Chapter 25
A Strategy for Improving Performance in Real Time Hybrid Testing

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Abstract Hybrid testing is an emergent technology encompassing a variety of contemporary methods such as hardware-in-the-loop, pseudo-dynamic, and real-time dynamic testing. A system to be tested is split into two or more subsystems, with some of the components represented by numerical models and the remainder being comprised of real, physical hardware. Forces and displacements are transmitted between subsystems via actuators and sensors. This paper is concerned with the challenges that emerge when trying to accommodate real-time simulations with highly nonlinear force characteristics in the physical substructure. A brief review of hybrid testing considerations is provided, including actuation hardware, controllers, and numerical time-integration schemes. An new approach is then proposed which unites novel methods in the numerical model, the integration scheme and the actuator control to achieve high performance levels independent of the physical component being tested, specifically for the case of highly nonlinear components and relatively coarse timesteps in the numerical substructure. Simulation results are provided to substantiate the projected benefits.

Keywords Hybrid testing • Real time dynamic substructuring • Hardware-in-the-loop • Independent integration • Nonlinearity • Force feedback

25.1 Introduction

Hybrid simulation is an experimental technique which is rapidly attracting interest in a number of fields. Improvements in the availability of actuation hardware, computing power and control techniques have recently allowed significant advances in the fidelity of these mixed numerical-physical simulations. Drawing principally from two established experimental testing platforms, hardware-in-the-loop (HiL) and substructured pseudo-dynamic testing (PDT), the method combines the advantages of numerical modelling and simulation with those of experimental testing to allow complex, strongly coupled structural components to be tested in the laboratory under realistic operating conditions.

The principle behind the method is to split a structure into two or more substructures. At least one of these substructures will be a physical piece of hardware while the remainder of the substructures will be simulated numerically. The physical substructure(s) will either be a critical part, where its exact performance is to be studied, or will be a complex system which is difficult to model numerically. Alternatively it may simply have unknown characteristics which are most easily determined experimentally. In contrast, the pertinent features of the numerical substructure will generally be well understood, and suitable for numerical modelling. Reasons for omitting these parts from a physical study are usually based around the logistics of laboratory testing: the parts could be too large to fit in a laboratory, too expensive, or they could have demanding load and displacement requirements for which the test equipment is unavailable. The latter may include distributed forces, coupled aero- or hydro-dynamic forces, or simply forces and displacements which are too large or too numerous for the available test equipment. Another reason for the growing popularity of this type of test is that it offers a realistic, repeatable environment for the rapid development of components where safety measures are easily put in place to mitigate catastrophic failures.

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The terms *hybrid simulation*, or *hybrid testing* encompass a group of techniques such as Hardware-in-the-Loop (HiL) testing, pseudo-dynamic (PsD) testing (PDT) [1–4] and real-time dynamic substructuring (RTDS) [5,6]. Sometimes the terms hybrid testing and HiL testing are used interchangeably, but here the term HiL is reserved for systems where the physical hardware is electronic and the interface between the physical and numerical domains is purely electrical. In contrast, most other types of hybrid test will incorporate structural and mechanical elements in the physical domain, with the interface between the two domains comprised of forces and displacements. Such tests have been pioneered predominantly by civil engineers for seismic testing of large structures, first with the quasi-static PsD tests, and more recently with the real-time substructured PsD tests, or RTDS.

Real-time dynamic substructuring differs from HiL and substructured PDT in the critical nature of the experimental coupling between the numerical and physical parts. For example, HiL is used to evaluate real electronic hardware with simulated operating conditions. Here, the coupling between the two systems is comprised of electrical connections, which will closely match those of the in-service equipment. In general there are no significant elements in the interface which will introduce spurious behaviour to the system. Like real-time dynamic substructuring, PDT is concerned with structural dynamics, where instead of electrical signals, forces and displacements are passed between the substructures. Here the interface is comprised of force and displacement transducers, with actuation devices (typically hydraulic or electromagnetic) providing displacement on the physical side. The control and response of the actuators, combined to some extent with the signal processing from the transducers, introduce significant dynamics into the coupled physical-numerical system. In PDT the tests are not conducted in real time, and these dynamics can effectively be eliminated. In real-time dynamic substructuring, however, the real-time response is critical and the spurious dynamics will have a detrimental effect on the accuracy and stability of the test results. Compounding this effect, the extent to which the results are affected is very difficult to determine without a reference system for comparison (and the existence of such a system would obviate the need for the real time dynamic substructuring).

The problems associated with real-time hybrid testing (RTHT) or structural components are therefore twofold: those of stability and fidelity. The former needs to be addressed for RTHT to be viable in the first place, but improvements in fidelity will ultimately resolve the problem of stability. Lack of stability is caused mainly by delays in the control loop at the interface, be those uniform delays in the form of latency [7, 8] or frequency-dependent delays in the form of lag [9]. Loss of fidelity is associated more generally with the intricacies of the dynamics at the interface, including but not limited to delays and lags.

The sources of the spurious dynamics can be separated broadly into two systems: Firstly, the physical actuation system and its controller will contribute unwanted dynamics to the system. Some approaches accept the limitations of the proprietary actuator and the associated inner-loop control and seek to mitigate their dynamics using an outer-loop controller. The purpose of this outer-loop control is to invert the action of the inner-loop controller and provide a feedforward signal to compensate the physical dynamics, with these two processes often combined into a single step [10–16]. A complication lies in the inevitable coupling of the physical dynamics to the component being tested, whose dynamics are generally not well-characterised. Adaptive control is commonly employed to compensate for this [17, 18], and also accommodates gradual changes in the physical characteristics of a component with time, for example in the case of fatigue failure of a component. The work in this paper does not address the actuator control problem, so for the examples presented an assumption of high-fidelity, high-bandwidth actuation is made as a prerequisite.

The second system where spurious dynamics may be introduced is in the ODE integration scheme used to time-march the numerical model. This can only ever offer an approximation to the full dynamics of the system, and inaccuracies here are a more general problem for numerical simulations as a whole, but the computational constraints imposed by the need for real-time evaluation mean that larger timesteps must be used and more frugal algorithms, leading to larger discrepancies. These are exacerbated by the need for an explicit scheme (due to a need for determinism in the physical substructure), and the unavailability of variable timesteps for real-time execution. Much work has been devoted to the development of schemes specifically for this purpose [19–21]. Fixed timestep, explicit integration schemes also suffer from an ostensibly inescapable loop delay introduced by the quantisation of the measured signals: the time step to which the measured force pertains has already been computed by the time the force is available to the integration scheme. A variety of techniques seek to mitigate this problem, often with the same measures used to counter pure delays or latencies in the actuator control, but ultimately they all amount to extrapolating from the current timestep to predict the actuator demands for the next (or even further ahead). While this works well for smooth, linear systems, and can be validated for the numerical model, it crucially does not account for large variations in the force-displacement, force-velocity and/or force-acceleration characteristics of the physical component within a timestep.

The work in this paper seeks to address two aspects of this problem. The first, and main purpose of the work is to allow the actuator demand to respond to changes in the measured reaction force from the physical component /it throughout a timestep. The motivation for this lies in the simulation of contact dynamics, in the first instance for the simulation of air-to-air refuelling scenarios (and also applicable to satellite docking simulations [22–26]). In these cases, abrupt, stiff contact is made between two bodies, and the small-scale, generally linear dynamics occurring in a fraction of a timestep will have a large influence on the behavior of the slower, usually nonlinear macroscopic behaviour of the system. Importantly, a failure to respond to reaction forces within a timestep will lead not only to artificially high contact forces, but also to the possibility of component damage or worse. The second aspect that must also be tackled is how to incorporate these transient dynamics into the slower timesteps of the numerical model.

25.2 Theoretical Framework

The premise of the work presented here lies in a closer integration, if you'll pardon the pun, between the numerical ODE computation and the physical actuation control.

Conventional approaches treat the two elements of the set-up separately, with the output of the numerical computation producing an actuator position or velocity demand, and a dedicated position- or velocity-controller used to meet this demand. The resulting measured forces are then used in the simulation under the assumption the position demands have been followed exactly. This introduces two problems: firstly, discontinuities and significant nonlinearities encountered in the test specimen between numerical timesteps are not accounted for in the demanded motion of the actuator. Secondly, the measured forces being used in the simulation will not reflect those that relate to the numerical position demands but those corresponding to the actual motion of the actuator, which will differ. The former can result in unrealistic and potentially destructive position demands that would not be encountered in a realistic simulation, and it is this aspect of the problem that motivates the work presented here.

In this study, an outer-loop actuator control is introduced, implementing a greatly simplified version of the numerical model by linearising about the full numerical model's state. This outer-loop control is an *independent integration* scheme which integrates the motion of only the interface degrees-of-freedom (DOFs), reacting to both force- and position-feedback, allowing the position and velocity demands to react with high bandwidth to nonlinearities in the physical specimen. The independent integration runs at a higher clock rate than the simulation itself and thus reacts to nonlinear behaviour encountered between the timesteps of the numerical simulation. The resultant discrepancy between the numerical and physical states at the interface are then reconciled using a filter on the physical displacement demand.

The linearisation for the independent integration is performed using the model state at each timestep. The full ODE solver for the numerical substructure computes the derivatives of the interface DOFs' motion with respect to time as well as the derivative of those derivatives with respect to the measured interface forces. The intention is for the second derivatives to be computed through parallelisation, allowing the model timestep to remain unchanged with respect to a more conventional approach. The parallelisation comes at a greater computational cost but has the advantage of versatility when using the technique with existing models.

The independent integration computes the displacement derivatives from a first order approximation:

$$\dot{\mathbf{y}} = \dot{\mathbf{y}}_\mathbf{0} + J(\mathbf{f} - \mathbf{f_0}) \tag{25.1}$$

where $\dot{\mathbf{y}}$ is the velocity vector for the interface DOFs at the most recent numerical model timestep, $\mathbf{f_0}$ is the measured force at the most recent timestep, $\mathbf{f}$ is the current measured force in the independent integration loop, and J is the Jacobian of the velocity vector with respect to interface forces (computed at the most recent timestep),

$$J = \begin{bmatrix} \frac{\partial y_1}{\partial f_1} & \frac{\partial y_1}{\partial f_2} & \cdots & \frac{\partial y_1}{\partial f_n} \\ \frac{\partial y_2}{\partial f_1} & \frac{\partial y_2}{\partial f_2} & \cdots & \frac{\partial y_2}{\partial f_n} \\ \vdots & \vdots & \ddots & \vdots \\ \frac{\partial y_n}{\partial f_1} & \frac{\partial y_n}{\partial f_2} & \cdots & \frac{\partial y_n}{\partial f_n} \end{bmatrix} \tag{25.2}$$

where n is the number of interface DOFs. The independent integration then performs a normal explicit Euler integration with respect to $\dot{\mathbf{y}}$ with a timestep the fraction of the size used for the main numerical model's integration:

$$\mathbf{y_{i+1}} = \mathbf{y_i} + \dot{\mathbf{y}}_\mathbf{i}\Delta t \tag{25.3}$$

where Δt is the independent integration scheme's timestep. Thus the independent integration will be equivalent to a standard Euler integration step for the main model if the measured forces are unchanged throughout the timestep so that $\mathbf{f} = \mathbf{f_0}$.

Where $\mathbf{f} \neq \mathbf{f_0}$, the position output from the independent integration, and thus the nominal actuator position, will show discrepancies with respect to the main model integration. To reconcile this difference, a simple reconciliation gain, α, is used so for each iteration of the independent integration:

$$\mathbf{y_{i+}} = (\alpha - 1)\mathbf{y_i} + \alpha \mathbf{y_0} \tag{25.4}$$

A value of unity for the reconciliation gain results in the actuator demand following the full numerical model integration. A value of zero allows the independent integration output to drift with respect to that of the main integration (although the integration parameters will still be tied to those of the main integration output).

A final step is to correct the forces used in the full numerical model. Because the independent integration is reacting to transient forces on much shorter timescales than the main model integration, sampled forces may not be representative across a full timestep. The approach taken here is to instead feed back the mean force across that timestep to the full numerical model. Future work will use a similar, but more rigorous approach akin to that of Darby et al. [8].

25.3 "Experimental" Setup

The scheme is evaluated here with respect to a simple one-DOF system. This system allows for fast numerical integration of an "ideal" hybrid test, using a variable timestep solver: the explicit 4th/5th order Runge-Kutta method implemented in MATLAB's ODE45 solver. While a numerical substructure of this nature does not require the techniques developed here, it can represent a more complex system by simply artificially slowing the speed at which its timesteps are computed.

The spring-mass-damper system used for the numerical substructure can be seen in Fig. 25.1. The properties are listed in Table 25.1. It uses an explicit Euler integration scheme with timesteps of 0.01 ms (sampling rate of 10 Hz). Also pictured is the nonlinear component used to simulate a physical substructure. This is comprised of a pre-loaded spring with a spring constant of 500 N/m, compressed by 20 mm with a natural equilibrium at a displacement of zero. The physical substructure is simulated using a ODE45 variable time-step solver to provide a realistic continuous representation of its physical behaviour.

This paper does not address the control of the actuator hardware itself; as explained above, for highly nonlinear physical systems and no a priori knowledge of the test specimen, there really is no substitute for high fidelity inner-loop actuator control. On this premise the tests that follow will assume a well-tuned inner-loop controller providing a high bandwidth actuator response that does not impinge on the system dynamics. This idealised representation can be justified by noting that the purpose of the tests that follow are to establish the merits of the independent integration approach; if successful, future work will focus on its implementation in conjunction with high-fidelity inner-loop controllers in a representative environment.

For all the tests that follow the independent integration is run at 2 kHz: a speed which is appropriate to the inner-loop demand rates for a range of typical actuators.

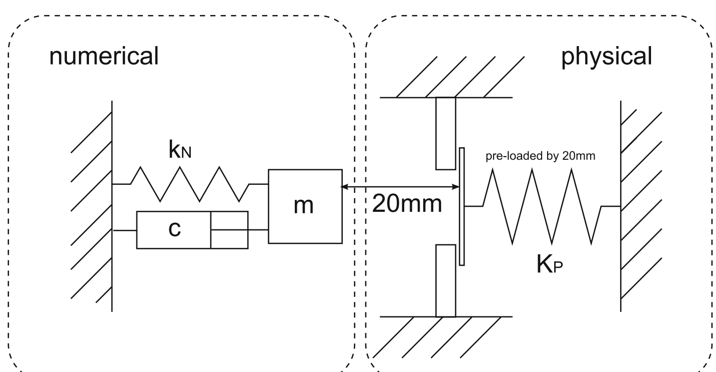

Fig. 25.1 Configuration of the hybrid test for the demonstration of the independent integration approach

Table 25.1 Parameters for the hybrid test simulation

k_N	40	N/m
K_P	500	N/m
c	6.3	Ns/m
m	1	kg

25.4 Results

Figure 25.2 shows the step response of the numerical system to a unit force using a range of discretisation timesteps. The figure shows the expected second order underdamped response with an overshoot followed by a fairly rapidly decaying oscillation. The coarser timesteps result in reduced damping from the explicit Euler integration scheme, but all of the cases presented are well-behaved and moderately well matched to the continuous model (simulated with a Runge-Kutta45 variable timestep solver). The notable exception is the 50 ms timesteps which shows clear discrepancies in both frequency and amplitude of response, but the 10 ms case in particular shows reasonably close agreement with the continuous model.

This behaviour changes when the "physical" spring is added to the system. The physical spring is more than tenfold stiffer than the numerical model's stiffness. The results can be seen in Fig. 25.3. The 50 ms response has been omitted this time, but even the 10 and 20 ms responses can be seen to differ wildly from those of the continuous model. (Note that the continuous model in this case is equivalent to a real hybrid system where the numerical model is evaluated at infinitesimally small timesteps by an extraordinarily powerful computer.) While the continuous version shows a small rebound from the impact with the spring, which decays to a steady state, the discrete versions show a larger rebound and no discernible decay. This behaviour is due to the overshoot of the numerical model's position caused by the large timestep. This effective delay in the force feedback adds negative damping to the numerical simulation and adds artificial energy to the hybrid test. This explains the undecayed response of the two discrete cases, even in the presence of significant damping in the numerical model.

Figure 25.4 shows the results of the independent integration test, for two different values of the reconciliation gain, α. The most important benchmark is how well the magenta line, representing the physical displacement of the interface DOF, agrees with the ideal displacement shown by the black line. In both cases the agreement is reasonably good, and a substantial improvement over the simple discrete solvers of Fig. 25.3. The higher gain in Fig. 25.4b ensures the closest alignment between the numerical and physical interface DOFs, and the result is a response decay that closely matches that of the ideal system. It does this at the expense of the temporal fidelity, however, so the spectral content of the response is compromised. This is manifested as a larger time interval between each rebound. The lower gain, in Fig. 25.4a, shows better temporal (and therefore spectral) agreement but with two drawbacks: firstly, the decay rate is less accurate; the amplitude response has been compromised. Secondly, and significantly, the reduction in the reconciliation gain has resulted in a larger discrepancy between the physical and numerical displacements at the interface DOF. In this simple example, the consequences are low, but with greater nonlinearity in the numerical substructure there could be a dramatic effect on the overall system's behaviour.

These early results show a promising new avenue to explore in providing more capable hybrid simulation capabilities in the presence of highly nonlinear behaviour in the physical substructure. It has been shown firstly that an integration timestep that's well suited to the dynamics of the numerical substructure can be inadequate when combined with a stiff, nonlinear physical component. It has then been shown that much closer agreement with the ideal hybrid system can be achieved with the independent integration scheme proposed here, with no change to the timestep used in the numerical model itself.

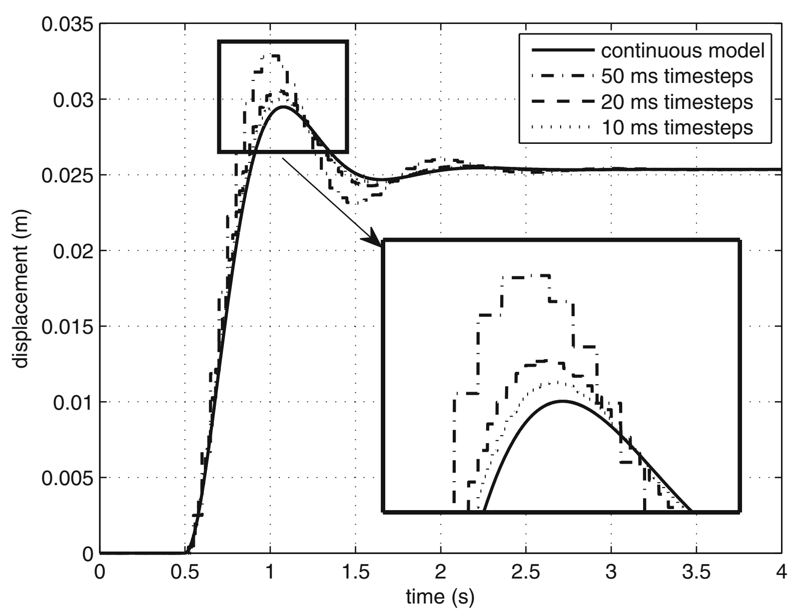

Fig. 25.2 Step response for the numerical system with a range of timestep discretisations

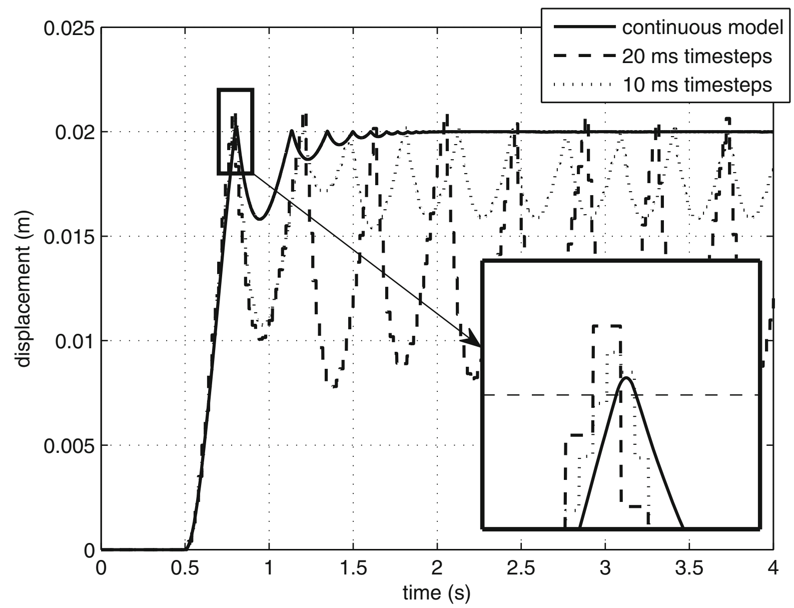

Fig. 25.3 Step response for the numerical system with a range of timestep discretisations

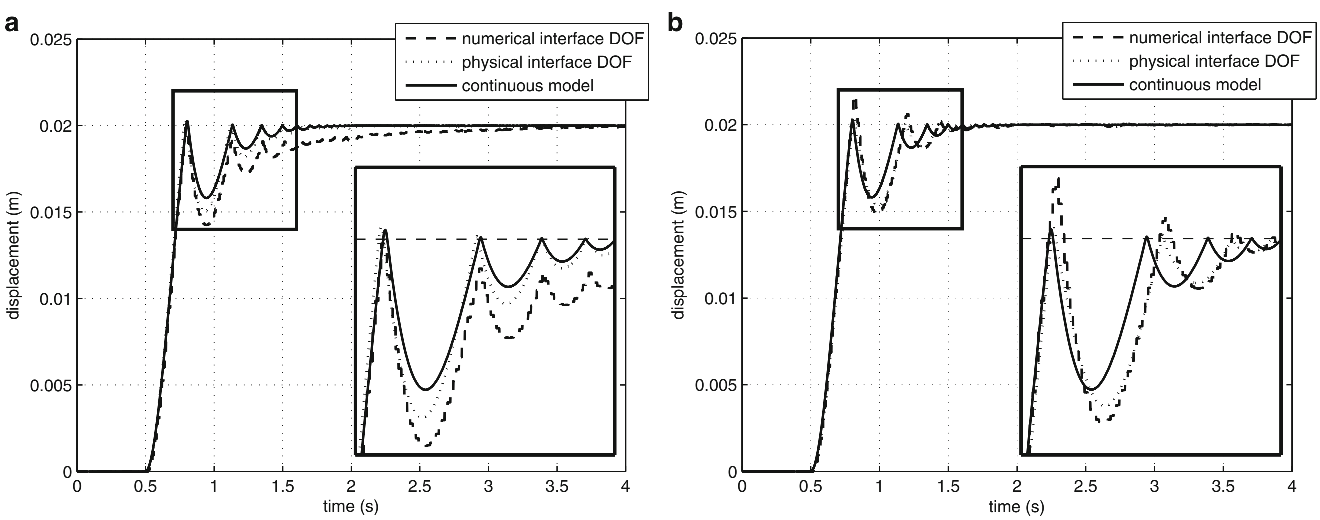

Fig. 25.4 Independent integration scheme compared with an ideal integration scheme for two different reconciliation gains. (**a**) Reconciliation gain of 1×10^{-3}. (**b**) Reconciliation gain of 5×10^{-3}

Rich pickings for future work are presented at this juncture: Firstly, the scheme needs to be demonstrated for a complex numerical substructure. Such a problem will offer a practical demonstration of the capabilities by introducing real constraints on the size of the timesteps. It will also introduce nonlinear behaviour to the substructure that will further test the assumptions that form the premise for the technique. Secondly, the technique needs to be verified in an actual hybrid test, with genuine physical hardware in contrast to the simulated physical hardware presented here. This more challenging environment will help demonstrate the relevance of the technique in the presence of other factors (loop delay, integration errors away from the interface, actuator control, etc.). Thirdly, the effect of the reconciliation gain needs a deeper investigation. Other schemes for reconciling the discrepancies between the numerical and independent, physical-based integration results can be explored. This could be combined with the force feedback to the numerical substructure, in a more rigorous formulation.

25.5 Conclusions

An independent integration scheme has been proposed to complement the full integration scheme of a numerical model in a hybrid simulation exercise, to permit the interaction of slower numerical model dynamics with fast, stiff, and highly nonlinear physical structures. The method uses a simple linearised model derived from the full model at each time step to implement a fast integration loop based on the physical measurements at the interface. Results from a simulated experiment show that the scheme has the potential to ameliorate the discretisation artefacts from a slow numerical simulation and permit fast dynamic interactions at the interface, with close agreement between the hybrid simulation and the ideal system it seeks to emulate. Plans for further work have been outlined, to verify the results in more complex systems and to extend the concepts presented here.

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