output current of the lagging motor and is subtracted from the controller output current of the leading motor. Additionally, the controller outputs are weighted with increasing drive positioning difference, so that the influence of the normal axis control over the weighting function $1 - e_v k_{gv}$ is reduced from 1 to 0 and the influence of the synchronous controller of the weighting function $e_v k_{gv}$ is increased from 0 to 1.

The weighting function $k_{gv}$ is arbitrary and defines the sensitivity and responsiveness of the synchronous controller to the drive difference. However, it has been shown however that a function with continuous transitions of the controller stability is conducive. Accordingly, the initial quadratic weighting function with a progressive response was replaced by a sinusoidal function (Fig.8).
This controller assembly with conventional P-PI cascade controllers allows the topology to be implemented on industrial controllers and to facilitate the commissioning by using known technologies. Successful implementation was done on a Siemens Sinamics S120 drive system with a Siemens CU320-2DP as a control unit. The controller implementation was carried out under the Siemens software package Simotion Scout by creating the control structure in Drive Control Charts. The following measurements are realized by an dSPACE realtime control system with the synchronous controller implementation shown in Fig.9.

5 SIMULATIONS AND MEASUREMENTS

The most fatal unbalance to be handled is a breakdown of one drive. Therefore the drive breakdown scenario has been defined as a standard test for synchronous controller topologies. Simulation results and measurements, carried out in a research of Nathanael Bach [1] and [2], show that the developed controller works in principle. But the stable area for the possible weighting gain parameters of the synchronous controller and the axes controllers, which has to be defined manually, is very small, so that the controller cannot be called robust.

The drive differential angle gain, which defines the gradient of the used weighting functions, is the most sensitive controller parameter. It scales the maximum drive difference to one. This parameter has to be smaller than the maximum possible elastic tension remaining after the user defined pretension. Stable operation can be expected with one half of possible tension reserve. Exceeding this value causes the drives to lock.
Figure 10. Simulated operation with drive break down in not pretensioned operating mode

(a) Detail: Drive break down, not pretensioned
(b) [Detail: Set value crossing in negative direction, not pretensioned]
By using correct adjusted parameters, the controller holds the position with the maximum predefined pretension. Because the setpoints of the differential controller are calculated from drive positions of an inertial system, no setpoint steps are expected. The setpoint curve can be treated as quasiacceleration-limited. By using the overlayed synchronous controller, each of both drives acts on the errors of the other and so limits the maximum possible drive position difference to the predefined maximum value, even if one drive breaks down. In fault cases this behaviour possibly causes setpoint differences from the NC-setpoints but on the other hand leads to a reduction of normal tooth forces in the gears and prevents the wormgears from damage. To predict axes behaviour, a scalable axis simulation model has been developed and combined with a discrete controller model by using the software packages Matlab and Simulink. The principle of this model is a coulombic friction model to represent the static and dynamic friction of the real system. The implementation is founded on an example of H.A. Gall [5]. The model has been verified on the IPP tested and on a large swivelaxis system from Rückle GmbH. The measurement and simulation results show a very big compliance and so the simulation results could be expanded to other swivel axes and rotary tables.

The measurement data in Fig.11 show in detail fragments of the trajectory driven without pretensioning(Fig.10). The curve setup is a drive breakdown of drive one at 6.95 sec. The subfigures(b) and (c) show the setpoint trajectory crossing the actual position, accompanied by master drive changing. The still active drive follows the given curve by doubled predefined pretension angle kgv up to the inverted controller balance. After reactivation of the failed drive, both drives are going back to the given path and will follow it. This option will not be used on reality, because due to that the predefined maximum motion speed for the axis will be set for driving to actual setpoints and in that zone no path accuracy will be given. In case of an error a
controlled system emergency stop, which forces an synchronous drive shut down, hast to be initiated. Only after reaching a complete halt the drives can be switched off.

Figure 12. Simulated operation with drive break down in pretensioned operating mode

(a) Detail: Drive break down, pretensioned
(b) Detail: Set value crossing in negativ direction, pretensioned
The illustration Fig.12 shows the same experiment as used in Fig.10 but in this case tensed up. This test also proves, in simulations as well as in practice, that the controller keeps the synchronization. While drive failure occurs, the active drive of the tensed operation mode driven testbed oscillates a bit more than in not tensed state. This behaviour is founded in the internal friction. The missing backlash area, which could be used as control range, causes instantly rising normal and friction forces. The system time constant has been reduced without increasing the controller performance. Nevertheless the oscillation amplitudes of the drive position differences are between approx. 0.5 times of the allowed differences and therefore at max. 75% of the remaining tension reserve. By that the essential working of the controller is shown.

Considering the corresponding simulated tooth flank normal forces of an pretensioned system in case of a simulated drive break down shows a force reduction of 80% by using the synchronous controller compared to the standard gantry controller (see Fig.14 and Fig.15). In cases of non pretensioned operation, reduction factors of 95% are reachable.
Figure 14. Normal force characteristic during drive break down for different operation states at the IPP-Test bed, operating without synchronous controller.
Figure 15. Normal force characteristic during drive break down for different operation states at the IPP-Testbed, operating with synchronous controller

So, the tooth flank normal forces don’t exceed the operation range or the design range and thus a damage free operation is possible. When operating the axes with coupled selflocking drives without the synchronous controller, a drive break down can’t be handled and the axis seizes. Caused by the static friction value of 0.18, which is compared to the dynamic friction value of 0.07 more than 2 times higher, as well as the system dynamics, the locked axis can’t be relieved by the motor torque. The extremely high normal forces and the thereby displaced lubricant increase the static friction value additionally. The probability of scuffing, broken teeth of the wheel and bronze weldings to the worm is quite high.

6 Emergency Shut Down

As described above for handling a drive fault safely, the system has to be shut down controlled by the remaining active drive. The required electrical power can be pulled from the grid. But to handle a grid fault, no external power can be used. So power errors like electrical power outage or
a power breakdown after emergency power-off switch has been used is much more critical to catch. In this cases, were no electrical power can be consumed from the grid, the power has to be provided by the link capacity until the system halts.

Figure 16. Simulated link power consumption of used testbeds while controlled emergency stop

The plots in Fig.16 show a simulated emergency shut down after one drive breaked and coasts down freely. This experiment gives an appraisal of the effective required shut down energy. Here is to be considered that the system is able to regenerate and so the required energy is just the integral of power over shut down time. Caused by the selflocking characteristic, the controlled drive has just to overcome the system’s friction. Thereby the total power consumption is quite low and, under ideal conditions, amounts just 0.04 Ws for the IPP-Testbed and about 100Ws for the real axes. Caused by the higher internal friction, pretensioned operation increases the power consumption depending on the drive offset and has to be removed when an power supply error occurs. For real systems the efficiency factor of the regeneration, the drives and the controller has to be considered as well, so that the real used energy is some higher. With an estimated consumption factor of two and two active drives each consuming the same, the used energy is 4 times higher and has to be considered at dimensioning the link capacitor.
7 CONCLUSIONS

Using axes with distributed self-inhibitive actuation and pretension have apparently advantages. But the benefit of the high stiffness and the lower backlash of worm gears in comparison to multistage spur gears cannot be completely used. A simple backlash compensation by using additional drive torques, which is used with spur gears, is not implementable for worm gears. A backlash compensation applied to worm gears is just realisable by adding position differences to the drives. Because of small accepted drive position deviations, each controlled system in reality has some deviations from the setpoints, a position difference tolerance range is necessary. This range has to be as small as possible, but so wide that neither pretension is removed nor axes tension gets such high, that the drives jam. That needs a minimum flexibility between both drives, favourably located in the bridge, that reduces the whole axis stiffness and thereby in parts the advantages of using worm gears. The necessary minimum flexibility of the bridge depends on internal system clock therewith from the synchronous controller stiffness. In actual control units the system clock is so slow that, caused by the synchronous control loops time constants, the bridge stiffness and controller dynamic had to be reduced severely.

An additional disadvantage of backlash reduced or backlash compensated operation of worm gears is founded in the friction. During pretensioned operation the tooth flanks are touching each with no controlled normal force. With a torque controlled pretension in spur gears, the normal forces are proportional to the pretension torque. By using position controlled pretension in worm gears, extremely high tooth flank normal forces caused by geometrical deviations and position difference errors can occur. In connection with the high sliding ratio of the motion between tooth contact surfaces in worm gears, the already high abrasion increases additionally. In addition to the geometrical errors thermal effects have consequences to the backlash and therefore to the normal forces. That means, that for operation of axes, driven by coupled selflocking gears, a position depending and temperature compensated pretension angle control is needed to limit this disturbing influences. Due to the high abrasion a frequent identification of axis backlash over the worm gear wheel girth is needed to create an actual backlash look-up-table which is needed for automatically generated presets for backlash compensation. Besides the lifetime of the gear is reduced and service costs increase. Therefore a protected axes operation of coupled self locking drives is possible, but because of the high abrasion a pretensioned operation should not be used.

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