Development of the Actively-controlled Beds for Ambulances

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Abstract: During transportation by ambulance, the inertial acceleration acts on a patient when an ambulance decelerates or turns a corner. Such acceleration often gives a supine patient physical stress such as blood pressure variation and body sway, which results in pain or a feeling of discomfort. To reduce this undesirable effect of the acceleration, the actively-controlled bed, which controls a posture of the bed to cancel the inertial acceleration by the gravitational acceleration, was developed. This paper gives an overview of its development, including control system design and performance evaluation.

1. INTRODUCTION

An ambulance transfer service is required to take a patient quickly and safely to hospital. However, it is difficult to improve these requirements simultaneously since there is essentially a trade-off between quickness and safety: Quick transportation exposes a patient to large vibrational acceleration or inertial acceleration, which causes a strong pain or a feeling of discomfort of a patient. Although it is possible to reduce the acceleration by driving an ambulance carefully, it mostly results in a longer delay to hospital arrival.

As a solution to the trade-off problem above, the stretcher support systems which block transmission of the acceleration to patients have been developed. They can be classified roughly into two types. One is a suspension to isolate a patient from a road-induced vibration or shock (Stammres and Leyshon, 1985; Leyshon and Stammers, 1986; Raine and Henderson, 1998; Abd-El-Tawwab, 2001). With the passive type of suspension, the front-to-back acceleration on a supine person in the 3-20Hz range can be well reduced. In Japan, the air or magnetic isolator suspensions are widely used in high-standard ambulances. The other type is a posture control bed, which is called an actively-controlled bed (ACB), to absorb an inertial acceleration in braking and/or curve driving which causes blood pressure variation and/or side-to-side body sway (Sagawa and Inooka, 2002; Kawashima, 2002; Ono and Inooka, 2003, 2005). By actively controlling the posture angle of a stretcher with actuators, it effectively reduces the foot-to-head and/or the lateral acceleration on a supine person over the range of 0-1Hz without slowing the ambulance down.

This paper reports control system design and acceleration reduction effects of the 2-degree-of-freedom ACB which was developed as a prototype by cooperation of the authors and a Japanese ambulance manufacturer (Ono and Inooka, 2005). Essentially, robust posture control is required for the ACB since an inertia load of the actuator changes depending on the body weight of a patient. First, a special hardware mechanism for the robust control is explained. Also, accurate posture control is required for the ACB to reduce the acceleration sufficiently. To this end, the principle of matching (Zakian, 1989, 1991, 1996, 2005) was utilized to design a servo compensator for posture control. This design procedure and the control performance of the designed system are described in detail. Finally, the performance of the ACB is examined through driving experiments and simulation.

Fig. 1. 2-degree-of-freedom ACB

The following notation is used in this paper: The symbols $\mathbb{R}$ and $\mathbb{R}_+$ denote a set of all real numbers and a set of all non-negative real numbers, respectively. A response y of a system to an input u at time t is represented by $y(t, u)$.

2. OVERVIEW OF THE ACB

2.1 Structural specifications

The pictures of the developed ACB are shown in Fig. 1. It is approximately 2.0 meters long, 0.8 meters wide and 0.6 meters high: It is a little bigger than the commercial stretcher support system with air or magnetic suspensions. The stretcher is mounted on the upper frame of the ACB. The purpose of use of the ACB is to reduce the inertial

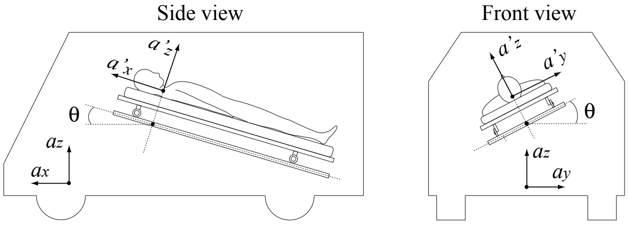

Fig. 2. Posture control of the ACB

acceleration in braking and curve driving. To this end, as Fig. 2 indicates, the ACB tilts and rotates a stretcher with two motors so that the acceleration acting on a supine person is cancelled by the gravitational acceleration. For example, the ACB tilts down the stretcher when the ambulance decelerates and rotates it when the ambulance goes round a curve or turns a corner. However, to ensure a working space for medical treatment in an ambulance, its motion range is restricted: The bed tilts from a horizontal position up to 12 degrees down and rotates from side to side up to 12 degrees. To isolate a patient from the road-induced front-to-back vibration and shock, an air suspension is also installed under the stretcher. Hereafter, as Fig. 2 illustrates, the acceleration of a vehicle is denoted by a_x, a_y and a_z and the acceleration acting on a supine person is denoted by a'_x, a'_y and a'_z.

2.2 Control system

The control system of the ACB consists of two independent servo systems for tilting and rotating the stretcher. They are illustrated by the same block diagram as shown in Fig. 3, where each block component and each signal are as follows:

P	:	Integrated system including the bed, the motor, the servo amplifier and a supine person
K	:	Servo compensator
F	:	Lowpass filter to adjust ride quality
a	:	a_x or a_y
r	:	Reference angle of the bed
r_m	:	Reference angle of the motor
u	:	Control input to a servo amplifier
y	:	Tilt or rotational angle of the bed
y_m	:	Rotational angle of the motor
$f(a)$	:	Reference function
$f_m(r)$	:	Conversion function from r to r_m
$f_b(y_m)$	:	Conversion function from y_m to y

The accelerations a_x and a_y are measured by accelerometers fixed on the floor of a vehicle. These acceleration signals include high-frequency noises due to engine vibration or shock from a road. For the system not to respond to such noises, they are smoothed by the lowpass filters F. This filter plays an important role to maintain good ride quality. The reference tilt and rotation angles of the bed are determined by the reference functions: $r = f(a)$. If a_x and a_y are constant, they can completely be cancelled by the gravitational acceleration g by tilting and rotating the stretcher by an angle of

$$\theta = \tan^{-1}\left(\frac{a}{g}\right) \tag{1}$$

However, since there are hardware constraints on the motion range of the bed, the reference angles are determined

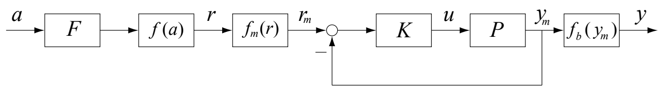

Fig. 3. Control system of the ACB

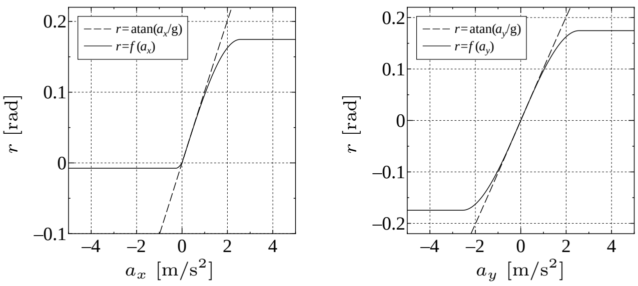

Fig. 4. Reference function (Left: reference tilt angle. Right: reference rotation angle)

by the saturated functions shown in Fig. 4. In these figures, the dashed line shows the reference angle calculated by (1) and the solid line shows the actual reference angle. Therefore, only the acceleration less than $2\mathrm{m}/s^2$ can be cancelled. The servo compensator K outputs an input voltage to the servo amplifier, which is proportional to a desired rotation speed of the motor. The servo amplifier works as a PI compensator for speed control of the motor. The operating frequency of the servo control unit is 100Hz.

Remark: Both the control systems for tilt and rotation control have the same structure. For this reason, the corresponding components and signals are represented by the same symbol without distinction.

3. CONTROL SYSTEM DESIGN

This section describes the hardware and software design of the servo control systems of the ACB.

3.1 Support mechanism for robust control

An inertial load of the motor is dependent on the body weight of a person, so robust control is essentially required for the ACB to control the posture of the bed accurately for any person. Generally, as actuator motors, smaller-powered motors are preferable in terms of power consumption. However, even for light weight persons, it was impossible to tilt up and down the bed with only a 200W motor because of shortage of torque. To do the tilt control without changing the 200W motor to a higher-powered one, an air suspension was attached between the upper and the bottom frames of the ACB to support the body weight of a person. The air pressure of the suspension is adjusted automatically depending on the motor torque: The air is supplied to or removed from the suspension when the motor torque exceeds pre-specified thresholds. As a result, an excessive load on the motor could be avoided and a small variation of the inertial load could be maintained for any person. On the other hand, the bed can rotate quickly with a 200W motor without any additional support mechanism since it is connected to the motor by a high-reduction gear.

3.2 Design of the servo control unit

The design components of the servo control unit are K and F, which are constructed as linear time-invariant (LTI) components. The role of K is to maintain a small tracking error between r and y, while the role of F is to adjust sensitivity of the control system to the acceleration in order to maintain good ride quality. This subsection describes the design of these components.

Model of the ACB The system from r to y has two major nonlinearities. One is a torque saturation of the motor due to an overload protecting function of the servo amplifier. The other is a nonlinear couple between the bed and the motor: $f_m(r)$ and $f_b(y_m)$ in Fig. 3 are nonlinear functions. However, as long as the motor torque is below a nominal torque (a maximum continuous torque), the torque can be generated as the PI compensator in the servo amplifier orders. Also, since the nonlinearities of $f_m(r)$ and $f_b(y_m)$ are not so strong, they can be approximated as linear functions:

$$f_m(r) \simeq \alpha r, \quad f_b(y_m) \simeq y_m/\alpha \tag{2}$$

for a certain positive real number α. Accordingly, the system from r to y can be approximately regarded as an LTI system unless the motor torque exceeds the nominal torque. That is, the system can be modeled as a conditional LTI system.

Design of K The servo compensator K must keep controlling the posture of the bed accurately for any possible reference command without malfunction due to overload of the motors. Otherwise, the acceleration is reduced halfway and a patient may feel bad rather than good. In this paper, such K is designed in the framework of the critical system design combined with the principle of matching (Zakian, 1989, 1991, 1996, 2005). In this case, a problem of designing K is formulated into an admissibility problem stated by the inequalities

$$\begin{aligned} \hat{e} &:= \sup\{|e(t,r)| : t \in \mathbb{R}_+,\ r \in \mathcal{P}\} \leq \varepsilon_e \\ \hat{\tau} &:= \sup\{|\tau(t,r)| : t \in \mathbb{R}_+,\ r \in \mathcal{P}\} \leq \varepsilon_\tau \end{aligned} \tag{3}$$

where $e := r - y$ is a tracking error, τ is the motor torque, ε_e is a tolerable bound of the error given by a designer, ε_τ is the nominal torque of the motor and $\mathcal{P}$ is a so-called possible set. The first inequality in (3) is associated with control accuracy, while the second inequality is a torque constraint to avoid system malfunction due to overload of the motor. The possible set $\mathcal{P}$ is given so that the feature of reference commands is fully reflected in $\mathcal{P}$. Generally, a servo control system is required to make a control variable respond quickly to change of a reference command. From this point of view, it would be natural to characterize a reference command with rate of change. For this reason, this paper gives $\mathcal{P}$ as a set of reference commands with bounds in rate of change:

$$\mathcal{P} := \left\{ r : \mathbb{R}_+ \mapsto \mathbb{R} : \begin{array}{l} |\dot{r}(t)| \leq D \text{ for } t \in \mathbb{R}_+ \\ r(0) = 0 \end{array} \right\} \tag{4}$$

where D is a positive real number. This set includes unbounded reference commands. But, note that the actual reference r is bounded by the reference function $f(a)$.

Under (3), the system from r to y can be regarded as an LTI system. So, assuming that all initial states of the system are zero, the inequalities condition (3) for (4) can equivalently be written as

$$\begin{aligned} \hat{e} &= \frac{D}{h} \int_0^\infty |e(t,h)| dt \leq \varepsilon_e \\ \hat{\tau} &= \frac{D}{h} \int_0^\infty |\tau(t,h)| dt \leq \varepsilon_\tau \end{aligned} \tag{5}$$

where $e(t,h)$ and $\tau(t,h)$ mean the tracking error and the motor torque for a step reference command with magnitude of h, respectively. The compensator K is constructed so that the set of inequalities (5) is satisfied.

The design under (5) requires ε_e, ε_τ and D. To maintain a small tracking error, give ε_e as

$$\varepsilon_e = 0.020 \text{ [rad]} \tag{6}$$

The nominal torque of the 200W motor is

$$\varepsilon_u = 0.637 \text{ [N·m]} \tag{7}$$

On the other hand, the value of D depends on F as well as a. At this stage, however, F is not yet determined. So, supposing $F = 1$, estimate D from the relation

$$D = \max \left| \frac{df}{da} \cdot \frac{da}{dt} \right| \leq \frac{\max |\dot{a}|}{g} \tag{8}$$

Table 1. Statistical information on $|\dot{a}|$

	Maximum value	99%-value
$|\dot{a}_x|$	6.18	1.91
$|\dot{a}_y|$	5.63	1.56

Table 1 shows the statistical information on $|\dot{a}|$. This is based on the acceleration which was measured for the ambulance in a flat city area in Japan. The data length is 1200 minutes. In the table, '99%-value' means a value of $|\dot{a}|$ at which the relative cumulative frequency of $|\dot{a}|$ is equal to 99%. From the table, it is found that the maximum value of $|\dot{a}|$ is considerably large. Such a large change of acceleration occurs, but it would be appropriate to treat it as an outlier. For this reason, instead of the actual maximum value, '99%-value' is used to determine D. Then,

$$D = \begin{cases} 0.195 \text{ [rad/s]} & \text{for tilt} \\ 0.159 \text{ [rad/s]} & \text{for rotation} \end{cases} \tag{9}$$

To satisfy the inequalities in (5), the system must have integral action: $e(\infty, h) = 0$ and $u(\infty, h) = 0$. According to the internal model principle, either of P and K needs to have integral action. Since the servo amplifier includes the PI compensator, this requirement is satisfied for any stabilizing K. For this reason, K is constructed as a proportional and lead-lag compensator, which is parameterized by three real numbers $p_i \in \mathbb{R}$ $(i = 1, 2, 3)$ as follows:

$$K(s) = \frac{p_1(1 + p_2 s)}{1 + p_3 s}, \quad p_3 > 0 \tag{10}$$

Accordingly, the design problem of K is formulated into an admissibility problem of finding a vector $p := [\, p_1\ p_2\ p_3 \,]^t \in \mathbb{R}^3$ which satisfies (5). First, a numerical search was conducted based on the LTI model of the system to find some candidates of the solution vector. Next, by inputting a step reference command directly to the ACB, the condition (5) was checked for those candidates. As a result, most of the candidates were confirmed as solutions to (5). Finally, one solution vector was chosen to implement K in the servo control unit.

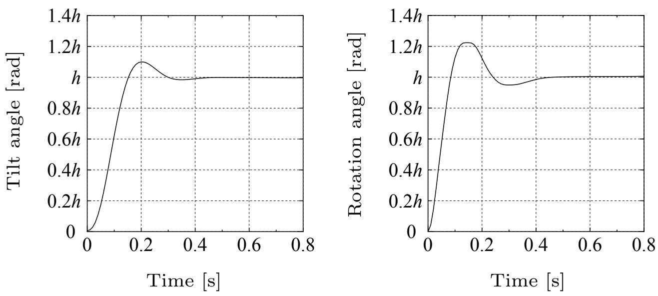

Fig. 5. Responses of the designed control systems to the step reference command $h = 0.01745$ [rad]

Table 2. Performance index

	Tilt control	Rotation control
Overshoot [%]	10.0	22.4
Delay Time [s]	0.09	0.05
Rise Time [s]	0.09	0.05
Setting Time [s]	0.26	0.31

The control performance of the designed compensator is examined here. Fig. 5 illustrates the response of y to a step reference command. Table 2 gives the performance indices. From the step responses in Fig. 5, the supremum of $|e|$ is estimated by (5) as

$$\hat{e} = \begin{cases} 0.019 \text{ [rad]} & \text{for tilt} \\ 0.012 \text{ [rad]} & \text{for rotation} \end{cases} \tag{11}$$

From these results, it can be confirmed that the designed control system has good tracking performance. However, the accurate response is not always good: When a vehicle decelerates rapidly due to a hard brake, the nose dive occurs and then a_x changes rapidly from a positive value to a negative value. If the system responds to such an excessive change of a_x, the bed rebounds upward and then ride comfort is lost. To prevent such rebound due to the nose dive, the system tilts up the stretcher slowly regardless of r when the hard brake is detected.

Design of F The main bandwidth of a_x in braking and a_y in curve driving are up to 1Hz, while the estimated servo bandwidth of the control system are up to 3.3Hz and 5.0Hz for the tilt control and the rotation control, respectively. So, the control system has wide servo bandwidth enough to respond to the acceleration. But, this also means that the system is rather sensitive to the reference command. Additionally, as Fig. 5 illustrates, the system has a small damping effect: The resonance appears at 1.3Hz and 2.8Hz for the tilt control and the rotation control, respectively. Therefore, if r is determined directly from a, the ACB may respond to r vibrationally. Such a vibrational response is not preferable to a patient since it increases a pain or a feeling of discomfort. To avoid this, the lowpass filter F is introduced to adjust the sensitivity of the system to the acceleration. The filter was determined by subjective evaluation of ride quality through test runs. As a result of trial and error, the following filter was adopted for both the tilt and the rotation control systems.

$$F(s) = \frac{1}{(1 + 0.2s)^2} \tag{12}$$

The cutoff frequency of this filter is 0.5Hz, that is, the path-band of r is limited up to 0.5Hz. Therefore, the ACB controls the posture of the stretcher smoothly without generating a large overshoot/undershoot.

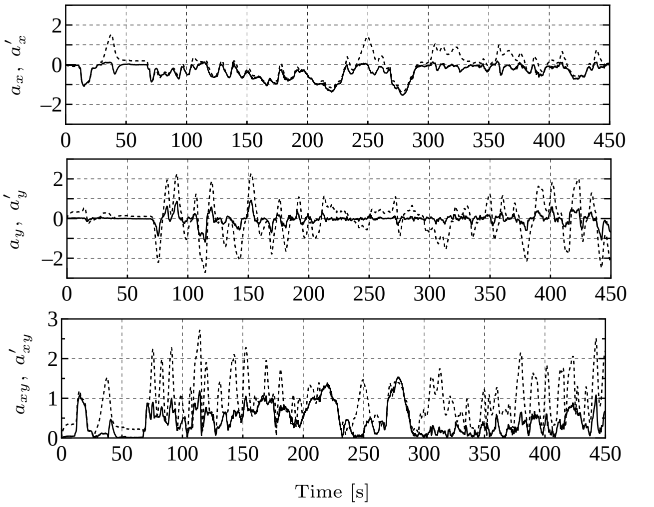

Fig. 6. Experimental result for Subject A (Unit of the acceleration : m/s^2)

4. PERFORMANCE EVALUATION

This section examines an acceleration reduction effect of the ACB through driving experiments and simulation.

4.1 Driving experiment

First, a reduction rate of acceleration is examined through driving experiments. The test car is a one van, which is used as a welfare car in Japan. Compared to the high-standard ambulances manufactured in Japan, it has a shorter wheelbase: The wheelbase of high-standard ambulances is 3.4m to 4.0m, whereas the wheelbase of the test car is 3.1m. The experiment is conducted for six subjects. For convenience, they are referred to as subjects A to F. The subjects lie on the stretcher quietly in a supine position with their eyes closed. The test car runs up and down a winding road at a speed of less than 60km/h for all the subjects. But, a driving pattern is different for each subject. The accelerations a_x, a_y, a'_x and a'_y are measured at a sampling frequency of 100Hz. Fig. 6 shows the acceleration measured for Subject A. In these figures, the dotted line shows a_x or a_y and the solid line shows a'_x or a'_y. The bottom figure shows the horizontal accelerations which are defined by

$$a_{xy} := \sqrt{a_x^2 + a_y^2}, \quad a'_{xy} := \sqrt{a'^2_x + a'^2_y} \tag{13}$$

From these figures, it turns out that the foot-to-head acceleration on braking and the lateral acceleration could be reduced sufficiently unless they exceed 2m/s^2. Introduce the indices η_x and η_y to evaluate the reduction rate of the acceleration acting on the subject.

$$\eta_x := \frac{\int_0^T |a'_x(t)|H(a_x(t))dt}{\int_0^T a_x(t)H(a_x(t))dt}, \quad \eta_y := \frac{\int_0^T |a'_y(t)|dt}{\int_0^T |a_y(t)|dt} \tag{14}$$

in which T is a data length and $H(\cdot)$ is the Heaviside function. These indices are less than 1.0 if the acceleration on the subject is reduced by the ACB. But, note that η_x evaluates only the reduction rate on braking. The result

Table 3. Reduction rate of acceleration

	Subject						Avg.
	A	B	C	D	E	F	
η_x	0.27	0.26	0.29	0.22	0.48	0.50	0.34
η_y	0.22	0.24	0.23	0.21	0.26	0.26	0.24

is shown in Table 3. From this result, it was confirmed that the ACB could reduce the acceleration on the subject approximately by 70%.

4.2 Simulation

The acceleration reduction effect depends on a driver's skill and a traffic condition even if drivers drive the same vehicle on the same road. Next, based on the acceleration data measured for the ambulance, hardware simulation is conducted to examine the reduction rate of acceleration which could be achieved in an actual transportation.

A simulation is conducted by inputting a_x and a_y, which were measured for the ambulance in Japan, to the servo control unit of the ACB. The length of input data is 255 minutes. During the simulation, the subject lies on the stretcher quietly in the parked test car. The body weight of the subject is 67.5kg. The acceleration a'_x and a'_y is estimated by

$$\begin{aligned} a'_x &= a_x \cos y_x - g \sin y_x \\ a'_y &= a_y \cos y_y - g \sin y_y \end{aligned} \tag{15}$$

where y_x and y_y are the tilt and the rotation angles of the bed, respectively.

Fig. 7 shows the relative frequency distributions of a_x and a_y used in the simulation. Fig. 8 shows the relative frequency distributions of a'_x and a'_y estimated by (15). Table 4 shows the statistical data of a_x and a_y which are derived from Fig. 7. Table 5 shows the statistical data of a'_x and a'_y which are derived from Fig. 8. In these tables, '+' and '−' mean a positive side and a negative side of the acceleration, respectively. The peak value and the 99%-value are obtained from the frequency distribution in each side. From these statistical information, it is expected that the ACB can has a good acceleration reduction effect in the actual transportation. Table 6 shows the reduction rate of acceleration. On an average, the acceleration acting on a supine person is reduced approximately by 70%. The reason why the average of η_x is larger than that of η_y is that the ACB tilts up the stretcher slowly in hard braking and so at that time a'_x cannot be reduced sufficiently.

5. CONCLUSIONS

This paper described the control system design of the 2-degree-of-freedom ACB and its acceleration reduction performance. From the experimental and simulation results, it is expected that the foot-to-head and lateral acceleration acting on a supine patient can be reduced approximately by 70%. Accordingly, the ACB will be effective in preventing the variation of blood pressure in braking and the body sway in curve driving.

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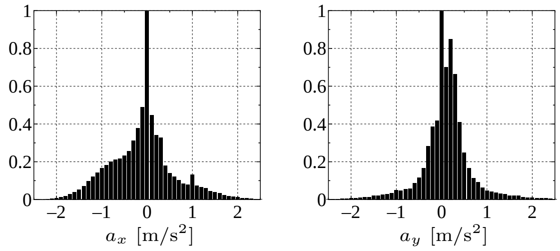

Fig. 7. Relative frequency distributions of a_x and a_y of the ambulance in Japan

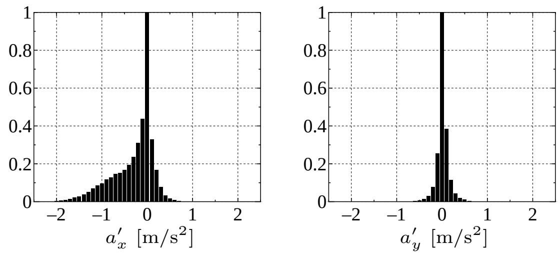

Fig. 8. Relative frequency distributions of a'_x and a'_y estimated by the simulation

Table 4. Statistical data of a_x and a_y

	a_x [m/s^2]		a_y [m/s^2]	
	+	−	+	−
Peak	2.69	-2.37	3.21	-3.09
99%-value	1.96	-1.75	1.93	-1.95

Table 5. Statistical data of a'_x and a'_y

	a'_x [m/s^2]		a'_y [m/s^2]	
	+	−	+	−
Peak	1.84	-2.34	1.48	-1.30
99%-Value	0.62	-1.68	0.53	-0.53

Table 6. Expected reduction rate of acceleration achieved by the ACB

	Avg.	Min.	Max.	Std.
η_x	0.35	0.13	0.51	0.08
η_y	0.25	0.10	0.33	0.06

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