Proceedings of the ASME 2012 11th Biennial Conference on Engineering Systems Design and Analysis
ESDA2012
July 02-04, 2012, Nantes, France

ESDA2012-82392

DRAFT: A UNIFIED DESIGN PROCEDURE FOR FLYING MACHINING OPERATIONS

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ABSTRACT

One of the most important aspects of the design of "flying machining" operation is the choice of the proper law of motion of the slave axis. In literature, technical reports and papers can be found concerning this subject, but they deal with specific problems and the solutions or suggestions proposed are specific as well, suitable for those cases. In order to try to overcome this limitation, in this paper we analyze the subject of the flying machining operations from a wider point of view. We propose a unified design procedure that has general validity being suitable for the choice of the slave axis' law of motion for whatever "flying machining" operation. Moreover the unified design process proposed includes also methodologies for the choice of motor and transmission.

NOMENCLATURE

$T_{M,N}$ motor nominal torque
$T_{M,max}$ servo-motor maximum torque
$\omega_{M,N}$ motor nominal angular speed
J_M motor momentum of inertia
T_M motor torque
$T_{M,rms}$ motor root mean square torque
ω_M motor angular speed
$\dot{\omega}_M$ motor angular acceleration
T_L load torque
T_L^* generalized load torque
$T_{L,rms}^*$ generalized load root mean square torque
$T_{L,max}$ load maximum torque
ω_L load angular speed
$\dot{\omega}_L$ load angular acceleration
$\dot{\omega}_{L,rms}$ load root mean square acceleration
J_L load momentum of inertia
$\tau = \omega_L/\omega_M$ transmission ratio
τ_{opt} optimal transmission ratio
η transmission mechanical efficiency
α accelerating factor
β load factor
τ_{min} minimum acceptable transmission ratio
τ_{max} maximum acceptable transmission ratio
$\tau_{M,lim}$ minimum kinematic transmission ratio (defined for each motor)
$\omega_{M,max}$ maximum speed achievable by the motor
$\omega_{L,max}$ maximum speed achieved by the load
J_T transmission inertia
t_C cycle time
τ_g generalized transmission ratio

INTRODUCTION

To improve the outturn of the automated production systems the flying cut method is often used. Flying machining operations are primarily characterized by the processing and cutting of material or product while in motion. In other words the main production process is not interrupted, hence the machine throughput

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is maximized.

Generally speaking, “Flying machining” represents synchronization of an axis (slave) with a master axis in motion. A simple example for “flying machining” is a knife that slices to length rolls of material in continuous operation. In order to cut on the length, the cutting tool has to be accelerated to synchronize itself with the speed of the product or material to be worked (line speed). Then it has to move at line speed while the cut is completed; afterwards the tool returns to its starting position and it is ready for the next cutting cycle. Most of these systems are developed with only one degree of freedom.

Typical “flying machining and processing” applications include: cutting to length, cutting in two, printing, embossing, gripping and checking moved workpieces, die cutting, sealing and cutting products, etc. Figure 1 depicts the sketch of a wrapping machine of “Flow pack” type, where the plastic wrap and the thermal welding heads for cutting and sealing are shown. In figure 2, the sketch of a rotary die-cutting machine is represented.

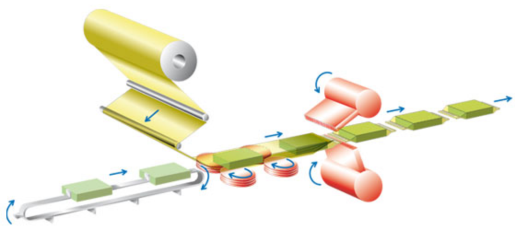


Figure 1. Wrapping machine.

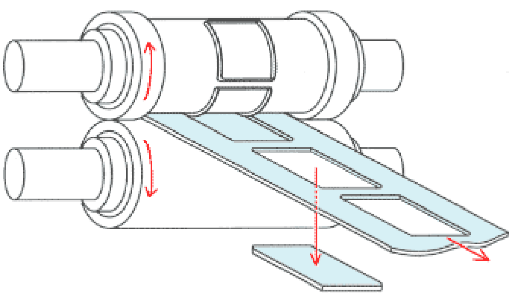


Figure 2. Rotary die cutter system.

When the tool moves along a straight line, the application is generally called “flying saws” (fig. 3). In this case the tool is typically mounted on a slide that moves either parallel to the product flow or at an angle across it. For flying saws, the motion of the carriage is synchronized to the motion of the product. In the cutting range, the saw moves at the same speed as the material web. Once the task is completed, the tool returns to its home position. This process is characterized by a null average speed of the tool axis in a work cycle.

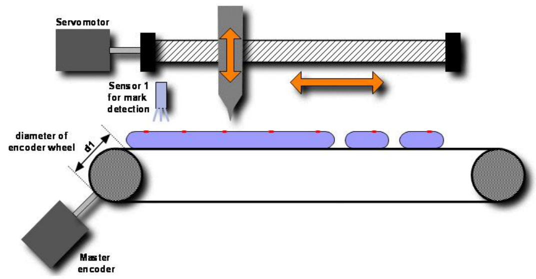


Figure 3. A flying saw web-cut application.

If the tool moves along a closed trajectory (generally a circular trajectory) the flying process is referred as Rotary Knife or Cross Cutter as for the cases of fig. 1 and fig. 2. One or more tools (knives) are mounted on a drum whose motion is synchronized in terms of velocity and position to the product’s one. For such application the average speed of the tools depends on the tool’s circumference and on the prescribed cut length.

For Rotary Knife operation it is possible to define a “base” or “design” length which represents the cut length got when the speed of the tool axis is kept constant during the operation (i.e. the speed of the knife match the speed of the line). In this case there will not be dynamic load acting on the system.

As previously mentioned, the flying cutting systems are widespread, in different configurations, in industrial production plants. Each flying cutting configuration has its own typical design and technological characteristics as the cutting depth, the maximum cutting speed, the temperature of the welding tools, etc. However, such kind of systems have some common characteristics, as the cutting length, the design speed. Hence a unified both kinematic and dynamic design process can be followed.

At present, a unified and systematic approach to this topic doesn’t exist. Each system is designed separately, hence there isn’t an useful improvements’ sharing between the different cutting technologies. Literature shows that the attention is mainly turned to the control of the cutting system; as an example, in [1] one of the first studies on the control, by means of a computer, of a “Flying shear” system for steel rods is presented. In [2] a new concept for the control of Flying shear systems, based on the optimization of the algorithm for the position control of the cutting tool in order to minimize the motor’s torque, is discussed. Some parameters typical of the flying cutting systems are introduced, but with reference only to rolling systems. In [3] only design and implementation issues of a control system for a steel tube cutting rotary saw are presented, without any effort to generalizing the problem.

In this paper a generalized approach to the flying machining is presented, in particular an unified set of design parameters will be identified and their influence on the system productivity will be discussed. The main aim of this work is to provide useful guideline for the synthesis of the tool motion profile, furthermore methodologies for the selection of the drive system will be proposed [5–11]. The procedure is described applying it on a cross sealing operation, typical of wrapping machine.

THE FLYING MACHINING ISSUE

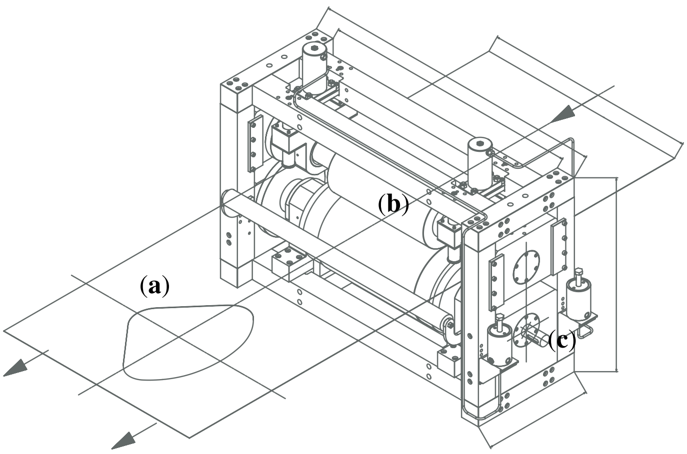


Figure 4. Rotary Knife configuration.

In figure 4 the sketch of a cross-cutter module is represented. It shows the main constituent parts of a general flying cutting system. It consists of two rolls with one cutter respectively. Between the rollers the material web to be cut flows (a); the rollers (b), coupled by means of two gears, rotate in opposite directions and carry the cutting blade. On the shaft (c), the motor-transmission unit, which gives motion to the system, is coupled.

Cutting plane

Whichever example of flying machining operation described in the previous paragraph can be modelled as a 2-D problem.

The cutting-edge, the hollow punch or the generic tool moves in the so called cutting-plane. In such a plane, two orthogonal directions can be identified: the first one, named x, is the direction of motion of the material(the tracking direction); the second one, named y, is the working direction (cut, printing, dinking...).

Generally, the motion of the tool in the cutting-plane is accomplished by means of a single motor, while the planar trajectory of the tool depends on the the transmission between the motor and the tool (for example a drum, a linkage mechanism etc.) Hence, the position of the tool strictly depends on the angular position of the main shaft. Figure 5 shows a part of a general trajectory of a tool's point; the point has a reciprocating motion, start and end being the same.

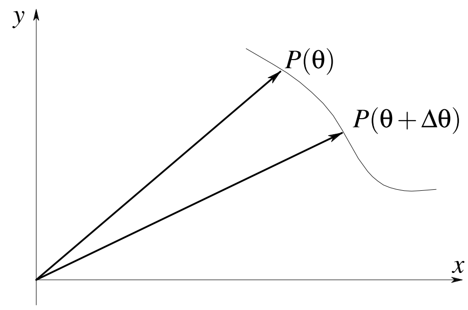


Figure 5. Tool's point motion.

Being the tool position described by vector P and its the displacement by s, it follows:

$$s = P(\theta+\Delta\theta) - P(\theta) \quad (1)$$

by derivation, the tool velocity can be obtained:

$$v = \frac{ds}{dt} = \frac{ds}{d\theta}\frac{d\theta}{dt} = \tau\omega_M(t) \quad (2)$$

where $\omega_M(t)$ is the motor's angular velocity. The vectorial term:

$$\tau_g = \frac{ds}{d\theta} \quad (3)$$

is generally named *generalized transmission ratio* or *geometric velocity*. Function $v(t)$ is periodic and its period is equal to the machining time. In general, being L_p the product's length and v_L its speed, the machining time can be written as:

$$t_c = \frac{L_P}{v_L} \quad (4)$$

hence $v(t) = v(t+t_c)$. Equation (2) projected along directions x

e y gives:

$$\begin{cases} v_x = \tau_{g_x}\omega(t) \\ v_y = \tau_{g_y}\omega(t) \end{cases} \tag{5}$$

Depending on the specific problem analyzed, velocity component v_y has specific characteristics; for instance in machine vision applications it could be null. Hence, v_y strictly depends on the specific application.

The velocity component v_x is generally characterized by a constant part, on which the tracking occurs, and a variable part to bring back the tool in the working position.

On a cycle, the velocity component v_x changes its sign; hence, being periodic, its mean value is null. Such a behaviour can be obtained following two different ways:

- a system characterized by a transmission ratio with a not null mean value $\tau_{g_{x\text{mean}}} \neq 0$ where the velocity sign change is due to the motor's law of motion.
- a system characterized by a transmission ratio with a null mean value: $\tau_{g_{x\text{mean}}} = 0$

Systems where $\tau_{g_{x\text{mean}}} \neq 0$

A typical example is a *cut on the fly* operation where a transmission with constant transmission ratio is used, as a the cutting edge mounted on the slide of a ball-screw [4].

In this case motor's speed mean value must be null; the system's design mainly consists in the design of the motor's law of motion, in order to get the tracking and the re-set of the tool.

Systems where $\tau_{g_{x\text{mean}}} = 0$

This is the most widespread and interesting case: by means of a variable transmission ratio it is possible to drive the motor at constant speed during the whole machining time.

Such a way allows to minimize the energy required by the system and to maximize the productivity.

In systems where $\tau_{g_{x\text{mean}}} = 0$ the transmission is designed for a specific machining length called *design length*: at constant motor's speed, the system works products with length equal to the design length.

For products with different lengths a motor driven at variable speed is needed.

Apart from the transmission ratio, the velocity component v_x depend also on angular speed $\omega(t)$, that can be separated in two components:

$$\omega_M(t) = \omega_{M\text{const}} + \omega_{M\text{var}} \tag{6}$$

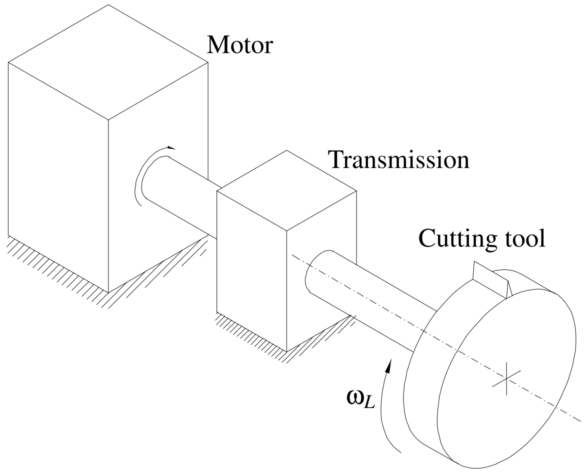


Figure 6. Principle scheme.

where $\omega_{M\text{const}}$ is the mean value while $\omega_{M\text{var}}$ is the variable speed term whose mean value, evaluated on the working time, is obviously null ($\omega_{M\text{var}_{\text{mean}}} = 0$).

By substituting equation (6) in the first of (5) we get:

$$v_x = \tau_{g_x} \cdot (\omega_{M\text{const}} + \omega_{M\text{var}}) \tag{7}$$

Transmission ratio τ_{g_x} can be seen as the product of two terms:

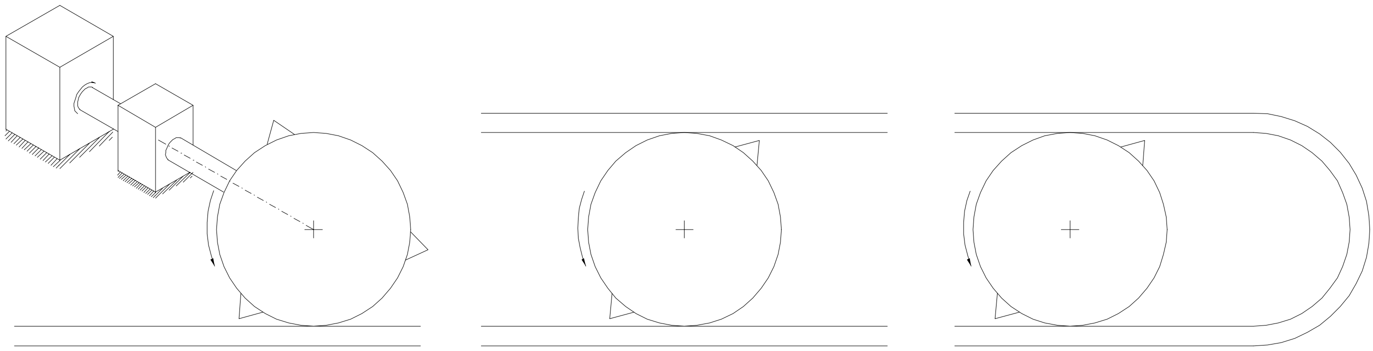
$$\tau_{g_x} = \tau \cdot \tau_v \tag{8}$$

where τ is a constant term (generally a speed reducer which optimizes the coupling between motor and load) while τ_v is a variable one (generally a mechanical system carrying the working tools). Hence, equation (7) can be written as:

$$v_x = \tau\tau_v \cdot (\omega_{M\text{const}} + \omega_{M\text{var}}) \tag{9}$$

In the systems where $\tau_{g_{x\text{mean}}} \neq 0$, since $\omega_{M\text{var}_{\text{mean}}} = 0$, the constant component must be null ($\omega_{M\text{cost}} = 0$). Hence in such systems a design length can't be defined: systems with constant transmission ratio can't work products at constant motor speed.

Therefore the systems where $\tau_{g_{x\text{mean}}} = 0$ fully describe the "flying machining" issue, including the problems concerning systems with constant transmission ratio. As far as the choice of τ term is concerned, it must satisfy the design criterion of matching the motor and load inertia. Figure 6 shows the principle scheme of a flying machining system typical of the flow pack packaging field.



(a) Many tools configuration (b) Configuration with two different lines. (c) Configuration operating in two phases.

Figure 7.

In such a system the working force (cutting force, pressing force...) acts in y direction and depends on the transmission ratio component τ_{g_y}, which must be appropriately chosen according to the specific application. Hence, as regards x direction, the load can be considered just inertial, while the working force just causes an easily definable increase of the motor power to be installed.

In order to increase the system's productivity, the principle scheme of figure 6 could be changed. The possible changes can be summarized as follows:

1. More than one tool housed on the *working head*: the machining frequency is higher than the motor's rotation one.
2. The working head operates on different lines: the lines must be accurately synchronized, but nothing changes as regards the behaviour of the working head.
3. The working tool operates on the products in two phases during the same cycle: with a very precise re-positioning, in order to avoid inaccuracy, the scraps can be reduced.

KINEMATIC ANALYSIS

Without loss of generality, in this section we develop the kinematic analysis of a rotary cross sealing and cutting station typical used in wrapping machine.

Figure 8(a) depicts the outline of a rotary drum holding three sealing yaws with integral cutters, equally spaced about the periphery of the drums. The rotating sealing jaws close the end of the product while the integral knife cuts the pack during the cross sealing.

As mentioned in the previous section, this system exhibits a transmission ratio with a null mean value: $\tau_{g_{x\text{mean}}} = 0$. The variable transmission ratio is due to the roller which houses the sealing and cutting tools: $\tau_{g_x} = R_t \sin\theta$, where R_t is the working drum's radius.

For this system the main kinematic design parameters are:

- n number of working tools
- θ_T sealing angle (synchronous angle)
- $\theta_0 = 2\pi/n$ angle between the tools
- $\theta_R = \theta_0 - \theta_T$ re-positioning angle
- R_t roller's radius
- L_p product length
- v_L line speed (the material speed)

It is worth to note that the above design parameters can be generalized and extended to a wide variety of "flying machining and processing" operations.

In figure 8(b) a sketch of the packaged product obtained after the cross sealing. is shown. In this work we will assume that the "product length" L_P is the sum of the worked part, which length is $2(L_T/2)$, and the packaged product length $L_{P,a}$; thus $L_P = L_T + L_{P,a}$

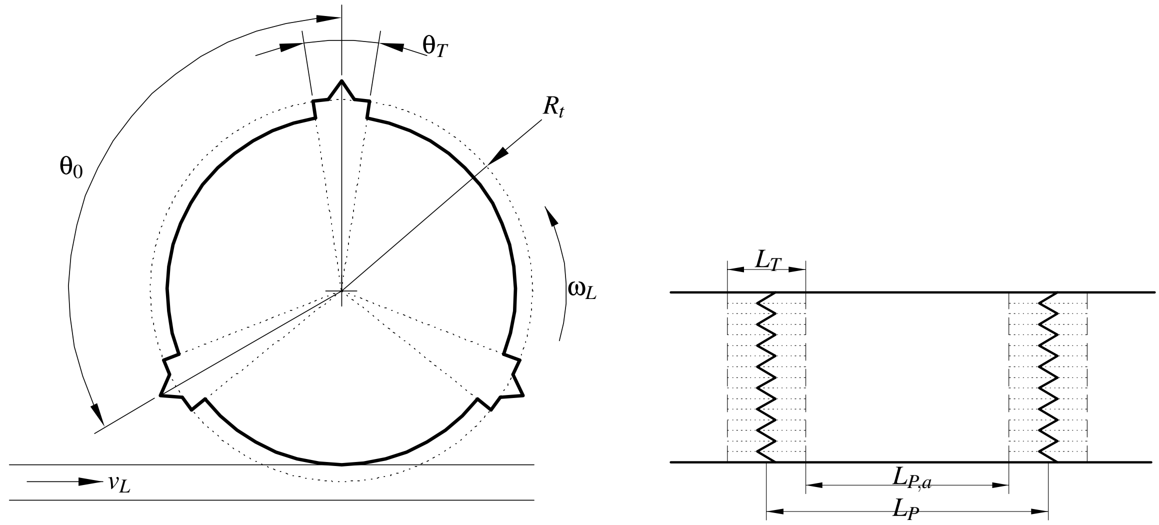
For the kinematic analysis is useful to re-write the above angular parameters as follows:

- $L_0 = \theta_0\, R_t$ design length
- $L_T = \theta_T\, R_t$ tool length (synchronous length)
- $L_R = L_0 - L_T$ repositioning lenght

With reference to figure 9 the method of operation of the rotary cross sealing under analysis can be easily explained. During a rotation θ_0 of the roller, two phases can be identified: the machining phase and the re-positioning phase.

During the machining phase, in the case of synchronous machining, the tool and the material web have to move at the same speed, in this way the material is neither compress nor torn. In other words in this phase the peripheral speed of the roller should match the line speed v_L.

During the re-positioning phase the product is not worked; if the product length corresponds to the design length L_0 the drum



(a) System under analysis (b) Product example

Figure 8.

moves synchronously with the material web and thus the peripheral speed of the roller remains constant. However when it is required to produce different formats it is necessary to modify the motion law of the drum.

In particular if the product length is smaller than the design length the roller first has to be accelerate and afterwards has to be braked to the line speed . If the product length is longer than L_0 the drum must be braked and than accelerated to the line speed.

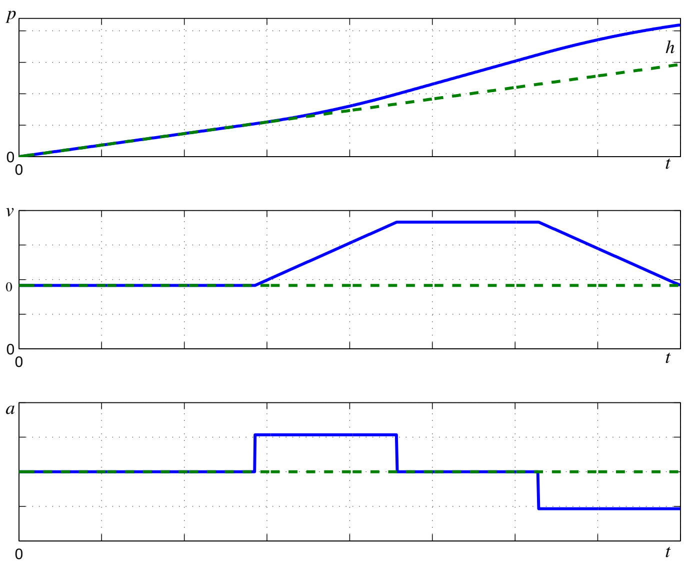


Figure 9. Motion sequence of the sealing process. Over-synchronous motion

In both cases the drive system has to be able to superimpose a suitable motion law to the synchronous motion.

The machining time t_C depends only on the product length and on the line speed, thus:

$$t_C = \frac{L_p}{v_L}$$

The synchronous time t_T , or in other words the machining time, can be obtained as:

$$t_T = \frac{L_T}{v_L}$$

while the re-positioning time t_R is simply:

$$t_R = t_C - t_T = \frac{L_P - L_T}{v_L}$$

During the re-positioning phase the drum has to rotate for θ_R, or the corresponding peripheral displacement L_R; this displacement can be viewed as the sum of two terms: the displacement due to the synchronous motion ($h_s = v_L\, t_R$) and the "dynamic" displacement h due to the superimposed motion law (see figure 9). Thus we can obtain the "dynamic" displacement h as:

$$h = L_R - h_s = (L_0 - L_T) - v_L \frac{L_P - L_T}{v_L} = L_0 - L_P$$

Motion law parameters

In the re-positioning phase it is necessary to choose a suitable motion profile that satisfies the boundary conditions defined above. More in detail the constraints are the motion time t_R, the displacement length h and the initial e final velocities that must be null.

Generally speaking, it is not possible to define a priori what is the best motion profile for this type of applications. In any case, one may select the shape of motion profile taking into account several kinematic parameters such as the maximum velocity, and the peak and root mean square (RMS) accelerations, that are of great importance for sizing the drive system.

It may be convenient to define some dimensionless coefficients, which do not depend on the displacement or on the duration of the motion, but only on the shape of the chosen motion law. As showed in [12, 13] these dimensionless parameters allow to quantify how the peak and RMS values of velocity and acceleration overcome the ideal reference values.

Among the others, we recall:

- Coefficient of velocity $C_v = \frac{v_{max}}{h/T}$
- Coefficient of acceleration $C_a = \frac{a_{max}}{h/T^2}$
- Coefficient of root mean square acceleration $C_{a,rms} = \frac{a_{rms}}{h/T^2}$

where h is the displacement, T is the motion time, v_{max}, a_{max} represent the peak value of the velocity and acceleration and a_{rms} is the RMS value of the acceleration.

Once the shape of the motion law is chosen, the peak and RMS value can be readily obtained.

The maximum variation of the linear velocity with respect to the line velocity v_L becomes:

$$\Delta v_{max} = C_v \frac{h}{t_R} = C_v \frac{L_0 - L_P}{L_P - L_T} v_L \quad (10)$$

and the corresponding angular velocity variation is:

$$\Delta \omega_{Lmax} = \frac{1}{R_t} C_v \frac{L_0 - L_P}{L_P - L_T} v_L \quad (11)$$

The maximum acceleration is:

$$a_{max} = C_a \frac{h}{t_R^2} = C_a \frac{L_0 - L_P}{(L_P - L_T)^2} v_L^2$$

and the corresponding angular acceleration is:

$$\dot{\omega}_{Lmax} = \frac{1}{R_t} C_a \frac{L_0 - L_P}{(L_P - L_T)^2} v_L^2 \quad (12)$$

The *RMS* acceleration value will depend on the coefficient $C_{a,rms}$ of the motion law and on the ratio between the re-positiong time and the cycle time. Being the acceleration null during the cutting phase, it easy to prove that the RMS acceleration on the total cycle time (t_c), can be expressed as:

$$\begin{aligned} a_{rms} &= \sqrt{\frac{1}{t_c} \int_0^{t_c} a^2\, dt} = \sqrt{\frac{1}{t_c} (\int_0^{t_r} a^2\, dt + \int_{t_r}^{t_c} a^2\, dt)} = \\ &= C_{a,rms} \frac{h}{t_R^2} \sqrt{\frac{t_R}{t_C}} = \\ &= C_{a,rms} \frac{L_0 - L_P}{(L_P - L_T)^2} \sqrt{\frac{L_P - L_T}{L_P}} v_L^2 \end{aligned}$$

and the relevant angular RMS acceleration is:

$$\dot{\omega}_{Lrms} = \frac{1}{R_t} C_{a,rms} \frac{L_0 - L_P}{(L_P - L_T)^2} \sqrt{\frac{L_P - L_T}{L_P}} v_L^2 \quad (13)$$

Variable length cut

One of the most important characteristic of a flying machining system is the flexibility, i.e. the capability to easily vary the product length. The larger is the difference between the product length and the design length L_0, the higher is the dynamic performance required; Hence the dynamic behavior of the system depends on the displacement h. According to the sign of h, three cases can be identified:

- $L_p = L_0$. The re-positioning occurs at constant speed; there isn't any inertial load.
- $h > 0$ (for $L_P < L_0$). In this case the re-positioning mean speed is greater than the line speed v_L. The drum first has to be accelerated and afterwards it has to be decelerated to match the line speed; its maximum acceleration and velocity depend on the chosen motion law, on the product length an obviously on v_L. The smallest product length is determined by both a high peak speed and acceleration.
- $h < 0$ (for $L_P > L_0$). The re-positioning takes place at mean speed lower than v_L. The drum first has to be decelerated and than has to be accelerated to the line speed. With respect to the previous case the maximum values of the required velocity and acceleration are lower; as a matter of fact an increase of the product length results in an increase of the re-positioning time. Obviously, when the absolute value of the velocity variation $|\Delta v_{max}|$ reaches the line speed v_L the total drum's velocity is null. In particular solving the equation (10) when $|\Delta v_{max}| = v_L$, we get:

$$L_{P,lim} = \frac{C_v L_0 - L_T}{C_v - 1}$$

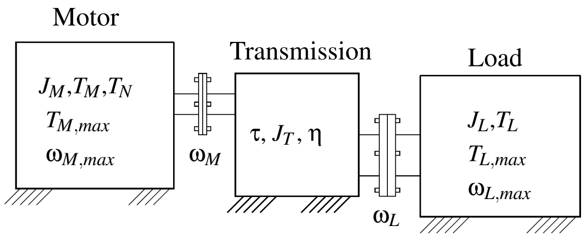


Figure 10. Motor/load principle scheme.

This condition depends only on the chosen motion profile. If the product length is greater than $L_{P,lim}$ the drum has to be brought to a standstill. Another possible strategy is to allow the roller to reverse its motion.

MODEL OF DYNAMIC SYSTEM

Figure 10 depicts motor and load connection and their respective moments of inertia J_M and J_L. Assuming power losses negligible and naming τ the constant transmission ratio connected to the motor, the dynamic system equation can be written:

$$\frac{T_M}{\tau} = T_L + \left(\frac{J_M}{\tau^2} + J_L\right)\dot{\omega}_L$$

As well known [6], the selection of the actuator means checking the following conditions:

- $T_{M,rms} \leq T_{M,N}$ rated motor's torque:
- $\omega_M \leq \omega_{M,max}$ maximum motor's speed
- $T_M(\omega_M) \leq T_{M,max}(\omega_M)$ maximum servo-motor's torque

The terms on the right side of inequalities are characteristic of each motor and the quantities $T_{M,rms}$, ω_M and T_M depend on the load and on the reducer transmission ratio τ. As known, the selection of the motor and the transmission should be performed simultaneously, because of the mutual dependence between the two components. Several methods are available to face the problem of the motor reducer sizing. In this article the procedure shown in [10] is used and it is applied on a generic flying cut system. By this sizing criterion the inequalities previously written can be expressed as functions of the transmission ratio τ. In particular, solving the inequality related to the rated motor torque leads to:

$$T_{M,N} \geq T_{M,rms} = \int_0^{t_a} \frac{T_M^2}{t_a}\mathrm{d}t = \tau^2 {T^*}^2_{L,rms} + J_M^2 \frac{\dot{\omega}^2_{L,rms}}{\tau^2} + 2J_M(T_L^*\dot{\omega}_L)_{mean}$$

From the inequality concerning the and maximum motor speed limit, it is possible to get, for each suitable motor, a range of acceptable transmission ratios defined as:

$$\tau_{max} \geq \tau \geq \max(\tau_{min};\tau_{M,lim})$$

where:

$$\begin{cases} \tau_{M,lim} = \dfrac{\omega_{L,max}}{\omega_{M,max}} \\ \tau_{min,max} = \dfrac{\sqrt{J_M}}{2T^*_{L,rms}}\left[\sqrt{\alpha-\beta+4\dot{\omega}_{L,rms}T^*_{L,rms}} \pm \sqrt{\alpha-\beta}\right] \end{cases} \quad (14)$$

are defined for each motor. Parameters α and β are the motor *accelerating factor* and the *load factor* respectively:

$$\alpha = \frac{T^2_{M,N}}{J_M}; \qquad \beta = 2\left[\dot{\omega}_{L,rms}T^*_{L,rms} + (\dot{\omega}_L T^*_L)_{mean}\right] \quad (15)$$

They respectively define the performance of the motors and those required by the task [10]. Using α and β, inequalities concerning rated motor torque becomes:

$$\alpha \geq \beta + \left[T^*_{L,rms}\left(\frac{\tau}{\sqrt{J_M}}\right) - \dot{\omega}_{L,rms}\left(\frac{\sqrt{J_M}}{\tau}\right)\right]^2. \quad (16)$$

Since the term in brackets is always positive, or null, the motor accelerating factor α must be sufficiently greater than the load factor β, for the inequality on rated motor torque to be verified.

The generalized load torque T^*_L and its root mean square value $T^*_{L,rms}$ are defined respectively as:

$$T^*_L = T_L + J_L\dot{\omega}_L; \qquad T^*_{L,rms} = \frac{1}{t_c}\int_0^{t_c} {T^*_L}^2 \mathrm{d}t$$

Assuming the load torque T_L negligible with respect to the inertial loads (load is purely inertial), the terms related to the load become: $T^*_L = J_L\dot{\omega}_L$ and $T^*_{L,rms} = J_L\dot{\omega}_{L,rms}$. In this case the load factor expression (15) becomes more simple:

$$\beta = 4J_L\dot{\omega}^2_{L,rms}. \quad (17)$$

For each motor, the values of τ_{min} and τ_{max} can be obtained by:

$$\tau_{min}, \tau_{max} = \frac{\sqrt{J_M}}{2T^*_{L,rms}}\left[\sqrt{\alpha} \pm \sqrt{\alpha-\beta}\right]. \quad (18)$$

Finally, it is possible define the value $\tau_{opt} = \sqrt{J_m/J_r}$ called *optimal transmission ratio* and introduced by [5]. Using the optimal

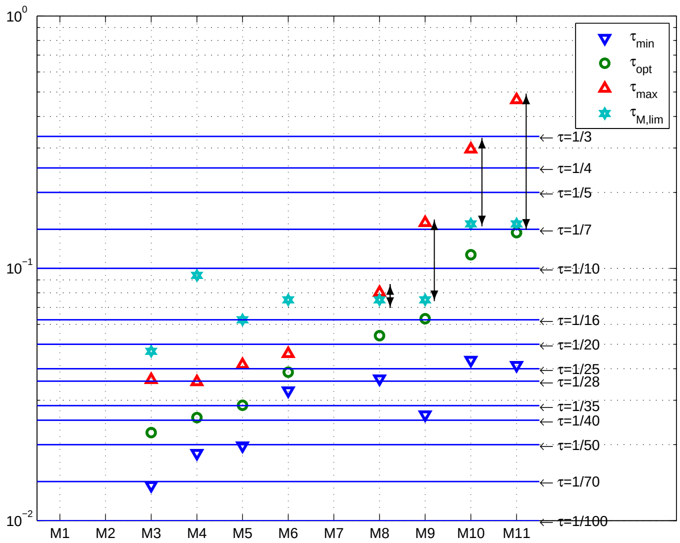


Figure 11. Overview of available motor-reducer couplings.

transmission ratio, the inertia of the load reflected to the motor shaft equals the motor inertia; it is said that "motor and load are balanced" and the power supplied by the motor to accelerate the system is equally distributed on the two transmission shafts.

The main selection steps are:

STEP 1: Creation of a database containing all the commercially available motors and reducers useful for the application. For each motor the accelerating factor (α_i) must be calculated.

STEP 2: Calculation of the load factor β.

STEP 3: Preliminary choice of useful motors: all the motors for which $\alpha < \beta$ can be immediately rejected because they haven't enough torque, while the others are admitted to the next selection phase.

STEP 4: Identification of the ranges of suitable transmission ratios for each motor preliminarily selected in step 3. In order to easily implement the selection procedure, a suitable graphical representation can be used (for example Fig. 11) displaying, for each motor, the value of the transmission ratios τ_{max}, τ_{min}, τ_{opt} and $\tau_{M,lim}$. The graph is generally drawn using a logarithmic scale for the y-axis, so τ_{opt} is always the midpoint of the suitable transmission ratios range. A motor is acceptable if there is at least a transmission ratio τ for which equation (14) is verified. These motors are highlighted by a vertical line on the graph.

STEP 5: Identification of useful commercial speed reducers: the speed reducers available are represented by horizontal lines. If one of them intersects the vertical line of a motor, this means that the motor can supply the required torque if that specific speed reducer is selected. These motors and reducers are admitted to the final selection phase.

STEP 6: Optimization of the selected alternatives: the selection can be completed using criteria as economy, overall dimensions, space availability or any other criteria considered important depending on the specific needs.

STEP 7: Checks. For each set of motor and reducer unit, their moment of inertia J_M and J_T and the transmission mechanical efficiency η are fully known. Now it's possible to check: the maximum torque supplied by the servo-motor for each angular velocity achieved, the effect of the transmission's mechanical efficiency (η) and its moment of inertia (J_T) on the root mean square torque and the resistance of the transmission as supplied by the manufacturer.

ANALYSIS OF THE DESIGN PARAMETERS

Joining the equations (13) and (17) it is possible to show the load factor, for a generic flying cut system, in terms of the machine design parameters, by this equation:

$$\beta = 4J_L \left[\left(\frac{C_{a,rms}\, v_L^2}{R_t} \right)^2 \frac{(L_0 - L_p)^2}{L_p (L_p - L_T)^3} \right] \tag{19}$$

Substituting the expression (19) in the equation (18) and calculating the difference between maximum and minimum transmission ratio, it can be obtained:

$$\Delta\tau = \frac{\sqrt{J_M}}{J_L \dot{\omega}_{L,rms}} \sqrt{\alpha - \beta} = 2\sqrt{\frac{J_M}{J_L}(k-1)} \tag{20}$$

where $k = \alpha/\beta \geq 1$.

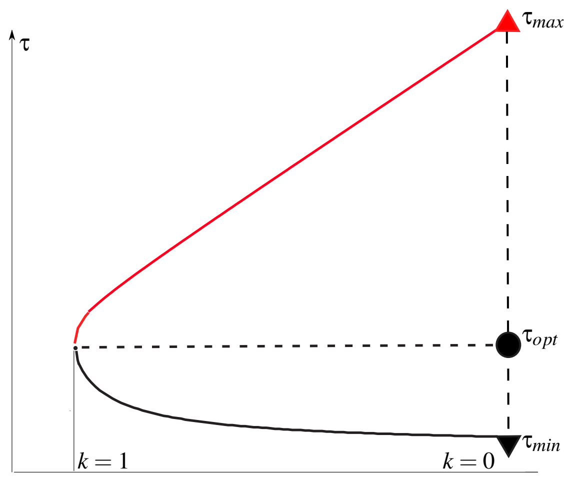


Figure 12. $\Delta\tau$ range.

Figure 12 shows a diagram of the behaviour of the maximum transmission ratio (red line), the minimum transmission ratio (black line) and the optimal one (dashed line) varying the load factor β, for a generic motor coupled with a flying cut system. If β increases towards α, k tends to become equal to 1 and both the maximum and minimum transmission ratios become equal to the optimal one. When the motor is already chosen, the optimal transmission ratio does not depend on load factor: this result is true if the load reflected inertia J_L doesn't depend on the design length L_0.

It is possible to increase the load factor β to α factor by modifying the design parameters or the working conditions of the cutting system as described by the equation (19). For example increasing the velocity v_L, i.e. increasing the system productivity, β increases being proportional to the fourth power of the line speed.

The equation (19) can be also used to optimize the design of the cutting system when the working conditions are defined: for example, maximum difference between L_0 and L_P or variation range of line speed.

Figure 12 shows that the optimal transmission ratio is the better choice in terms of working flexibility of the cutting system. Equations (19) and (20) are very important: they can be used to evaluate β_{min} and β_{max} and to verify if the motor and transmission set is able to guarantee the required performance.

CONCLUSIONS

The paper suggests a comprehensive view of the issues concerning the flying machining systems. In particular an unified design procedure is proposed, suitable for whichever flying machining operation.

In the first part of the paper, a classification of the different kinds of machines and operations has been outlined and then a main configuration has been identified from which all the possible systems can be derived.

First of all, the motion of the working tool has been modelled as a plane motion and it has been projected along two perpendicular directions: working direction and material direction. As outlined in the relevant section, the attention has been pointed out on the motion along x axis; for this motion a kinematic analysis has been carried-on and some motion law parameters have been derived as a function of the design parameters (equations (10), (11),(12)).

Afterwards, a dynamic analysis has been carried on too, leading to the definition of the expressions (19) and (20) for the load factor and for the difference between the maximum and minimum transmission ratio. Also these equations depend only on the design parameters and, along with the previous ones, allow to properly choose the law of motion and the motor/transmission set.

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