

Implementation of Transient Stability Model of Compressed Air Energy Storage Systems

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Abstract—This paper discusses the implementation of a transient stability model of Compressed Air Energy Storage (CAES) systems in a power system analysis package. A block-diagram based model of a two-machine CAES system is proposed, including specific controls for active power, reactive power, and State of Charge (SoC), which consider limits associated with the cavern pressure. As an application, the model is implemented in Powertech's TSAT[®] software connected to the 9-bus WSCC benchmark power system, which is then used to study the impact of a CAES facility in the transient and frequency stability of the system. Several contingencies are simulated comparing the CAES performance to a gas turbine and a base-case without storage, demonstrating that the CAES system improves the system transient stability due to its charging stage, controls, and additional inertia. Finally, the CAES model is used to study the effects of cavern sizes in the frequency of the system. It is shown that CAES systems have certain special characteristics that make them attractive as a storage technology to provide stability and regulation services, besides their energy arbitrage capabilities.

Index Terms—CAES, control, dynamic modeling, energy storage, frequency control, frequency stability, power system analysis software, transient stability, voltage control.

I. INTRODUCTION

The ability of Energy Storage Systems (ESSs) of decoupling generation and demand, which facilitate the integration of renewable generation, and their fast response and flexibility to provide multiple grid services is changing the way electrical systems are planned and operated. In this context, studying the impact of connecting ESSs to the grid under normal and abnormal conditions is essential. This requires the development of appropriate generic ESSs models for commercial power system software packages, which has been the focus of industry and academia, such as the Western Electricity Coordinating Council (WECC) and the Electric Power Research Institute (EPRI) [1], who have proposed generic transient-stability models of Battery Energy Storage Systems (BESS) for various software packages, including Powertech's TSAT[®] [2]. However, while research has concentrated on standardizing these models for storage technologies such as BESS and Flywheels, Compressed Air Energy Storage (CAES) has received little attention. Indeed, most of the existing CAES models aim to study the thermodynamic properties of the CAES components, such as the compressor and expander,

while few attempt to model the system connected to an electric grid, which is the focus of this paper.

Only a few CAES models suitable for transient stability studies have been proposed in the literature. Thus, in [3] and [4], a model of a CAES system is proposed and tested for some contingencies in a power grid; however, the generator/motor modeled is an induction machine, the equations of the compressor are steady-state only, and the expander is modeled and controlled as a Gas Turbine (GT). In [5], a model of a CAES system interfaced with the grid through a converter is proposed and studied for disturbances in the WSCC 9-bus test systems; however, the thermodynamic relations of the compressor and turbine are simplified, representing both with the same equation, while the dynamics associated with the expansion valve and Inlet Guide Vanes (IGVs) are not modeled. In [6], a CAES system used to buffer power output from a wind farm is modeled using converters and a dc link as interfaces between the wind farm, the CAES system, and the grid; however, the controls presented focus mostly on the converters, with a simple CAES model based on steady-state equations only, thus limiting their use as parameterizable generic models. In [7], detailed and simplified parameterizable CAES system models are proposed for frequency regulation; however, the controls associated with the State of Charge (SoC) are not modeled, compressor and turbine maps are not considered, the effects of the cavern pressure in the compressor and turbine controls are not modeled, and electrical machines are not represented in the simulations. None of the existing models include the reactive power controls of the machines.

Based on the previous discussion, the objective of this paper is to propose a comprehensive generic transient-stability CAES model, including all the relevant controls, and discuss its implementation in a power system simulation software package for transient stability studies. Therefore, the contributions of this work are:

- Propose a model architecture for implementation of a transient stability CAES model.
- Present coordinated reactive power controls for two-independent synchronous machine of a CAES system, and develop an SoC control logic.
- Model of the valve and associated controls for the air flow in the expansion stage.
- Include compressor and turbine maps in a transient stability CAES model.
- Study of the impact of a CAES system in the transient and frequency stability of a benchmark system, as well as analyze its oscillation damping and voltage regulation capabilities.

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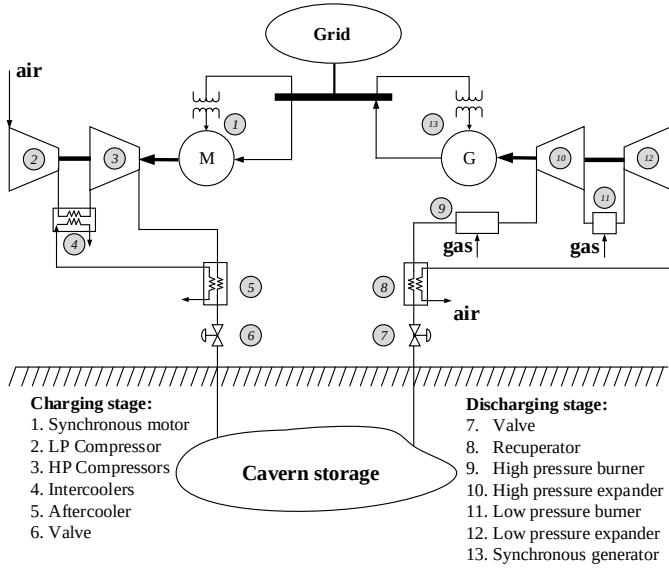


Fig. 1. Diabatic CAES system modeled.

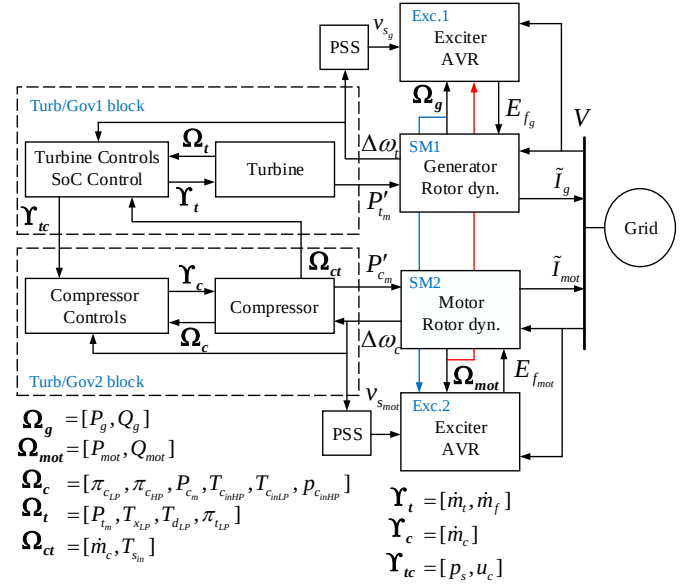


Fig. 2. Proposed CAES system model architecture.

The rest of the paper is organized as follows: In Section II, an overview of the CAES system and its component models are presented, and in Section III, novel control strategies for the CAES system are discussed. Section IV presents the implementation of the proposed model and the results of simulations, to evaluate the impact of a CAES system on the transient and frequency stability of a test system based on the WSCC 9-bus benchmark system. The main conclusions and a summary of the work are presented in Section V.

II. PROPOSED CAES TRANSIENT STABILITY MODEL

A. Overview of CAES Systems

As discussed in [8], in charging mode, a CAES system uses an electric motor driving a compressor to pressurize the air inside an underground cavern to store energy. In discharging mode, the compressed air is expanded in a turbine to produce electricity. In diabatic CAES systems, the air is preheated before being combined with gas and burned during the expansion, while in adiabatic CAES, the air is preheated using the remanent heat from the compression through a thermal storage element, thus avoiding burning fuel. A single synchronous machine can be used as generator/motor or, alternatively use two machines working as generator and motor independently, as assumed in this paper. These machines facilitate the active and reactive power control of CAES systems, compared to those based on induction machines, such as in [3] and [4], where the speed varies with the loading of the machines, thus requiring additional controls. Fig. 1 illustrates the different components of the independent two-machine diabatic CAES system modeled here, which are explained in detail in [7]. It is important to add that the nonlinear relations between the air flow, rotor speed and pressure ratio in compressors and expanders are defined in charts known as maps, usually provided by the manufacturers, which must also be taken into account in the modeling, especially for simulations of off-nominal conditions.

Active power control is possible in both CAES compressor and turbine. Compressor control is accomplished by regulating the Inlet Guide Vanes (IGVs), thus changing the airflow and the mechanical power. A complementary control is used to prevent the compressor from operating in surge condition [9]. The turbine has two main controllers: a temperature control system, which aims to regulate the exhaust temperature of the low-pressure turbine to its nominal value by adjusting the gas in the burners; and the power control, which is realized by a valve installed before the expansion, which changes the airflow through the turbine [10]. Reactive power control is accomplished by the Automatic Voltage Regulators (AVRs) of the synchronous generators.

B. CAES Representation

Time domain simulations in power system software packages require solving a set of Differential and Algebraic Equations (DAE), representing the system's components and their interconnections, using integration methods [11]. Most commercial packages, such as PSS/E®, DigSilent®, or TSAT® provide user-friendly interfaces, which ease the modeling of new components through the connection of fundamental blocks, such as gains, summation blocks, transfer functions, etc. Therefore, the CAES model proposed here is developed in block-diagram format, so that it can be readily integrated in popular packages for power system analysis.

The proposed architecture for the CAES system model is presented in Fig. 2. Thus, this system is connected to the grid through two synchronous machines which operate synchronized with the grid all the time. The synchronous machines inject the current phasors $\tilde{I}_g$ and $\tilde{I}_{mot}$ into the system, and regulate the input voltage V at the connection point. The compressor and turbine are modeled as turbine/governor blocks transferring mechanical powers P'_{cm} (negative) and P'_{tm} (positive) in the machine bases, with rotor speeds ω_c and ω_t , respectively. The cavern is modeled within the turbine

block and its pressure p_s is used to calculate the SoC. The information exchanged between the various blocks in Fig. 2 is presented through the vector of control variables Υ and mechanical and electrical variables Ω , where π stands for pressure ratios, $\dot{m}$ for mass flow rates, and T for temperatures, as explained in more detail in the next section.

The reactive power is controlled by the AVRs and the exciters, which define the field voltages E_{fg} and E_{fmot} of the machines. Additional control is required to ensure a proper distribution of the reactive power between the two synchronous machines; thus, feedback of their active and reactive power Ω_g and Ω_{mot} to both AVRs is required. Power System Stabilizers (PSS) are also modeled, which feeds the stabilizing signals ν_{sg} and ν_{smot} into the AVR to damp low-frequency oscillations in the system. For power-flow calculations, both motor and generator are modeled as two independent power injections in a voltage-controlled bus (PV bus), with the motor consuming power (negative).

The proposed CAES model overcomes the limitations of existing models, such as [5] and [6], by considering the dynamics and thermodynamic relations of some important subsystems such as intercoolers, controlling valves, air dynamics, and others. Furthermore, it also improves the model in [7] by considering compressor and turbine maps, anti-surge, discharging pressure, SoC, and reactive power controls.

C. Electrical System Model and Reactive Power Control

The synchronous machine model is extensively discussed in existing literature, and is usually available as a parametrizable component in a power system package. One of the most commonly used models for dynamic studies is the classical subtransient model, which is described in detail in [12]. In this paper, it is assumed that the rotor dynamics are part of the synchronous machine blocks; hence, the p.u. speed deviation $\Delta\omega$ is an output of this block, as shown in Fig. 2.

The CAES system is capable of providing reactive power through its two synchronous machines simultaneously; however, a special control is necessary to avoid instability, as the two independent AVR act to achieve the same objective, and to also avoid circulating currents between the two machines operating in parallel [13]. Hence, a proportional droop controller is used as shown in Fig. 3, which modifies the reference voltage in the main AVR loop to achieve the desired reactive power distribution. Reactive power reference values are calculated using the factors β_g and β_{mot} , such that $\beta_g + \beta_{mot} = 1$, which multiply the total reactive power produced by the CAES system, so that a fraction of it is assigned to each machine. These reference values are compared with the actual reactive power injections, and the error is passed through the proportional controllers with gain K_{vr} . The values of β determine the desired reactive power share, based on the following two alternatives:

$$\beta_{r1} = \left(\frac{S_{nr} Q_{r_{\max}}}{S_{ng} Q_{g_{\max}} + S_{nmot} Q_{mot_{\max}}} \right) \quad (1)$$

$$\beta_{r2} = \left(\frac{Q_{r_{\max}}}{Q_{g_{\max}} + Q_{mot_{\max}}} \right) \quad (2)$$

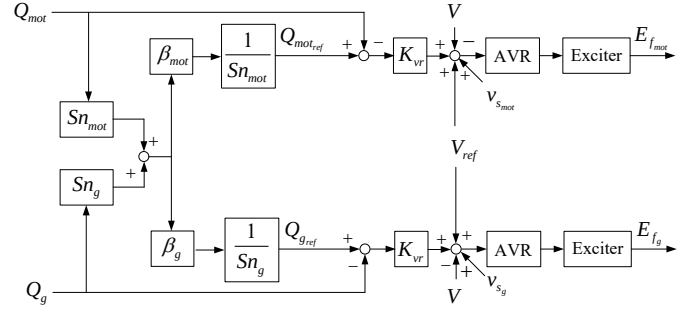


Fig. 3. Proposed reactive power controls.

where r is an index representing the generator g or the motor mot ; S_{nr} is the nominal power of the machine r ; and $Q_{r_{\max}}$ is the maximum reactive power the machine r can produce, given its current p.u. active power injections P_r and its nominal capacity S_{nr} in MVA. Notice that the value of β_r changes with the operating points of the generator and motor. It is assumed that the maximum loadability of the machines is determined by the thermal limit of the stator windings, represented by its rated capacity S_n ; hence, $Q_{r_{\max}} = \sqrt{1 - P_r^2}$. The choice of β_r depends on the objective sought; thus, β_{r1} prioritizes the amount of remaining capacity (in MVA) over the machine loading when assigning the reactive power share, while β_{r2} assigns more reactive power to the machine that is less loaded regardless of its size.

D. Mechanical System Model

The block diagram of the CAES mechanical systems and their controls are discussed in this section, with the rounded blocks representing functions of the inputs as opposed to standard transfer functions, which are represented with rectangular blocks. The thermodynamic models of the compressor and turbine are based on those proposed in [7], with the additional assumptions that both expander stages operate choked due to their large pressure ratios, the valve controlling the airflow during the expansion operates unchoked, and that maps for compressors are used. In the expander, these assumptions result in a constant pressure ratio in the High-Pressure (HP) expander $\pi_{t_{HPo}}$, while the pressure ratio in the Low-Pressure (LP) expander becomes a function of the per-unit airflow $\dot{m}_t$, per-unit LP inlet temperature T_{dLP} , and nominal LP pressure ratio $\pi_{t_{LPo}}$, as follows:

$$\pi_{t_{LP}} = \pi_{t_{LPo}} \sqrt{T_{dLP}} \dot{m}_t \quad (3)$$

1) *Compressor*: The compressor presented in Fig. 4 is modeled in a turbine/governor block; therefore, its output to the synchronous motor block P'_{cm} is the p.u. mechanical power P_{cm} transformed to the machine base. An LP compressor with one inter-cooler and an HP compressor are considered, with the latter representing a 3-stage centrifugal HP compressor with an after-cooler that have the same pressure ratios at the same inlet temperature, which result in the same mechanical power [7]. The compressor maps are represented by the LP Compressor Map Block (CMB) and HP CMB, which are used to calculate the pressure ratios $\pi_{c_{LP}}$ and $\pi_{c_{HP}}$, respectively

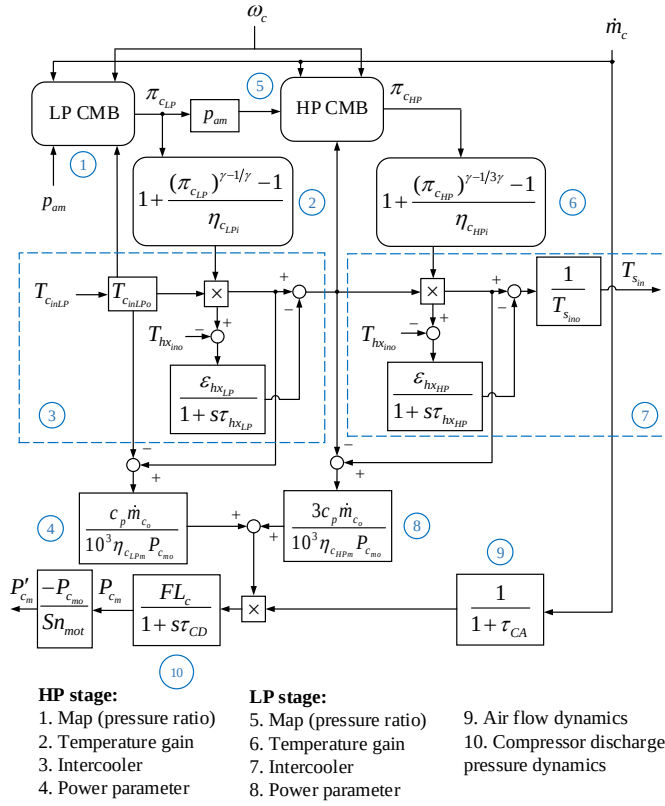


Fig. 4. Compressor model.

[14]. In this figure, η represents efficiencies; ϵ effectiveness of heat exchangers (intercooler and aftercooler); τ time constants; γ heat capacity ratio; c_p specific heat capacity of the air at constant pressure; hx stands for heat exchangers; in stands for input; p_{am} is a model parameter representing the ambient pressure in bar; and o is used for nominal values expressed in actual units.

The flag $FL_c = \{0, 1\}$ determines if the CAES system is operating in charging mode ($FL_c = 1$) or if the motor is only synchronized with the grid with the compressor load disconnected ($FL_c = 0$). A first-order transfer function of time constant τ_{CA} is proposed to represent the air inertia in the compressor and piping system.

2) *Turbine*: The turbine model is presented in Fig. 5, where the indices b, d, s, R and x stand for burner, inlet, cavern, recuperator, and exhaust, respectively. The flag $FL_t = \{0, 1\}$ allows simulating the CAES system in discharging mode ($FL_t = 1$), or the generator as synchronous condenser when $FL_t = 0$. The minimum limit of 0.01 p.u. in the turbine airflow prevents the model from becoming singular when the air flow is zero during the turbine shut down.

It should be mentioned that the compressor and turbine models used, which are based on those proposed in [7], assume constant mechanical and isentropic efficiencies. These efficiencies are kept constant in order to reduce the level of complexity of the thermodynamic models, as it is usually done for the gas and steam turbine models used in power systems (e.g. [15]). The efficiency values used for the compressor and expander correspond to the most typical operating point, i.e.,

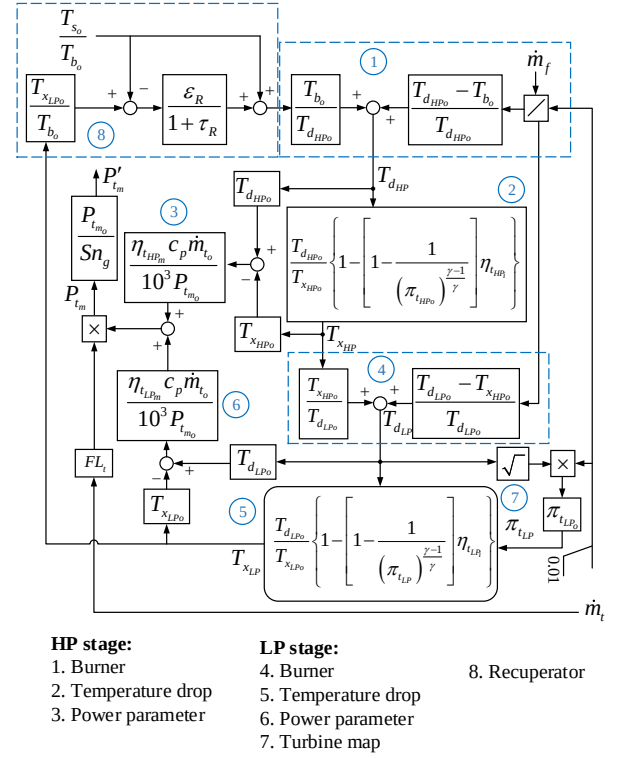


Fig. 5. Turbine model.

nominal conditions.

3) *Cavern*: As in Huntorf [10], the cavern model presented here considers the following pressure limits: $p_{s_{max}}$ and $p_{s_{min}}$ for the maximum and minimum cavern pressures, and $p'_{s_{max}}$ and $p'_{s_{min}}$ for the maximum and minimum “economic” pressures. Between $p_{s_{max}}$ and $p'_{s_{min}}$, the valve should be able to throttle the pressure down to $p_{t_{HPO}}$ when the turbine operates at nominal conditions, thus defining $\Delta p_{min} = p'_{s_{min}} - p_{t_{HPO}}$. The p.u. SoC of the cavern is thus proposed to be calculated as a function of p.u. the cavern pressure p_s , with $p_{s_{max}}$ representing the pressure base in bar, as follows:

$$SoC = \left[1 - \left(\frac{1 - p_s}{1 - p_{s_{min}}/p_{s_{max}}} \right) \right] \quad (4)$$

This equation is based on the principle that the net energy injected in the cavern can be found as the integration of the enthalpy rate coming in and out, which is proportional to the integration of the cavern pressure rate, both equivalent in p.u.

III. CONTROL STRATEGIES

A. SoC Control

The SoC limits the operation of the cavern; thus, when the SoC reaches its maximum SoC_{max} limit, the controller sends the boolean signal $u_c = 0$ to the compressor control to shut it down by bringing the air flow to zero. When the SoC reaches SoC_{min} , the controller sends the signal $u_t = 0$ to the turbine control to close the air valve. Notice that SoC_{max} and SoC_{min} can be chosen by the operator; for example, if $SoC_{min} = SoC(p'_{min}/p_{max})$ and $SoC_{max} = SoC(p'_{max}/p_{max})$, the CAES system would only be allowed

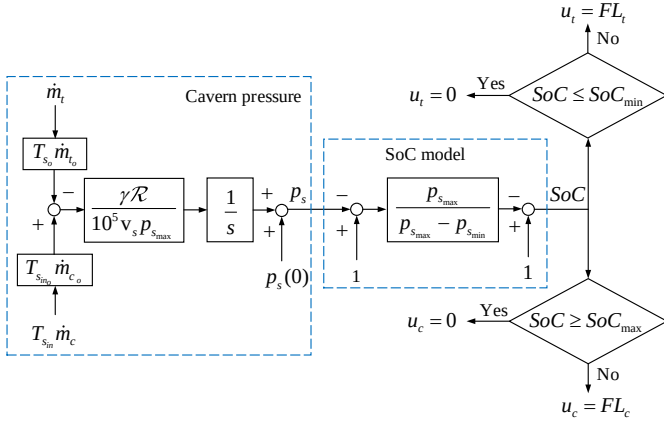


Fig. 6. Proposed SoC control logic.

to operate within its economic range. The control logic block is presented in Fig. 6, where $\mathcal{R}$ is the specific gas constant, v_s is the cavern volume, and $p_s(0)$ is the initial p.u. cavern pressure.

B. Turbine Control

Since the pressure ratio across the discharging valve connecting the cavern to the HP expander is relatively close to unity, the airflow can be calculated using the equation of an incompressible flow through a nozzle [16]. Hence, using the incompressibility approximation and assuming that the turbine can deliver nominal power when the pressure drop across the valve is $\Delta p_{\min}$, the p.u. turbine air flow can be calculated as follows:

$$\dot{m}_t = \lambda \sqrt{\frac{p_s p_{s_{\max}} - \pi_{t_{HP}} \pi_{t_{LP}} p_{am}}{\Delta p_{\min}}} \quad (5)$$

where λ is a dimensionless quantity representing the valve opening, which is zero when the valve is totally closed. The expression (5) is used to map λ values to airflow values; hence, the limits $\lambda_{\max}$ and $\lambda_{\min}$ can be calculated as functions of the limits of the air flow $\dot{m}_{t_{\max}}$ and $\dot{m}_{t_{\min}}$, which ensure a minimum pressure drop across the valve $\Delta p_{\min}$ when the cavern pressure falls below $p'_{s_{\min}}$, thus limiting the inlet HP turbine pressure $p_{in_{HP}}$. Hence, the following valve limits are proposed:

$$\lambda_{\min} = \frac{\dot{m}_{t_{\min}} \sqrt{\Delta p_{\min}}}{\sqrt{p_{s_{\max}} - p_{am} \pi_{t_{LPo}} \pi_{t_{HPo}} \dot{m}_{t_{\min}}}} \quad (6)$$

$$\lambda_{\max} = \min \left\{ \frac{p_s p_{s_{\max}} - \Delta p_{\min}}{p_{am} \pi_{t_{LP}} \pi_{t_{HP}}}, \dot{m}_{t_{\max}} \right\} \quad (7)$$

The control system of the expander is presented in Fig. 7. Notice that since the inlet temperature at the LP expander is kept approximately equal to 1 p.u. by the temperature controller depicted in Fig. 8, the pressure ratio of the LP expander in (3) requires that $\dot{m}_{t_{\min}} > 1/\pi_{t_{LPo}}$ to ensure that $\pi_{t_{LP}} > 1$; this avoids obtaining a less than unity pressure ratio, which means that the air is undesirably being compressed in the turbine rather than expanded. The PI block with gains K_{t_p} and K_{t_i} controls the valve, the dynamics of which are modeled

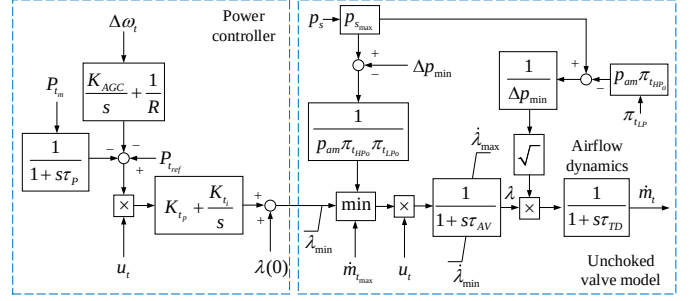


Fig. 7. Proposed turbine air control.

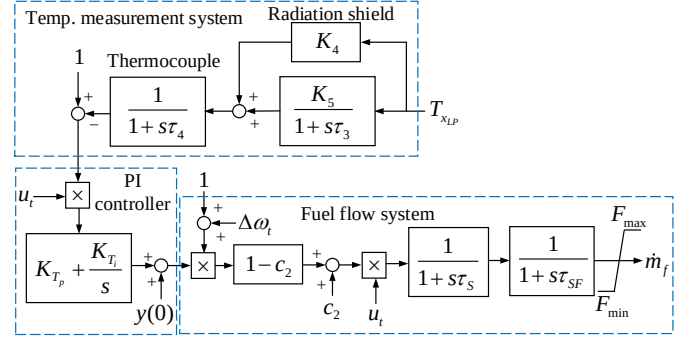


Fig. 8. Turbine temperature control.

using a first-order transfer function with time constant τ_{AV} and valve rate limits $\{\lambda_{\max}, \lambda_{\min}\}$, in order to produce the reference power $P_{t_{ref}}$. A power transducer of time constant τ_p is used to measure the p.u. mechanical power P_{t_m} . A feedback of the speed deviation $\Delta\omega_t$ modifies $P_{t_{ref}}$ to provide primary frequency regulation with constant droop R , and operate in automatic generation control (AGC) mode for the parameter $K_{AGC} \neq 0$. The control boolean signal u_t is used to shut down the expander by bypassing the PI output when $SoC = SoC_{\min}$; this signal also multiplies the input error to the PI to avoid integral windup. A first-order transfer function of time constant τ_{TD} is used to account for delays in the air transportation in the turbine.

The temperature controller, depicted in Fig. 8, regulates the temperature of the expander by comparing the LP exhaust temperature with a reference [7], and the error is processed through a PI block of gains K_{T_p} and K_{T_i} , which controls the fuel flow $\dot{m}_f$. The measurement system comprises a radiation shield (K_4 , K_5 and τ_3) and a thermocouple (τ_4). The fuel flow system is modeled by two first-order transfer functions of time constants τ_s and τ_{SF} , representing the fuel valve and fuel downstream piping and gas distribution manifold, respectively. The c_2 constant represents the fuel flow at no load, and $F_{\max}$ and $F_{\min}$ correspond to the fuel limits. The boolean control signal u_t has also been added here to stop the gas supply when the turbine is shut down.

C. Compressor Control

The compressor control is illustrated in Fig. 9 and is based on [14]. The main control loop realizes the desired compressor power $P_{c_{ref}}$, allowing also primary frequency regulation, as

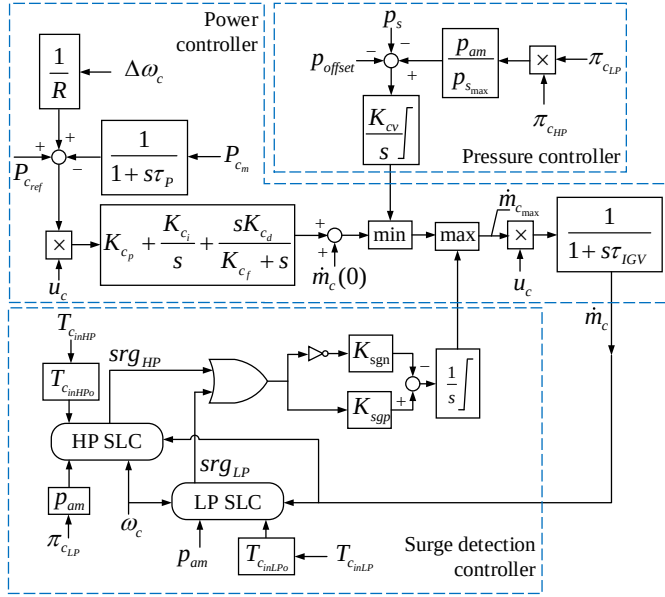


Fig. 9. Compressor control.

the signal $\Delta\omega_c$ is used as feedback. A second controller is used to reduce the compressor airflow when the desired P_{cref} results in an output pressure in the HP stage that is less than the pressure in the cavern; the speed of this controller is defined by the gain K_{cv} . The gain K_{sgp} defines the speed of the anti-surge controller used to increase the airflow when approaching to a surge point, detected by the low-pressure and high-pressure Surge Logic Control (SLC) blocks defined in [14], while K_{sgn} represents the speed of resetting the controller after the surge point has been avoided. The maximum air flow limit is represented by $\dot{m}_{cmax}$, and a new boolean control signal u_c is introduced to simulate the closing of the inlet guide vanes, and also to prevent integral windup in the PID controller with a time delay.

D. Model Initialization

Initialization of the model involves solving its differential equations for a steady-state condition using the power-flow results as the operating point. In some power system packages, this procedure is performed automatically after the power-flow; however, initial conditions for pure integrator blocks might need to be defined externally. In the controls proposed here, the initialization requires the definition of a vector of initial conditions $\mathbf{x}(0) = [p_s(0), \dot{m}_c(0), \lambda(0), y(0)]$ for the regulators, as depicted in Figs. 4 and 7 to 9. The initial pressure of the cavern $p_s(0)$ is independent from the power-flow and is rather a parameter of the model; $\dot{m}_c(0)$ is found by solving the nonlinear system of equations defined by the compressor model in Fig. 4, considering that P'_{cm} is known; and the turbine's controllers ($\lambda(0), y(0)$) are initialized assuming that $T_{XLP} = 1$ and solving the nonlinear system of equations defined by the turbine model in Fig. 5, considering that P'_{tm} is known.

IV. IMPLEMENTATION RESULTS AND COMPARISONS

The models and controls proposed in Sections II and III are used in this section to study and compare the impact on the transient stability of a modified benchmark test system using either a CAES facility or an equivalent GT plant, while considering the effect of the cavern size of the CAES system on the frequency stability of the system.

A. DSATools[®] Model Implementation

The system was simulated in Powertech Lab's Powerflow and Short circuit Assessment Tool (PSAT[®]) and Transient Security Assessment Tool (TSAT[®]), which are used by several utilities and system operators in North America, including the Ontario's Independent Electricity System Operator (IESO) in Canada. The model described in Sections II and III was implemented in TSAT[®] as a fully parametrizable User Define Model (UDM) module, which comprises four sub-models: two governors and two exciters, which interact with two synchronous machines modeled using available machine blocks in TSAT[®]. The excitation system for the CAES machines is the IEEE Type DC1A exciter [17], including the reactive power control described in Section II-C.

The initialization of the controllers of the mechanical systems is performed externally after the power-flow using a Python script, which reads the PSAT[®] power-flow result file and the CAES dynamic parameter file, and then writes the CAES UDM file to set the initial values of the controllers. The turbine and compressor models were implemented using basic fundamental blocks; however, due to the limitations on the maximum number of 100 blocks per UDM sub-model, the CMB blocks, SLC blocks, recuperator, intercoolers, and thermodynamic relations in the compressor and turbine were modeled using special blocks available in the UDM editor known as Dynamically Linked Control Blocks (DLBs), in which the models are coded in C and then compiled into .dll files read by TSAT[®] during the dynamic simulations.

B. Test System

The CAES system model is studied via dynamic simulations when connected in a modified version of the WSCC 9-bus test system from [17]. It is assumed that additional load requires about 350 MVA of additional generation, which is assumed here to be either a set of GT units or a CAES system of equivalent discharging power. The dynamic data of the generators and parameters of the excitation systems were obtained from [17], and for the governors and PSS from [18]. The speed of the valves of the steam and hydro turbines were set to ± 0.2 and ± 0.1 p.u./s, respectively [19], [20]. The model parameters for the GT were obtained from [21], and for the CAES from [7]. A voltage droop control with a gain of 5% was used in the AVRs of the GTs for stable parallel operation.

Seven system configurations were used in the simulations to study the dynamic response of the CAES system model as follows:

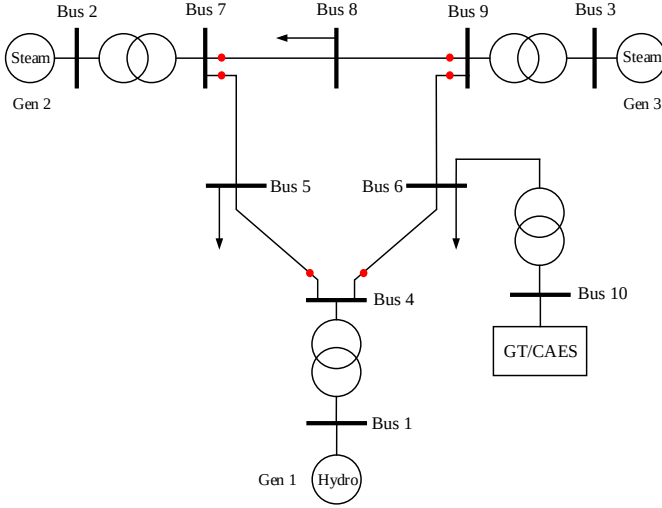


Fig. 10. Benchmark test system.

- Case 1: No GT nor CAES are considered, none of the PSS of the generating units are active, and Gen 1 is on AGC.
- Case 2: Same as Case 1, but the PSS of the three units are active.
- Case 3: Same as Case 2, with four identical 95 MVA GT units connected at Bus 10, which supply a total of 100MW, while reducing the power output of Gen 2 and Gen 3.
- Case 4: The GTs at Bus 10 are replaced by a 341 MVA discharging/75 MVA charging CAES system operating in discharging mode, with its PSS active, and the CAES system on AGC instead of Gen 1.
- Case 5: Same as Case 4, but with the CAES system operating in simultaneous charging and discharging mode.
- Case 6: Same as Case 4, but with two different CAES cavern sizes.
- Case 7: The CAES system at Bus 10 changes the cavern, Gen 1 is on AGC, and all PSS are active.

The aforementioned cases allow illustrating all the possible operating modes and controls alternatives of the proposed CAES system models and controls. Thus, Cases 1 and 2 are necessary to establish a base line, i.e., the system condition without adding any generation or storage at Bus 10. Case 3 is used to study the effects of connecting a GT plant at Bus 10 for comparison with the CAES system performance, which is illustrated in Cases 4 and 5 for two different CAES operating modes: discharging and simultaneous charging and discharging modes, respectively. Finally, Cases 6 and 7 demonstrate the impact of SoC limits associated with the caverns size on the frequency control capability of the CAES system. The PSS gains were tuned by trial and error to achieve acceptable damping in all the cases.

C. Transient Stability Analysis

In this section, the benchmark system's transient stability for the first 5 cases is studied by simulating three-phase short circuits (SC) on all the transmission lines, depicted as red dots

TABLE I
CRITICAL CLEARING TIME (CCT) IN SEC.

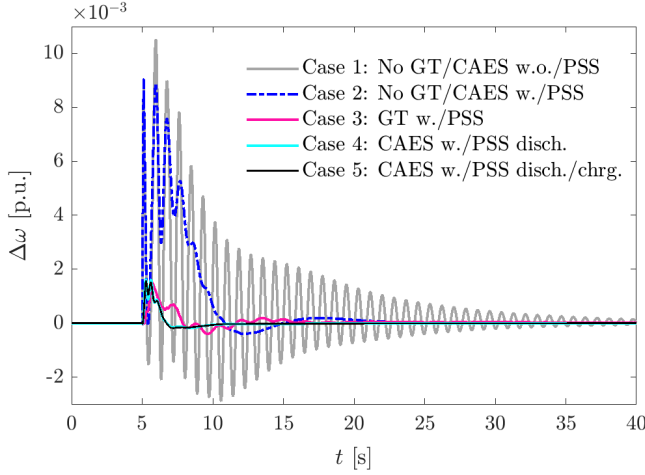
Line faulted	Case 1	Case 2	Case 3	Case 4	Case 5
Line 89	0.194	0.195	0.513	0.536	0.527
Line 96	0	0.164	0.573	0.590	0.573
Line 78	0	0.122	0.242	0.247	0.246
Line 64	0.255	0.256	1.336	1.567	1.471
Line 45	0.247	0.247	2.075	1.782	1.66
Line 57	0	0.101	0.287	0.296	0.293

in Fig. 10. Thus, in Figs. 11, and 12, the terminal voltage and speed deviation of Gen 3, GT and CAES, respectively, for a 100 ms three-phase SC located at 90% of the line from Bus 6 to Bus 4 are presented. The SC is cleared by removing the corresponding transmission line. Observe in Fig. 11 that Case 1 is underdamped; by using Prony analysis on the Gen 1 speed deviation, a 1.12 Hz frequency component with a damping ratio of 1.58% can be found, which justifies the use of PSS in the rest of the case studies. In Case 2, the oscillations are properly damped by the PSS action, but the voltage and speed of Gen 3 take longer to reach steady-state than in Cases 3, 4, and 5. When the GTs are connected in Case 3, a significant improvement in the frequency response is observed and the voltage drop decreases in all generator buses, which is reasonable since new generation has been added to the system; however, the damping is adversely affected despite the recalibration of the PSS, resulting in more sustained oscillations, as observed in Fig. 11. In this case, tuning the PSS is more complex, because more units are equipped with these devices.

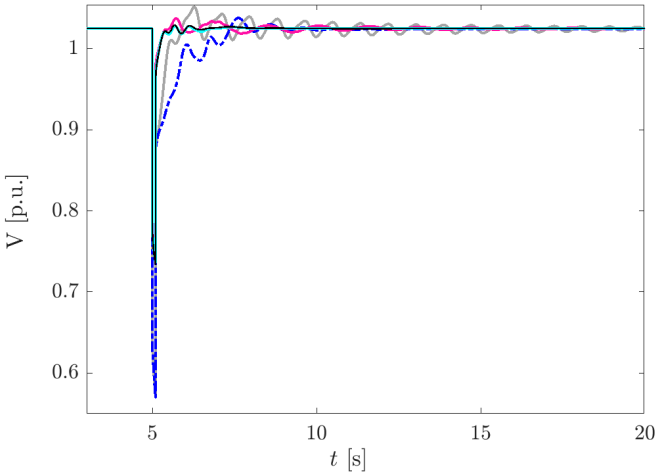
In Cases 4 and 5, the oscillations are damped much faster and the system reaches steady-state in less than 4s, as depicted in Figs. 11 and 12. The short time constants of the actuators that control the airflow (fast-acting air valve and IGVs), and the effectiveness of the PI regulators to realize the reference power in both compressor and expander are the main reasons for the CAES system being able to stabilize the system faster. The main difference between Cases 4 and 5 is that, in the latter, the compressor also contributes to frequency regulation by adding inertia to the system. Note also that although the compressor does not consume electrical power from the grid in Case 4, it contributes to voltage regulation as it operates synchronized with the grid, with the compressor decoupled from the motor's rotor; hence, a reduction in the voltage drop is also observed in Fig. 12b for Case 5.

In order to provide a broader assessment of the impact of the CAES system transient stability of the system, the Critical Clearing Times (CCTs) for SCs located at 90% of all the transmission lines are calculated. In Table I, the CCTs for each contingency and all simulated cases are presented. Observe that for almost all the contingencies, the CCTs improved when the CAES is connected (Cases 4 and 5), except for the SC in line Bus 4-Bus 5 for which Case 3 yielded the largest CCT.

It is important to highlight that most of the proposed controls become active after disturbances, when machines operating conditions change resulting in system oscillations. The pressure ratios of the compressors are particularly sensitive to changes in the airflow and rotor speed due to the maps modeled here; hence, these two quantities may vary

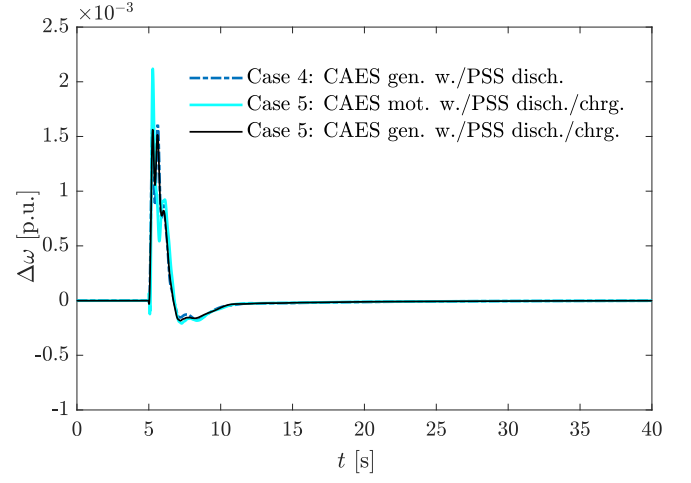


(a)

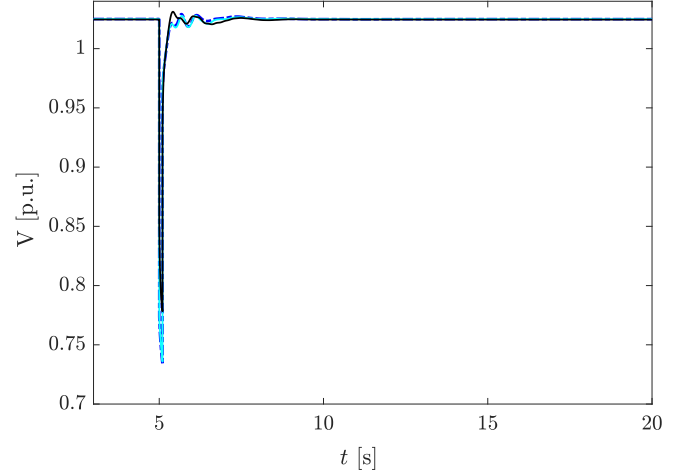


(b)

Fig. 11. Speed deviation and terminal voltage of Gen 3 for SC in line Bus 6-Bus 4.



(a)



(b)

Fig. 12. Speed deviation and terminal voltage of CAES for SC in line Bus 6-Bus 4.

considerably after a large disturbance, thus activating the surge logic control and discharging pressure control. Furthermore, since primary frequency control, which is part of the proposed power controllers, is active during transients, the outputs of the compressor and turbine governors change accordingly, acting on the air valve in the expander and IGV of the compressor to adjust the airflow. Moreover, changes in the airflow directly affect the ratio of gas to air in the burners, thus activating the temperature control also. Hence, considering airflow controls, pressure control limits, and temperature controls during transients improves the accuracy of simulations. On the other hand, SoC control, which affects the energy capacity of the CAES system, is not active during these short-term events, as the time constant of the cavern pressure is very large, thus resulting in an almost constant pressure, especially when a big cavern is considered.

D. Reactive Power Control

An increase of 30% in the system load in Case 4 is simulated to demonstrate the operation of the proposed reactive

power control discussed in Section II-C, which stabilizes and distributes the reactive power between the two CAES synchronous machines. Before the load increase, the CAES system is supplying 100 MW, and since it is operating on AGC, it takes all the new load reaching a new operating point of 192 MW after the event, while the motor is not consuming active power; in this scenario, $Q_{g_{\max}} = 0.826$ p.u., while $Q_{mot_{\max}} = 1$ p.u.

In Fig. 13, the reactive power supplied by the CAES motor and generator using the two proposed values of the parameter β is presented. When no additional control is used in the AVR (β_o), after the load increase, the generator consumes large amounts of reactive power, which has to be supplied by the motor, thus unnecessarily overloading the two machines; low frequency oscillations in the CAES reactive power are also observed. On the other hand, when β_1 is used, the generator, as the machine with the largest unused capacity (MVA), supplies most of the reactive power of 0.039 p.u. or 13.29 MVar, while the motor supplies 0.05 p.u. or 3.75 MVar; in this case, the unused capacity (MVA) prevails in determining the reactive

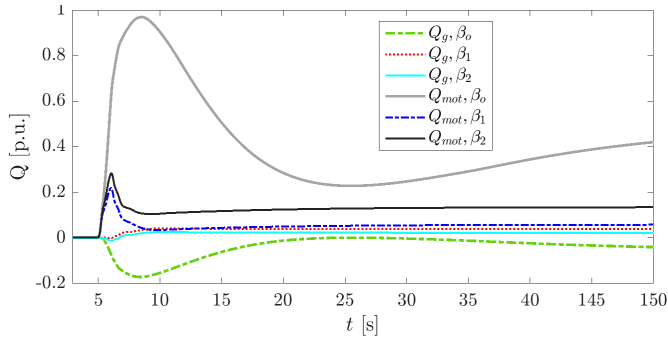


Fig. 13. CAES reactive power for a 30% system load increase and different reactive compensation control strategies in Case 4.

power distribution. When β_2 is used, the contribution of the motor increases considerably to 0.13 p.u. or 9.75 MVAR versus 0.022 p.u. or 7.5 MVAR of the generator, as β_2 improves the loading of the machines. The use of either β results in a more stable reactive power supply and eliminates the reactive power feeding from one machine to the other.

E. Impact of Cavern Size on Frequency Stability

The SoC control proposed in Section III-A is used to study the impact of different cavern sizes on the frequency stability of the system when a CAES facility is connected at Bus 10. For Cases 6 and 7 two cavern sizes are assumed: $v_{s_0} = 10,000 \text{ m}^3$, and $50\%v_{s_0} = 5,000 \text{ m}^3$. In order to demonstrate the SoC control, a maximum economic pressure of 62.5 bar is used to define the maximum SoC to shut down the compressor, i.e., $SoC_{\max} = 0.77$, with $SoC_{\min} = 0$ when the cavern pressure is $p_{s_{\min}} = 30$ bar. The system is perturbed by load changes in the following sequence at all load buses: +30% at 1 min, +5% at 2 min, -10% at 3 min, +5% at 4 min, and -15% at 5 min. The large disturbances on the load allow testing the CAES system under extreme conditions, with load increments indirectly representing generation trips that could lead to frequency instabilities, which are relevant contingencies typically considered in frequency stability studies. The results of the simulations are presented in Figs. 14 and 15.

Observe in Case 6 (Fig. 14) that, since the CAES generator is on AGC, its power follows the load changes. As the cavern pressure p_s decreases from 55 bar, the SoC falls too; thus, the air valve λ opens accordingly to maintain the airflow $\dot{m}_t$ at the required value to regulate the frequency, as shown in Fig. 14a. For the reduced cavern volume of $50\%v_{s_0}$, as soon as the cavern pressure falls below the nominal inlet HP expansion pressure $p_{HP_{in_0}} + \Delta p_{\min}$ bar, the CAES system loses its regulation capability and $p_{HP_{in}}$, which is limited by the airflow control to ensure the minimum pressure drop in the valve, starts following the cavern pressure, as shown in Fig. 14b. In this scenario, P_g is now determined by the cavern pressure, which along with the frequency starts to fall, as shown in Fig. 14c and Fig. 14d. This does not occur when the cavern volume is v_{s_0} because the pressure drop in the valve $\Delta p = p_s - p_{HP_{in}}$ is always larger than $\Delta p_{\min}$, as in Fig. 14b. Note that as soon as the SoC reaches zero at

around 6 min in the $50\%v_{s_0}$ case, the CAES turbine is shut down, forcing $P_g = 0$ (Fig. 14c), thus leading to a sustained frequency drop that cannot be recovered (Fig. 14d); for the v_{s_0} case, the frequency is successfully regulated by the CAES system.

In Case 7 (Fig. 15), the compressor is charging at its full power of 60 MW. Note that when the cavern volume is $50\%v_{s_0}$, the CAES system reaches its maximum SoC (Fig. 15c) at the maximum pressure of 62.5 bar (Fig. 15b), thus, the compressor is shut down by the SoC control at $t = 4.3$ min, which drives its power to zero (Fig. 15c). Given the relatively large load changes and the rate limits used in the hydro unit's governor, which is now on AGC, the frequency deviates more than in Case 6, as demonstrated in Fig. 14d; however, since the compressor represents a load for the system, the hydro unit is capable of bringing the frequency back to nominal after 45 sec. No frequency stability issues were detected in this case; however, this could be due to the small compressor size.

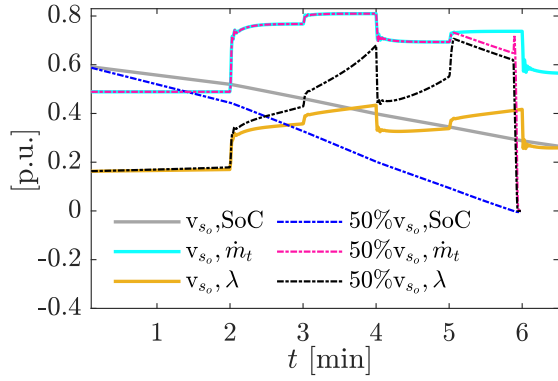
From this analysis, observe that if storage with large power ratings are required for regulation, systems with large energy-to-power ratings such as CAES would be preferable, as these delay the recharging times. This is clearly a disadvantage of other storage technologies such as flywheels, which are able to provide fast regulation, but for a very limited time.

V. CONCLUSION

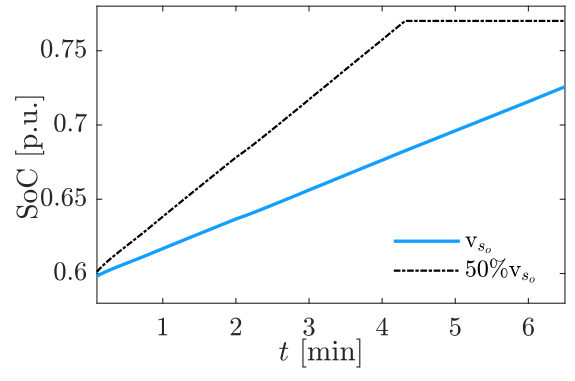
In this paper, the implementation of a block-diagram based transient stability model of a CAES system for power system analysis packages was proposed. A generic model architecture of the different CAES system's components was developed, and the models and controls of the CAES system were discussed. As an application, the model was implemented in TSAT[®], and the impact of a CAES facility on the transient and frequency stability of a modified WSCC 9-bus test system was studied. It was demonstrated that the CAES system improved the transient stability of the system with respect to an equivalent GT connected at the same bus, showing also that the CAES was more effective at regulating the voltage and damping oscillations. The effects of the reactive power control on oscillation damping and voltage regulation coincide with the capabilities of existing generating units, since CAES uses synchronous machines as interfaces with the grid; thus, a special feature the discussed CAES system offers on this regard is the parallel operation of its synchronous motor and generator, which increases the reactive power capacity of the CAES system when one of the machines is not being used to charge/discharge the cavern. However, this parallel operation requires special control for stable operation. Finally, the effect of the SoC and active power controls on the frequency stability for different cavern sizes was investigated, demonstrating that if the power rating of the CAES system is large enough, small cavern sizes may not allow the provision of frequency regulation.

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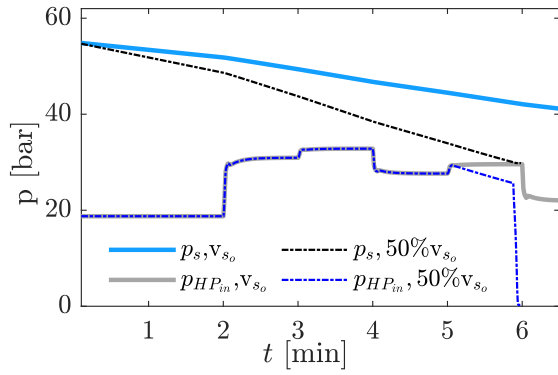
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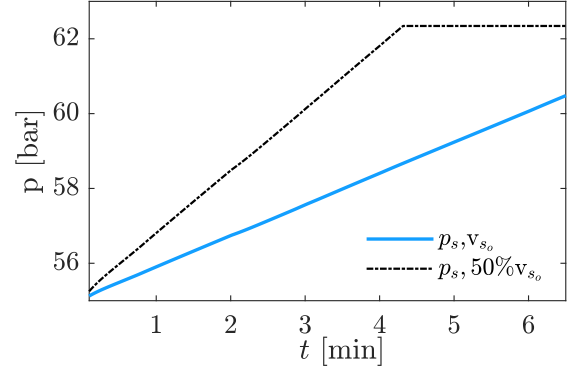
(a)



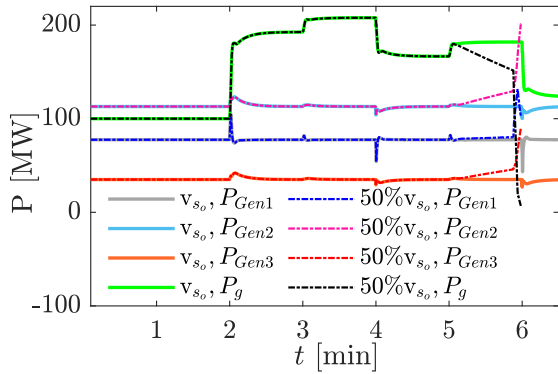
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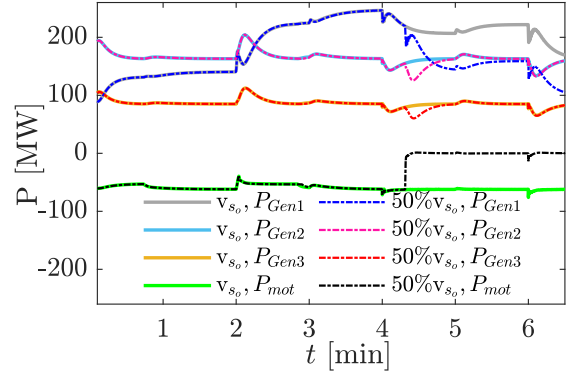
(b)



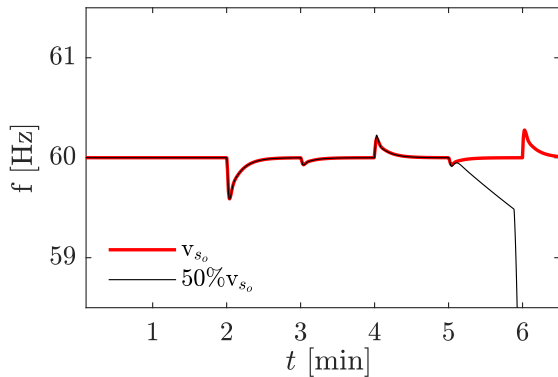
(b)



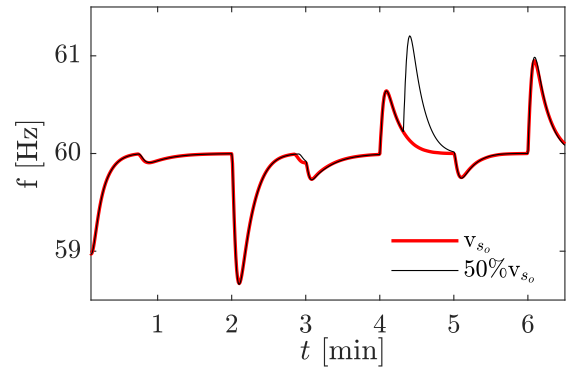
(c)



(c)



(d)



(d)

Fig. 14. System and CAES variables for Case 6, reflecting different cavern volumes.

Fig. 15. System and CAES variables for Case 7, reflecting different cavern volumes.

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