EICASLAB DEMO

*The professional software suite*
*for automatic control design and forecasting*

# EICASLAB DEMO
# Test Cases

### *Technical Note*

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# INTRODUCTION

With the aim of showing the potentiality of the EICASLAB software suite, the control design of a rotating table has been selected as demo case. It is a common problem in industrial automation, which does not present difficulty from theoretical point of view, but in practice it requires to make a trade off between performance and mechanical component cost. The demo case shows how the EICASLAB software suite can help you to make the selection of plant components and control algorithm, which are the most appropriate to your needs.
An actual case has been considered and all the plant component data (electric motor and mechanical gear) are derived from commercial data sheets.
The control has been designed on the basis of a "simplified model" and applied to a simulated "fine model" of the rotating table. This last model includes frictions, backlash, elasticity and hysteresis affecting the torque transmission from the motor to the table. The aim is to show how the control can be designed and its performance assessed in virtual environment on the basis of the plant design data only, before the component selection and the plant final design have been frozen.

The control software code is generated by EICASLAB and it should be used in the application without requiring any parameter set up in field. Specifically this is true when the EICAS model based control design approach is followed, as it has been assessed by the EICAS large experience in different application fields. Indeed, the EICAS model based control adopts robust algorithms, which offer guaranteed performance in presence of plant uncertainty.

Different control architectures are considered (varying the measured rotation angle) and different control algorithms are compared (PID, classical model based control and EICAS model based control). Moreover, the possibility to program an user control by adopting any one of the considered control architectures is given.

# 1 THE DEMO TEST CASE

## 1.1 The plant

The single axis plant represented in Fig. 1 is considered.

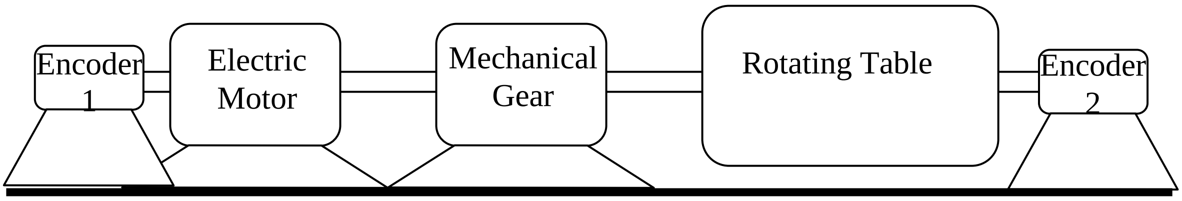


**Figure 1 The single axis plant**

A load (called “rotating table”) is rotated by means of an electric motor through a mechanical gear. The motor rotation angle is measured by the encoder 1 and the load rotation angle is measured by the encoder 2.
The main technical data deduced from each component data sheet are summarized in the following paragraphs.

## 1.2 User required performance

The plant mechanical structure has been designed with the aim of getting the following performance by means of a suitable automatic control:

- angular rate range -1 ÷1 rad/s
- angular acceleration range -1 ÷1 rad/$s^2$
- rotation reference signal frequency bandwidth up to 4 Hz
- perturbing torque applied to the rotating table
  - low frequency component
    - amplitude range 100÷100 N.m
  - random component
    - frequency power spectrum range 0.5 Hz
    - r.m.s. value 8 N.m
- the rotating table must track the rotation reference signal within an r.m.s. error value of 1 mrad.

## 1.3 Electric and mechanical component data

### 1.3.1 Electric Motor

A brushless DC motor commanded in current has been considered. The motor main technical data are given in Table 1.

| motor specifications | value |
|---|---|
| rated power output | 220 W |
| rated torque | 0.70 N.m |
| rated current | 12 A |
| maximum pulse current | 50 A |
| maximum angular rate | 314 rad.$s^{-1}$ |
| torque constant | 0.059 N.m.$A^{-1}$ |
| rotor moment of inertia | 0.00015 kg.$m^2$ |
| viscous damping constant | 0.0003 N.m.$rad^{-1}$.s |
| static friction torque | 0.03 N.m |

**Table 1. Electric motor rated data**

### 1.3.2 Mechanical Gear

A mechanical gear of harmonic type has been considered. The main technical data are summarized in the following Table 2.

| **Mechanical gear specifications** | **value** |
|---|---|
| ratio | 200 |
| input speed range | 0 – 360 rad.s$^{-1}$ |
| admissible maximum output torque in all the above input speed range | 114 N.m |
| moment of inertia | 0.000194 kg.m$^2$ |
| backlash | ± 0.0025 rad |
| torsional stiffness | 81870 N.m.rad$^{-1}$ |
| no load running torque.<br>input speed 0 rad.s$^{-1}$<br>input speed 360 rad.s$^{-1}$ | <br>0.05 N.m<br>0.20 N.m |

**Table 2. Mechanical gear rated data**

The mechanical gear characteristic representing torque versus torsional deformation is shown in Fig. 2. It includes the effects of the gear elasticity, backlash and hysteresis.

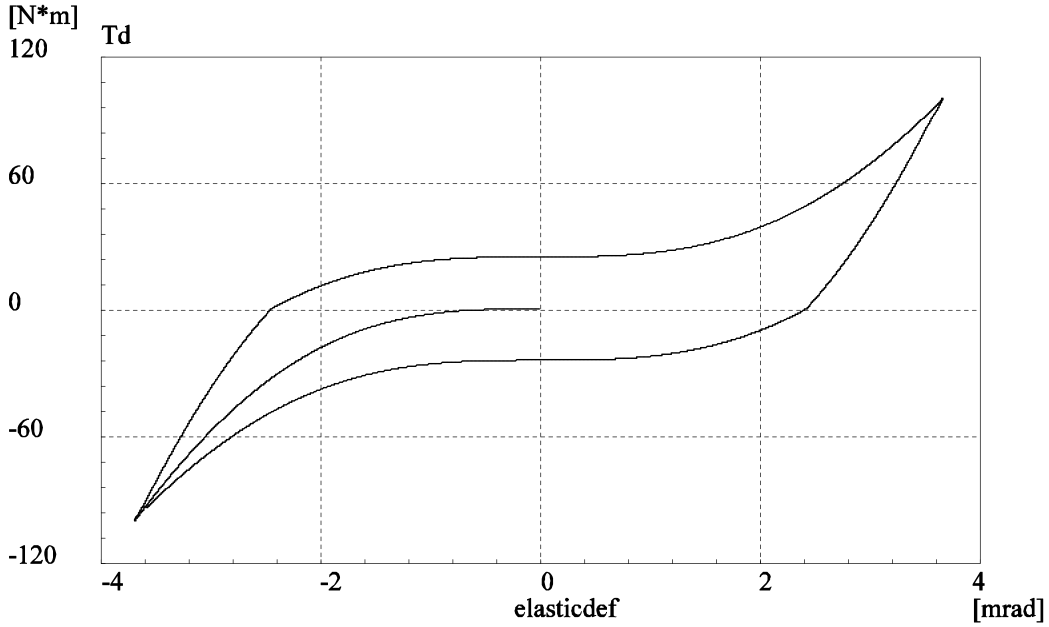


**Figure 2 Mechanical gear characteristic: torque versus torsional deformation**

### 1.3.3 Rotating Table

The rotating table is a rigid body with moment of inertia $5 \ \text{kg.m}^2$ .
It is subjected to the perturbing torque specified in §1.2.

### 1.3.4 Encoders

Resolution:

- encoder 1, measuring the electric motor angular rotation, quantization level 2 mrad
- encoder 2, measuring the load angular rotation, quantization level 0.1 mrad

## *1.4 Mathematical models*

Two mathematical models are introduced to describe the dynamic performance of the single axis rotating plant above described.
The first one, called "simplified model", is related to the "concept" according to which the plant has been designed. That is, it is a plant model simplified by neglecting all those physical phenomena, such as frictions, backlash, axis flexibility, etc., which are neither necessary nor useful to get the user required performance, but they are just tolerated in the plant design as a performance-cost trade off, evaluating that they do not prevent the attainment of the plant required performance.

The second one, called "fine model", gives a plant description finer than the one given by the "simplified model" with the aim to point out the main limits of the "simplified model" from the point of view of the closed loop control design.

### 1.4.1 Simplified model

The system concept, according to which the plant has been designed, is very simple. The goal is to rotate a rigid body (the "rotating table") and the required result is obtained by applying to such a rigid body the torque generated by an electric motor through a suitable mechanical gear, which - according to the purpose of the plant design - should transmit the torque from the motor to the rotating table in an instantaneous and rigid way. Then, the plant is considered described with a sufficient accuracy by a "simplified model" consisting in an ideal rigid body rotating around a fixed axis subjected to a torque proportional to the motor current commanded by the automatic control.

The above "simplified model" corresponding to the plant design concept is the plant mathematical model typically used in the closed loop control design and it has been used in all the following demo cases, where the plant control has been designed according to a model based approach.
For the control design purpose the "simplified model" has been expressed by the following discrete state equation set (1) graphically illustrated by the scheme of Fig. 3 .

$$\begin{vmatrix} x_1(i+1) \\ x_2(i+1) \end{vmatrix} = \begin{vmatrix} 1 & 0 \\ 1 & 1 \end{vmatrix} \begin{vmatrix} x_1(i) \\ x_2(i) \end{vmatrix} + \begin{vmatrix} a \\ 0 \end{vmatrix} (u(i) + d(i)) \tag{1}$$

$$y(i) = \begin{vmatrix} 0 & 1 \end{vmatrix} \begin{vmatrix} x_1(i) \\ x_2(i) \end{vmatrix}$$

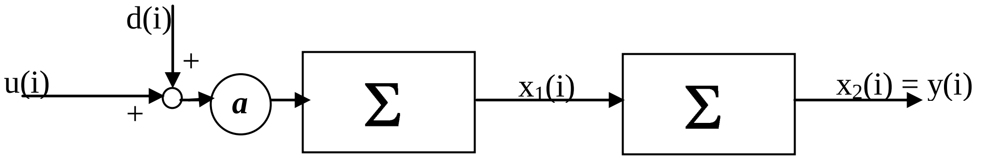


**Figure 3 The plant simplified discrete dynamic model**

The following notations have been used:

- *i* sampled time
- *u(i)* system input command corresponding to the electric motor current
- *d(i)* disturbance torque applied to the plant expressed in terms of equivalent motor current
- $x_1$, $x_2$ state variables
- *y(i)* system output corresponding to the load rotation angle
- *a* model parameter the value of which is derived from the plant technical data according to the following relation:

  $a = (r\, k_T\, T_s^2)/(r^2(J_m+J_g)+J_l)$

  where:
  - $r$ gear ratio
  - $k_T$ torque constant of the electric motor
  - $T_s$ time sampling step
  - $J_m$, $J_g$, $J_l$ moment of inertia, respectively, of motor, mechanical gear, rotating table.

### 1.4.2 Fine model

The “fine model” has been selected in order to point out the main limits of the “simplified model” from the point of view of the closed loop control design.
The above limits are mainly expressed by the lack of rigidity of the torque transmission axis from the electric motor to the rotating table. The following three causes, all related to the mechanical gear, have been identified:

1. finite torsional stiffness K = 81870 N.m/rad
2. backlash B = ± 0.0025 rad
3. hysteresis, illustrated together with the backlash effect by Fig. 2.

All the other delay causes, which exist in the chain going from the control input to the controlled output (such as the delay between the current command and the related electric torque effect) are all considered as negligible, because they are acting in a frequency domain which is outside, higher than the "a priori" assumed maximum frequency range of the feedback control to be designed.

Then, the "fine model" which has been used to simulate the plant has been conceived as two rigid bodies $J_1$ and $J_2$ coupled by a torque transmission axis ***Ta***, as in the following Fig. 4.

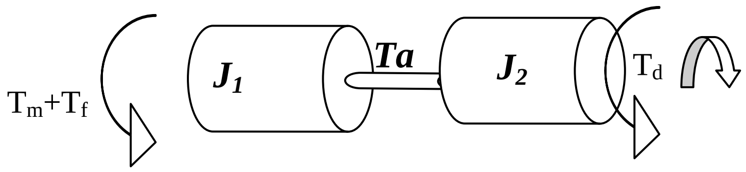


**Figure 4 The plant "fine model"**

The rigid body $J_1$ has a moment of inertia resulting by the sum of the motor and gear inertias. It is subjected to a torque equivalent to the sum of the motor electric torque $T_m$ and of the motor and gear static friction and viscous torque $T_f$.
The rigid body $J_2$ has the rotating table moment of inertia and it is subjected to the external disturbance torque $T_d$ acting on the load.
The torque transmission axis ***Ta*** has the characteristic illustrated in Fig. 2, which has been simulated by the backlash and hysteresis non-linear dynamic model included in the EICASLAB library.
As an effect of the above torque transmission axis flexibility, the plant presents a proper vibration frequency, which may be computed by neglecting backlash, hysteresis and frictions. In such approximated conditions and in the assumption that no closed loop control is applied to the plant, the proper vibration frequency is given by the relation:

$$f_o = (1/2\pi)\sqrt{K / J_{eq}} = 24\ Hz \qquad (2)$$

where

$$J_{eq} = (r^2(J_m+J_g)J_l)/ (r^2(J_m+J_g)+J_l)$$

In the assumption that the electric motor angular position is closed loop controlled, the relation (2) is no more true, but it results:

$$f_o = (1/2\pi)\sqrt{K / J_l} = 20\ Hz \qquad (3)$$

In working conditions, when the backlash effect cannot be neglected, the plant may present proper vibrations at lower frequency values. The effect is caused by a reduction of the transmission axis rigidity, which drops to zero together with the value of the torque transmitted by the mechanical gear.

# 2 CONTROL REQUIRED PERFORMANCE

## *2.1 Operating range*

The control system is required to operate within the following range, which derives from the user performance requirements (§1.2):

- electric motor rated torque 0.7 N.m
- maximum pulse torque 2.9 N.m
- load angular rate - 1 ÷ 1 rad/s
- load angular acceleration - 1 ÷ 1 rad/$s^2$

## *2.2 Operating modes*

### 2.2.1 Point to point mode

The control system is required to rotate the load from the current angular position to the commanded angular position.
The rotation must be carried out in the minimum time within the constraints of the above stated ranges of current, acceleration and rate (§ 2.1). Once the commanded position has been reached, it must be maintained until a new rotation command is received. In such a condition the positioning of 1 mrad is required under the perturbing torque already specified at § 1.2, here recalled:

- perturbing torque applied to the rotating table
  - low frequency component
    - amplitude range 100 ÷100 N.m
  - random component
    - frequency power spectrum range 0.5 Hz
    - r.m.s. value 8 N.m
- static pointing accuracy 1 mrad

### 2.2.2 Tracking mode

The control system has to track given reference trajectories with the following requirements:

- admissible reference trajectory set
  - angular rate and acceleration within the ranges stated at § 2.1
  - frequency bandwidth within 4 Hz
- perturbing torque acting on the load
  - low frequency component
    - amplitude range -100 ÷100 N.m
  - random component
    - power spectrum frequency bandwidth 0.5 Hz
    - r.m.s. value 8 N.m
- mean squared tracking error 1 mrad

# 3 TEST CASE AND CONTROL DESIGN APPROACHES

## *3.1 Test Case*

A specific test case has been defined in order to assess the control system performance. It includes two different trials of 30 s each one.
The first trial is working in point to point operating mode. The commanded sequence consisting in 10 rotations is illustrated in Fig. 5 .
The second trial is working in tracking operating mode. The trajectory to be tracked is stated by a sequence of values sampled at the frequency of 10 Hz. The above data sequence is interpolated on real time in order to get the control reference signal at the control sampling rate. The commanded sample sequence and the interpolated reference signal are illustrated in Fig. 6 and 7.
In both cases the rotating table is assumed to be subjected to the same perturbing torque illustrated in Fig. 8.

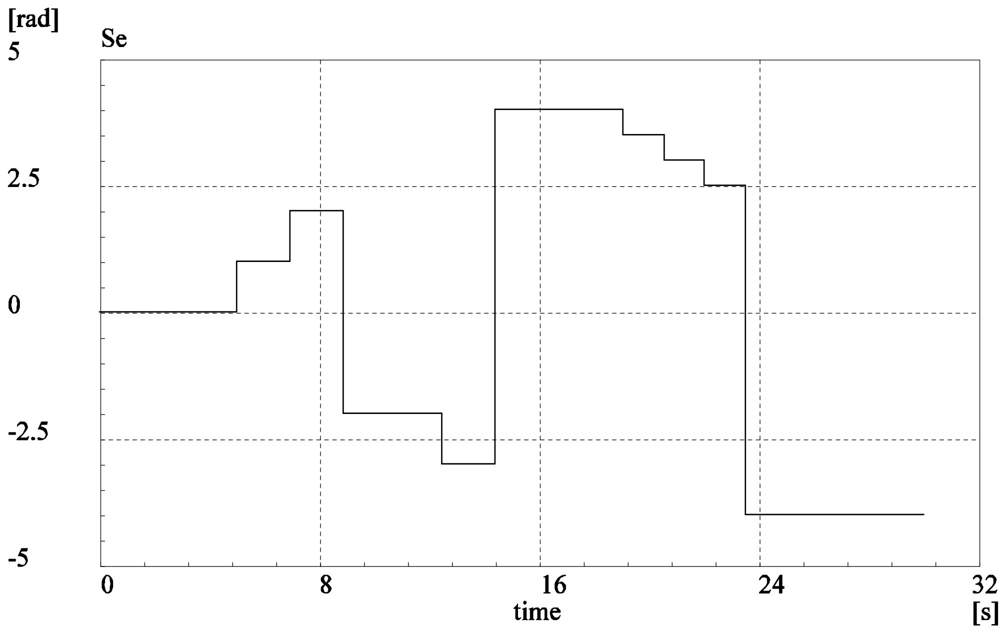


**Figure 5 The commanded sequence Se in point to point operating mode as function of time**

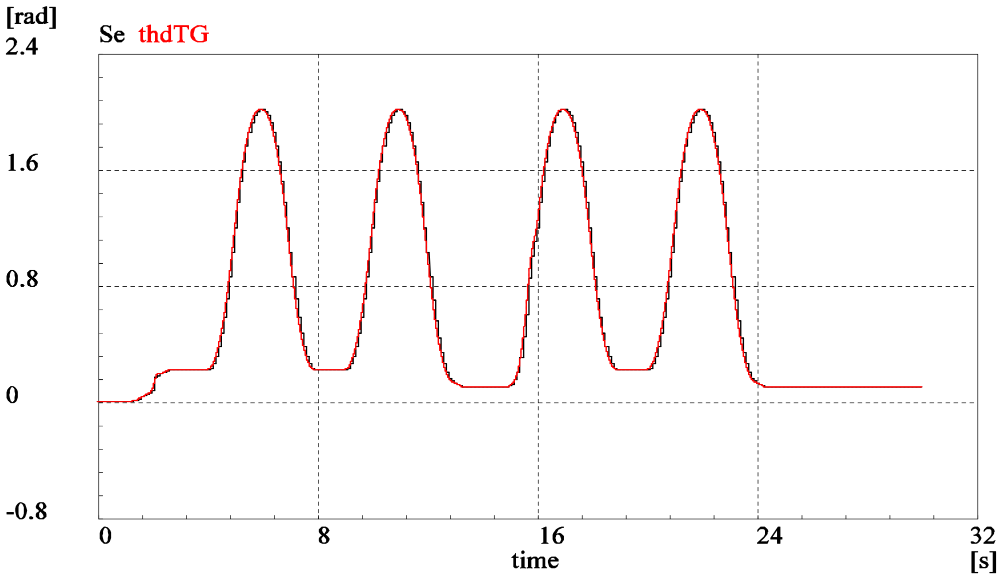


**Figure 6 The commanded sequence Se and the interpolated reference signal thdTG in tracking operating mode**

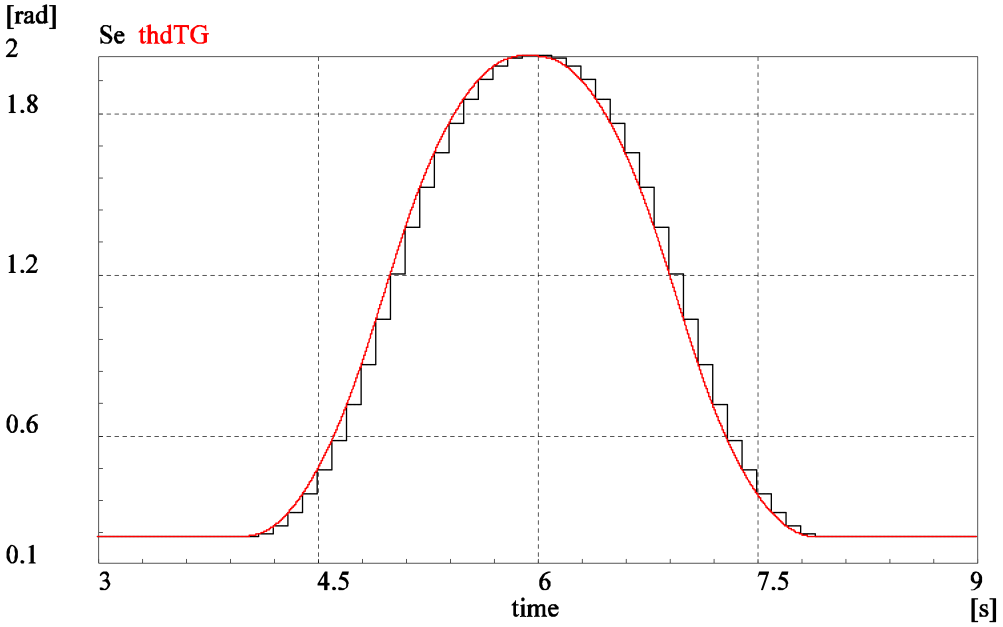


**Figure 7 Zoom of the commanded sequence in tracking operating mode, where:**
**Se = commanded sequence of points**
**thdTG = interpolated reference signal**

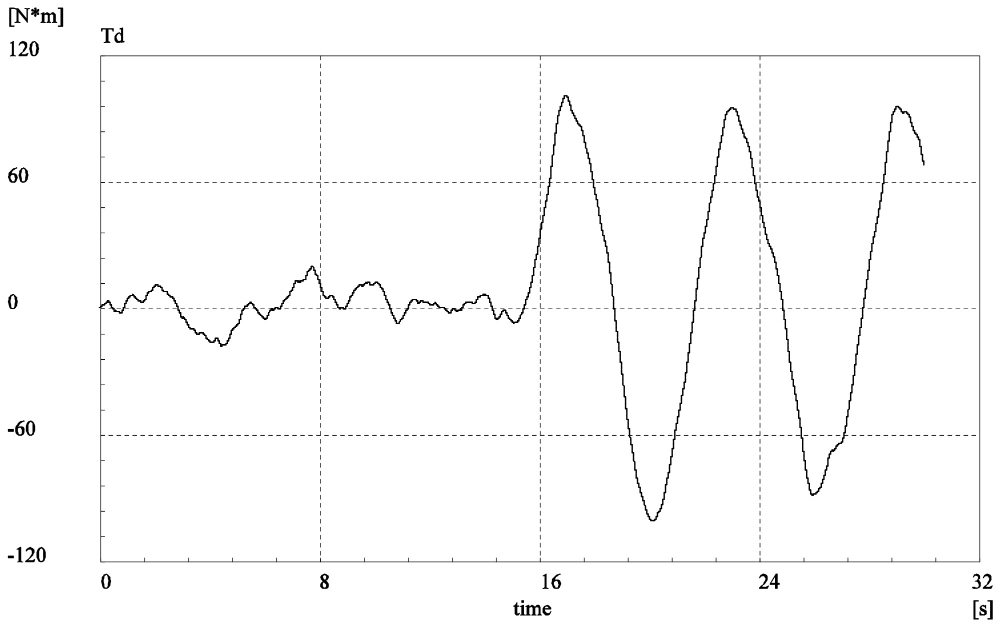


**Figure 8 Perturbing torque acting on the rotating table as a function of time**

### *3.2 Control Architectures*

The aim of the closed loop control system is to vary the angular position of the rotating table according to the user requirements by acting in a closed loop way on the electric motor current value.

Three different control architectures have been considered, as follows:

1. the control loop is closed by the measurement data of the encoder 1, put on the electric motor axis;
2. the control loop is closed by the measurement data of the encoder 2, put on the rotating table axis;
3. two control loops are closed at different hierarchical level: the low level control loop is closed by the encoders 1 and the superior one by the encoder 2 .

Let us recall that in all cases the final aim is to control the angular position of the rotating table, which is measured by encoder 2, even when, as in the above architecture 1, only the rotation of the electric motor is measured. Then, independently from the architecture adopted, the control performance always is evaluated by considering the error between the tracking reference signal and the actual angular position of the rotating table.

The **architecture 1** is the most commonly used one. Indeed, it offers the advantages of a low cost encoder and of the availability of a large control frequency band, which allows to get a very small tracking error between the reference signal and the measurement data obtained by

the encoder 1. Its drawback is the accuracy loss as an effect of the mechanical gear backlash and flexibility.
The **architecture 2** is potentially more accurate than the first one, because there is measured and closed loop controlled what is the true aim of the control design, that is, the angular position of the rotating table. Its drawback is the limit in the available control frequency bandwidth, which shall be significantly lower than the frequency value of the plant natural vibrations caused by the mechanical gear flexibility and backlash.
The **architecture 3** can offer both the large control frequency band (at low accuracy) of the architecture 1 and the high accuracy (within a low frequency band) of the architecture 2. Its drawback is the cost of two encoders.

## *3.3 Compared Control Design approaches*

### 3.3.1 PID Control

The commanded motor current is computed on the basis of the error between the reference signal and the measured angular rotation as a linear combination of the error value (proportional action), of its integral value (integrative action) and its derivate value (derivative action). In the discrete time control the integral value is substituted with the sum of all the past error values, the derivate value by the finite difference between the current error value and the previous one.
In order to avoid the noise induced by the measurement quantization level in the derivative action computation a suitable low pass band filter has been introduced.
An optimal design of the PID control has been performed according to the procedure described in § 3.3.4. The three PID coefficient values has been obtained by the minimization of the error cost function performed by means of the numerical optimisation procedure available in EICASLAB.
Two PID optimal controls have been computed respectively for the architectures 1 and 2.
The architecture 3 has not been implemented by means of PID control.

### 3.3.2 Classic Model Based Control

The “simplified model” described in § 1.4.1 has been adopted and the classic control system architecture, illustrated in Fig. 9, consisting in state observer and state controller has been implemented.
Observer and controller coefficients have been determined by the numerical minimization of the same control cost functional (defined in § 3.3.4) used in the optimisation of the PID control parameters.

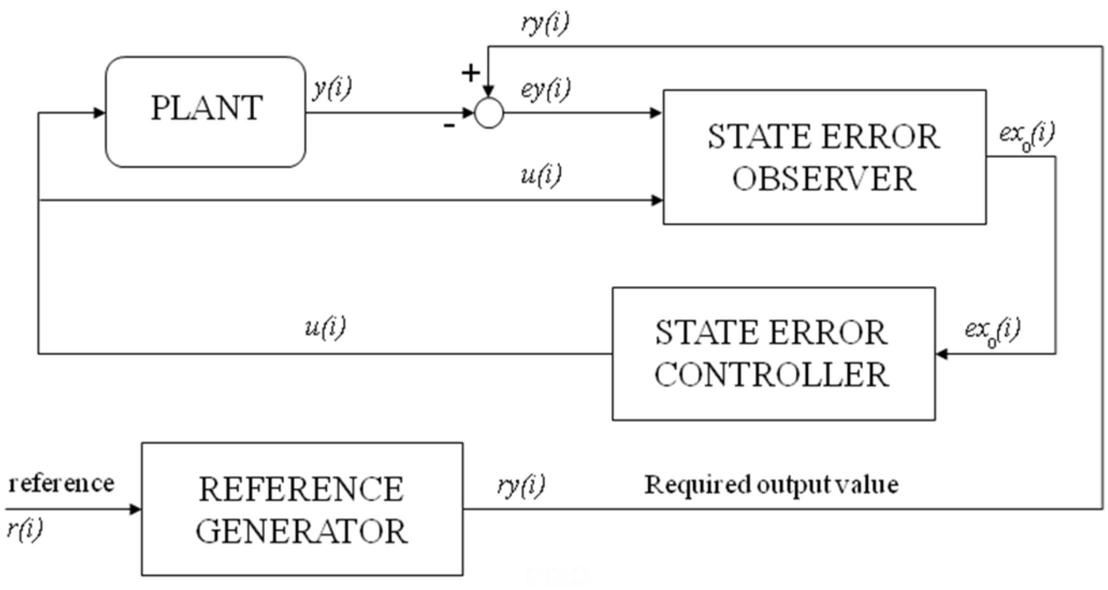


**Figure 9 The functional scheme of the classic model based control**

It is worth to point out the difference between the classic optimal control and the numerical control optimisation that has been here applied in the design of the model based control.
In the classic optimal control theory the assumption is made that the actual plant is strictly working according to the mathematical model on which the control design is based. Then, the optimal control design is the search of a theoretical solution of the optimisation problem.
On the contrary, here, the used model is assumed to be a "simplified model" of the actual plant and a more accurate "fine model" is also given to point out its inaccuracy. The cost functional is obtained by a trial simulation of the control designed on the basis of the "simplified model" applied to the "fine model". The optimal solution is obtained by the numerical minimisation of the cost function in a simulated trial.

Both the control architectures 1 and 2 have been implemented following the above described procedure.
The control architecture 3 has not been implemented by means of the classic model based control design methodology.

### 3.3.3 EICAS Model Based Control

The EICAS model based control design requires the availability of two plant models, respectively, the "simplified model" and the "fine model". The theoretical foundation and the results of on field test cases of the EICAS model based control design are reported in the Proceedings of "*Acoduasis Workshop: One step further in automatic control design*" held in October 2005 in Torino (Italy) [1-11], available at the following web sites:

- EICAS web site, news: "ACODUASIS WORKSHOP & EICASLAB COURSE "
  http://www.eicas.it/index_file/frame%20uk/news/news.html
- ACODUASIS web site:
  http://ids.fzi.de/acoduasis/

- EICASLAB web site: http://www.eicaslab.com/index_file/1.HOME/gotop.php?mainp=../5.TEST-CASES/TEST-CASES-M2.htm

The control system is designed in order that it is able to offer *guaranteed performance,* when applied to any dynamic system belonging to a stated *system set*, which includes both the “simplified model” and the “fine model”. Starting from the “simplified model” and the “fine model”, the designer derives the *system set* to which he deems the actual plant to belong with a suitable *factor of safety*. At this point the designer cannot impose also a required *guaranteed performance* for all the *system set,* but the best *guaranteed performance* level will be an output of the control system design.
The *guaranteed performance,* obtained by the EICAS model based control design, cannot be improved by applying any control design optimisation (f.i. the one applied in the demo test case to the PID and classic model based control design). A conflict, indeed, exists between an optimisation, which neglects any uncertainty in the plant knowledge, and the introduction of a *factor of safety* in the control design in order to take into account the fact that the plant is approximated by the “simplified model” and it is neither strictly corresponding to the “fine model”. The reduction of the cost function, which could be obtained by the numerical minimisation performed in a specific trial, is always paid by a reduction of the adopted *factor of safety* .
On the contrary, the *guaranteed performance* can be improved so much as required by reducing the “distance” between “simplified model” and “fine model”, what, in practice, means to act on the plant mechanical design in order to remove (or to attenuate) those physical phenomena which make the actual plant (and then the “fine model”) different from the plant ideal concept (that is the “simplified model”).

The functional architecture of the EICAS model based control system is illustrated in Fig. 10.
The EICAS functional architecture differs from the classic model based control architecture in the state observer and in the state controller. To the state observer, denoted as “state error and disturbance observer”, is attributed also the task to estimate the plant state error $ex_o(i)$ and disturbance $d_o(i)$. The state controller, denoted as “state error control and disturbance compensation”, together with the classic feedback action proportional to the plant state error performs the predicted disturbance compensation.
Another peculiarity of the EICAS approach to the control design is the “reference generator” . Starting from the reference command $r(i)$ received from the host, the “reference generator” (see Fig. 8) computes the open loop command $u_o(i)$ and the required output value $ry(i)$, which are strictly coherent with the plant simplified model adopted for the control plant design.

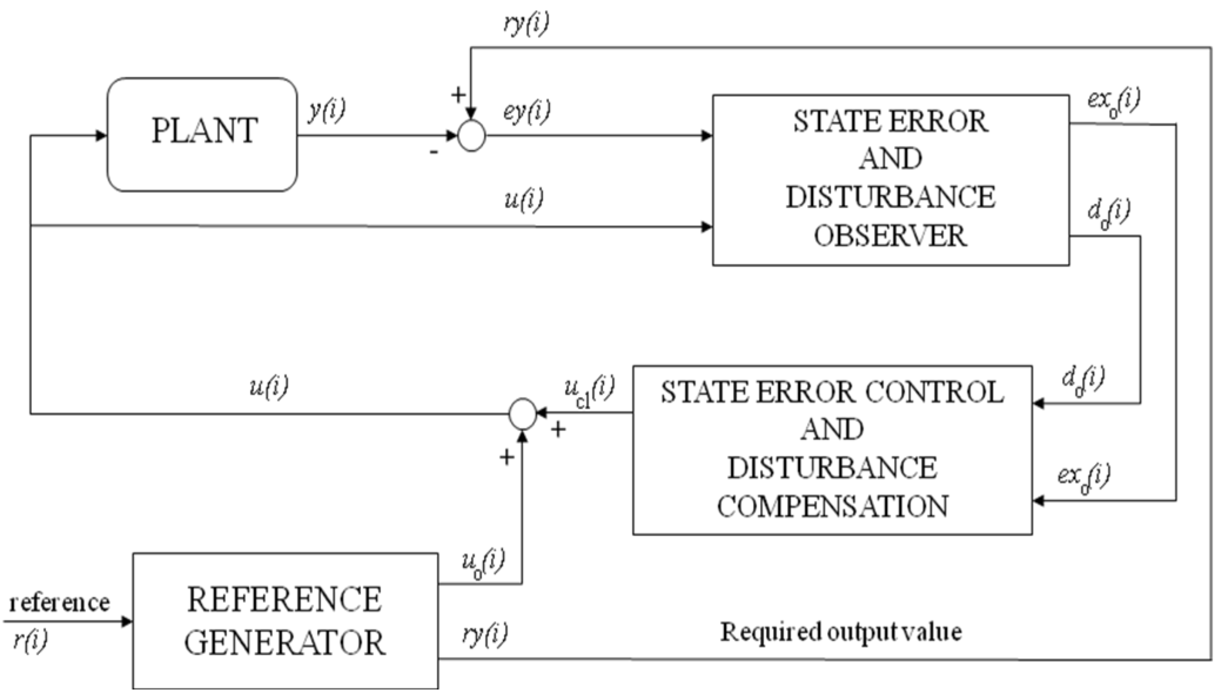


**Figure 10 The functional scheme of the plant control**

All the control architectures 1, 2 and 3 have been implemented following the EICAS model based approach, the main line of the control design is here shortly presented.

***Architecture 1.*** When the "simplified model" and the "fine model" are compared, the "simplified model" approximation does not result to be critical from the point of view of the feedback control design. In other words, denoting by $y_s$ and $y_f$ the motor angular rotation respectively output of the "simplified model" and output of the "fine model" under the same input $u(i)$, the following inequalities hold for any $u(i)$ belonging to the admissible input set:

$$||y_s\text{-}y_f|| < E\ ||y_s|| + D,\ \ E < 1$$

According to the plant control theory presented in [1] the above results mean that the "fine model" is not sufficiently "fine" to point out the upper frequency band limit outside of which the performance of control systems designed on the basis of the "simplified model" cannot be guaranteed. Indeed, in order to build a "fine model", which points out the limit of the adopted "simplified model" in modelling the relation between the commanded electric current (input) and the electric motor angular rotation (output), it was necessary to describe the relation between the commanded electric current and the resulting electric torque by a dynamic model, inside of the assumed simple proportionality law. In fact it is well known that a delay exists between the commanded current and the consequent electric torque produced by the motor. Anyway such a delay may be fully neglected in the frequency range specified by the control performance requirement.
In conclusion, when the architecture 1 is considered, if the control bandwidth is not enlarged too much the approximation of the "simplified model" in modelling the actual plant may be

neglected from the point of view of the feedback control design. In the EICAS control design the state observer bandwidth has been fixed at 40 Hz and in such a frequency band the delay between the commanded current and the produced electric torque may be fully neglected.
Then, in the architecture 1 case, the EICAS model based control differs from Classic Model Based Control just only for the EICAS state observer implementation.
The main limit of the control architecture 1 is in the fact that the controlled output is the electric motor angular rotation, while the control accuracy requirement is referred to the rotating table angular rotation, which differs from the controlled output for the mechanical gear flexibility and backlash.

***Architecture 2.*** When the “simplified model” and the “fine model” are compared, the “simplified model” approximation results to be critical from the point of view of the feedback control design for having neglected the mechanical gear flexibility and backlash, which cause a significant delay between the commanded motor current and the rotating table angular acceleration. As already pointed out in § 1.4.2, the axis flexibility causes plant proper vibrations, the frequency of which has been evaluated to be in the range 20 – 24 Hz.
Then, the frequency bandwidth of the control system designed on the basis of the simplified model has been limited by placing at the frequency of 4 Hz the poles of the EICAS “state error and disturbance observer”. The table proper vibrations, measured by the encoder 2, are attenuated by the “state error and disturbance observer” of a factor ranging in the field 1/200 ÷ 1/100, which is considered sufficient to guarantee that plant proper vibrations cannot be amplified by the feedback control.

***Architecture 3.*** This control architecture has been obtained as a synthesis of the architecture 1 and 2 above considered. The low level of control is strictly the same one designed for the architecture 1. The high control level has been designed with the same frequency bandwidth limits adopted in architecture 2.
The result is a large frequency bandwidth (40 Hz) in the control of the motor angular rotation and an accurate correction in the frequency bandwidth of 4 Hz of the effects of the mechanical gear flexibility, hysteresis and backlash.
The EICAS model based control design methodology is available in EICASLAB versions customized for specific application sectors. The user has to introduce the data values related to “fine model” and “simplified model”. Control algorithm and code are automatically generated.

### 3.3.4 Control performance evaluation and numerical optimization

The performance of the above control designs has been quantitatively evaluated by performing simulation trials where the designed control is applied to control the “fine model”. As illustrated in § 3.1 a specific test case has been defined including two different trials of 30 s each one. One trial is working in point to point operating mode, the other in tracking operating mode. In both cases the rotating table is assumed to be subjected to the perturbing torque illustrated in Fig. 8.
The actual error $e(i)$ of the rotating table angular position with respect to the reference value has been considered and decomposed in two components:

- low frequency component $e_l(i)$, obtained by a second order low pass band filter with frequency bandwidth of 4 Hz;

- high frequency component $e_h(i) = e(i) - e_l(i)$, obtained subtracting the low frequency component $e_l(i)$ from the rotating table angular position error $e(i)$.

Then, when the control system is working in tracking operating mode, the following performance indexes are computed.

- *Em* = mean value of $e(i)$
- *Et* = mean squared value of $e(i)$
- *Elf* = mean squared value of $e_l(i)$
- *Ehf* = mean squared value of $e_h(i)$

When the control system is working in point to point operating mode, the error $e(i)$ of the rotating table angular position with respect to the reference positioning value is considered starting from instant at which the required position should be reached until the instant at which a new point is commanded. Such an error is denoted as “positioning error” and the following two performance indexes are computed in the point to point trials:

- *PEm* = mean value of the “positioning error”
- *PEt* = mean squared value of the “positioning error”

Moreover when the control is working in tracking mode the following control cost functional *F* has been introduced:

$$F = \| e_l \|^2 + w \, \| e_h \|^2 \qquad (4)$$

Where *w* is a weight coefficient to which has been attributed the value *w* =10 with the aim of penalising in a strong way any plant vibration in the frequency field larger than 4 Hz.

In order to optimize the PID Control and the Classic Model Based Control the above cost functional F has been minimized by a numerical iterative procedure with respect to the control system parameter values, namely:

- *PID Control*: the three coefficients related respectively to the proportional, integrative and derivative control actions;
- *Classic Model Based Control*: the two coefficients related to the second order state observer and the two coefficients related to the second order state controller.

# 4 CONTROL PERFORMANCE RESULTS

The control performance results are summarized in the following two paragraphs.

The § 4.1 shows the results obtainable running the EICASLAB DEMO under Linux OS and the § 4.2 shows the results obtainable running the EICASLAB DEMO under Windows OS. Differences are due to the fact that EICASLAB DEMO makes use of random functions to generate disturbances (like - for instance - the random component of the perturbing torque applied to the rotating table), that are implemented with different algorithms in the two operative systems, bringing to little differences in final results.

Both in § 4.1 and in § 4.2, the results are related to the test case described in § 3.1 consisting in two trials performed respectively in point to point operating mode and in tracking operating mode. The reported data are referred to a simulation time of 30 seconds.
The control performance are always evaluated by considering the actual error *e(i)* of the rotating table angular position with respect to the reference value, independently from the fact that the electric motor rotation angle is measured and closed loop controlled (architecture 1) or the rotating table angle is measured and closed loop controlled (architecture 2) or both the above rotation angle are measured and controlled (architecture 3).

### 4.1 Performance in Linux OS

| Positioning Error in point to point operating mode | | | |
|---|---|---|---|
| **Control Algorithm** | **measured rotation** | **mean value mrad** | **mean squared value mrad** |
| **PID** | motor | 0.100 | 1.80 |
| | rotating table | 0.007 | 1.08 |
| **Classical Model Based** | motor | 0.048 | 1.85 |
| | rotating table | -0.134 | 2.59 |
| **EICAS model based** | motor | 0.053 | 1.76 |
| | rotating table | 0.015 | 0.78 |
| | motor & rotating table | -0.010 | 0.37 |

**Table 3 Comparison among different control structures in point to point operating mode**

| Tracking Error in tracking operating mode | | | | | |
|---|---|---|---|---|---|
| **Control Algorithm** | **measured rotation** | **mean value mrad** | **mean squared value** | | |
| | | | **total value mrad** | **low frequency mrad** | **high frequency mrad** |
| **PID** | motor | -0.415 | 2.48 | 2.43 | 0.49 |
| | rotating table | -0.068 | 1.71 | 1.35 | 1.04 |
| **Classical Model Based** | motor | -0.433 | 2.58 | 2.53 | 0.49 |
| | rotating table | -0.398 | 3.82 | 3.3 | 1.92 |
| **EICAS model based** | motor | -0.377 | 2.40 | 2.35 | 0.50 |
| | rotating table | 0.009 | 1.18 | 0.89 | 0.78 |
| | motor & rotating table | 0.002 | 0.61 | 0.12 | 0.60 |

**Table 4 Comparison among different control structures in tracking operating mode**

## 4.2 Performance in Windows OS

| Positioning Error in point to point operating mode | | | |
|---|---|---|---|
| **Control Algorithm** | **measured rotation** | **mean value mrad** | **mean squared value mrad** |
| **PID** | motor | -0.443 | 1.74 |
| | rotating table | 0.067 | 1.06 |
| **Classical Model Based** | motor | -0.422 | 1.80 |
| | rotating table | -0.165 | 2.57 |
| **EICAS model based** | motor | -0.415 | 1.70 |
| | rotating table | -0.021 | 1.00 |
| | motor & rotating table | -0.001 | 0.55 |

**Table 5 Comparison among different control structures in point to point operating mode**

| Tracking Error in tracking operating mode | | | | | |
|---|---|---|---|---|---|
| **Control Algorithm** | **measured rotation** | **mean value mrad** | **mean squared value** | | |
| | | | **total value mrad** | **low frequency mrad** | **high frequency mrad** |
| **PID** | motor | -0.099 | 2.45 | 2.39 | 0.52 |
| | rotating table | -0.054 | 1.70 | 1.31 | 1.08 |
| **Classical Model Based** | motor | -0.065 | 2.54 | 2.48 | 0.54 |
| | rotating table | -0.219 | 3.64 | 3.24 | 1.67 |
| **EICAS model based** | motor | -0.050 | 2.37 | 2.30 | 0.54 |
| | rotating table | 0.017 | 1.17 | 0.89 | 0.76 |
| | motor & rotating table | 0.006 | 0.70 | 0.09 | 0.69 |

**Table 6 Comparison among different control structures in tracking operating mode**

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