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PERFORMANCE EVALUATION OF GE MS6001B GAS TURBINE BASED ON EXHAUST GAS COMPOSITION AND TURBINE INLET TEMPERATURE

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Abstract

One of the sustainable development goals (SDG) of the United Nation is on climate control which this paper intends to address in the area of exhaust gas emission from a gas turbine plant. The flue gas from the expander contains residual heat and carbon dioxide, which if not properly managed, contribute to energy losses and environmental degradation. This study is essential for improving overall plant efficiency, reducing irreversibilities and identify opportunities for waste heat recovery and carbon capture. The study focuses on performance evaluation of a gas turbine plant by examining key thermodynamic and environmental parameters, including turbine inlet temperature, combustion products, flue gas temperature, mass flowrate and carbon dioxide concentration. The study showed that exhaust composition is governed by air–fuel ratio and essentially independent of pressure ratio. Air–fuel ratio therefore controlled power, emissions and capture potentials. The exhaust gas can be minimised by using technic such as carbon capture. Turbine inlet temperature fell steeply with air–fuel ratio, at roughly 16 K per unit, from 2,245 K to 954 K. The design point for GE MS6001B gas turbine of AF = 61.5 placed the machine at about 1,200 K, the metallurgical limit of the uncooled MS6001B, extracting maximum power consistent with safe blade temperature. It is recommended that waste exhaust heat is recovered rather than rejected in the cooling train, as this is where there is great thermodynamic gain and carbon capture be integrated in the plant to reduce harmful emission.

Keywords:

Exhaust gas, Gas Turbine, Combustion, Turbine inlet Temperature, carbon dioxide emission

1. Introduction

Energy generation, transmission and distribution has been a major challenge to Nigeria's Economic and technological development. Despite having abundant natural resources and various means of power generation the problem seems difficult to tackle. Gas turbine which is widely used for power generation in Nigeria has quick start-up , rapid response to changes in power demand and compatible (Dolor et al., 2026). Research in the area of gas turbine performance is very vital to an improved power generation in the country. There is room for

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great improvement if the right approach and technics of power generation and utilization are developed and implemented.

This work uses ASPEN plus software to evaluate the performance a gas turbine plant with focus on exhaust gas characteristics and recommending better ways to generating and utilizing power with minimal loss.

The need to minimize losses, improved thermal efficiencies, reduce thermal emissions to the atmosphere and increases in prices of energy in recent years has led researchers and industries to develop systems with maximum efficiency and minimum energy losses.

Objectives of the Study

The specific objectives of the study are to:

- develop and validate a steady state process simulation of the Warri Refining and Petrochemical Company (WRPC) GE MS6001B gas turbine power plant using Aspen plus
- analysis of the gas turbine power system across its operating envelope
- characterize the exhaust gas composition, pollutant emissions and flue gas conditions delivered to atmosphere

Gas Turbine Theory

The Gas Turbine plant of WRPC operates by

Continuously drawing in fresh air

Compressing the air to a high pressure

Adding and burning fuel in the compressed air to increase energy level

Directing the high pressure, high temperature air to an expansion turbine, which converts the gas energy into mechanical energy of rotating shaft. The resulting low pressure, low temperature gases are exhausted.

GAS TURBINE MAJOR COMPONENTS

COMPRESSOR

The GT compressor used in WRPC plant is of axial flow design that efficiently compresses a large volume of air. The compressor consist of many individual stages operating in series. Each compressor stage consist of a rotating row of blades that increases the velocity of the incoming air, raising its kinetic energy, followed by stationary rows of blades that act as a diffuser converting the kinetic energy to a pressure rise. The number of stages used for a particular GT compressor depends upon the design pressure ratio for the turbine. WRPC compressor has a pressure ratio of about 11, has 17 stages compression. At the compressor inlet there is a row of stationary blades called inlet guide vanes (IGV) that directs the incoming air to the first rotating stage in a smooth manner. On the MS6001 compressor, the IGV can be changed in flow angle to control the volume of air being drawn into the compressor. The variable IGV are used to achieve aerodynamically smooth operation throughout a large operating range. At the compressor discharge there are many rows of stationary diffuser blades and a cone shaped diffuser to obtain maximum pressure increase before going into the combustion chamber

system. The compressor also serves as a source of cooling and sealing air for the turbine nozzles, turbine wheels, transition pieces and other portions of the gas path as well as bearing.

COMBUSTION CHAMBER

The combustion system consist of several liners into which fuel is added and burnt with a portion of the compressed air. The excess compressed air is used to cool the products of combustion to a temperature level usable by the turbine. The individual liners are connected to the turbine to the turbine section by transition pieces. Fuel is ignited by electric igniters. Once the combustion is started, the combustion process is self- sustaining as long as fuel and air are available .

Thermodynamically, the combustor is the principal site of irreversibility in the gas turbine cycle: exergy analyses of selected Nigerian plants report that between 86.05% and 94.6% of total cycle exergy destruction occurs in the combustion chamber, owing to the large temperature difference between the flame and the working fluid and to the intrinsic irreversibility of the chemical reaction, and show that increasing the TIT reduces this destruction (Oyedepo et al., 2015). Ighodaro and Aburime (2011), in their exergetic appraisal of the Delta IV Power Station, Ughelli — a plant operating in the same Niger Delta gas-supply corridor as the WRPC facility similarly found the combustion chamber to be the most exergy destructive component of the unit. The combustor is also the point of origin of the CO_2 laden flue gas.

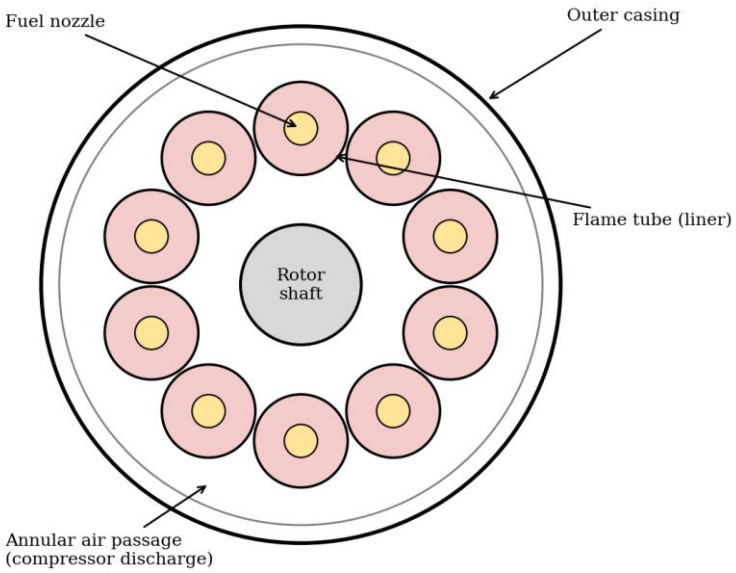

Figure 1: Cross-sectional representation of a can-annular (cannular) combustor arrangement typical of heavy-duty frame gas turbines, showing the flame tubes disposed circumferentially within the annular casing (adapted from descriptions in Boyce, 2012).

The Turbine (Expander)

The turbine consists of several stages. Each stage is comprised of stationary row of nozzles where the high energy gases are increased in velocity and directed towards a high rotating row of buckets where kinetic energy is converted into shaft power. The energy from the combustion system available to drive the turbine is changed by varying the amount of fuel injected into the combustion liners.

The turbine section extracts work from the hot combustion gases through the expansion process 3→4 of Figure 1, expanding them through alternating rows of stationary nozzle guide vanes, which accelerate and direct the flow, and rotating blades (buckets) mounted on the turbine rotor (Boyce, 2012). In single-shaft heavy duty machines such as the MS6001B, the turbine drives both the compressor and the generator on the common shaft shown in Figure 1, so that the net electrical output corresponds to the turbine work less the compressor work and mechanical losses (Cohen et al., 2017). Because cycle efficiency and specific work both rise with TIT, the first stage vanes and blades operate at gas temperatures approaching or exceeding the safe limits of their nickel-based superalloy materials, and are therefore protected by internal convective and film cooling using air bled from the compressor, together with thermal barrier coatings (Boyce, 2012). Energy analyses of Nigerian plants indicate that the turbine, after the combustor, accounts for the next largest share of cycle energy loss between 30.83% and 35.24% across the eleven units evaluated by Oyedepo et al. (2015) underscoring the sensitivity of installed capacity to hot-section condition and blade-path degradation.

(e) Exhaust System

The exhaust system, shown downstream of the turbine in Figure 1, comprises the exhaust diffuser, plenum, ducting, silencers, and stack through which the expanded gases are discharged to the atmosphere in simple cycle operation. The diffuser recovers part of the leaving kinetic energy as static pressure, thereby reducing turbine back-pressure and improving output (Boyce, 2012). In the context of the present study, the exhaust stream is of special significance.

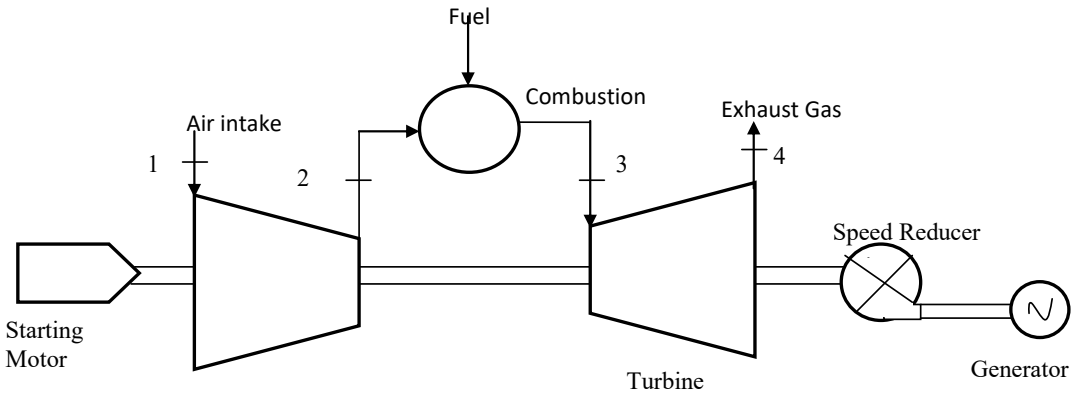

Fig. 1: The Schematic Diagram of Warri Refining and Petrochemical Gas Turbine Plant

Gas Turbine Performance Parameters

The performance of a gas turbine power system is conventionally expressed through a small set of interlocking parameters. The compressor pressure ratio, the ratio of compressor delivery to inlet pressure is the principal cycle design variable; raising it increases the temperature ratio across the cycle but also the compression work. The turbine inlet temperature (TIT) is the thermodynamic ceiling of the cycle, fixed jointly by the AF ratio and the compressor delivery temperature. The thermal efficiency is the ratio of network output to the fuel energy supplied (LHV basis), and the specific work is the network per unit mass of air ingested a measure of how large a machine must be to deliver a given output. The backmwork ratio (BWR), the fraction of turbine work consumed by the compressor, expresses the internal self-consumption of the machine; industrial values commonly lie between 0.4 and 0.5, and the ratio deteriorates

rapidly as pressure ratio rises at fixed firing temperature. The heat rate, the fuel energy per unit of electricity, is the commercial complement of thermal efficiency.

Two conceptual relationships among these parameters organise the parametric results of this study. First, at fixed firing temperature there is a trade-off in pressure ratio: efficiency and specific work do not peak together, and beyond an optimum the growing compression work erodes both. Second, at fixed pressure ratio the firing temperature (set through AF) lifts both efficiency and specific work, which is why the design point of a machine sits as close to the blade-metallurgy limit as cooling technology allows. Studies of installed Nigerian and comparable plants confirm the practical ranges of these parameters thermal efficiencies of roughly 20 to 43 percent and strong sensitivity to ambient and operating conditions (Oyedepo et al., 2015; Aliyu et al., 2020; Jamnani & Kardgar, 2020). The parametric grid of the present study pressure ratios of 5 to 25 crossed with AF ratios of 20 to 100 is designed to map exactly these relationships for the WRPC machine and to locate its design point within them.

Combustion Concepts: Diffusion Flames, Excess Air and Temperature Limits

Combustion in a conventional heavy duty gas turbine combustor is of the non-premixed (diffusion-flame) type: fuel issues from the nozzle into the swirling primary air, and burning occurs where fuel and oxidant meet at near-stoichiometric proportions, producing local flame temperatures approaching the adiabatic flame temperature of the fuel regardless of the overall leanness of the machine. This concept has two important consequences. First, the overall (global) AF ratio controls the mixed out temperature entering the turbine, but not the peak temperature inside the flame; the flame runs hot even in a lean machine. Second, because thermal nitric-oxide formation is an exponential function of peak flame temperature, diffusion-flame combustors produce NO at rates governed by the primary zone rather than by the global mixture the reason NO control in such machines historically relied on water or steam injection and, in modern machines, on lean-premixed combustion systems.

The concept of the metallurgical temperature limit completes the combustion picture. The first stage nozzle and bucket materials of a frame machine of the MS6001B generation tolerate gas temperatures on the order of 1,200 K to 1,300 K without sophisticated internal cooling; the machine's design AF ratio is precisely the dilution required to hold the turbine inlet at this limit. The low air-fuel-ratio (AF = 20) produces firing temperatures far above any admissible metallurgy and is retained only as a thermodynamic limiting case; the practically realisable domain begins at the design AF of 61.5 and extends leaner. This distinction between the physically meaningful and the merely computable regions of the parameter space is maintained throughout the results of the study.

Gas Turbine Emissions

The exhaust of a natural-gas-fired turbine contains, in addition to the nitrogen, oxygen, water vapour and argon of vitiated air, the products and by-products of combustion: carbon dioxide in proportion to the fuel carbon burned; carbon monoxide and unburned hydrogen or hydrocarbons where oxidation is incomplete; and nitrogen oxides formed predominantly by the thermal mechanism in the hottest regions of the flame. Conceptually the pollutant set divides into two families with opposite temperature behaviour: the products of incomplete combustion (CO, UHC, H_2), which are suppressed by high temperature and abundant oxygen, and thermal NO, which is promoted by exactly those conditions. Lean operation therefore drives CO and UHC to negligible levels while progressively reducing NO as flame dilution lowers peak

temperatures the pattern quantitatively reproduced in the results of this study, where CO and H_2 vanish for AF of 61.5 and above while NO falls steeply with leaning.

Two emission accounting concepts frame the environmental analysis. The carbon intensity of generation kilograms of CO_2 emitted per megawatt-hour of electricity permits comparison of the plant with and without capture on a service basis rather than a fuel basis. The distinction between CO_2 captured and CO_2 avoided recognises that the energy penalty of capture requires additional fuel (or displaced electricity) to restore the lost output, and that this make-up energy carries its own emissions; the net avoided quantity is accordingly smaller than the gross captured quantity (Kamolov et al., 2024). Both concepts are applied in the environmental results of this study. A third, subtler concept links emissions to the capture plant itself: flue-gas NOx and oxygen attack the amine solvent, forming heat-stable salts and oxidative degradation products, so the combustion emissions of the host machine are not merely an environmental output but an operational input to the capture system (Buvik et al., 2021).

2.0 MATERIAL AND METHOD

Description of the Flow Process

The complete process flowsheet, as implemented in Aspen Plus, is shown in Figure 2 Atmospheric air enters the compressor (COMPRESR) and is compressed to the combustor pressure. The compressed air (CMPOUT) is mixed with natural-gas fuel (FUELGAS) in the combustion chamber (CC-BLK), which is modelled as a Gibbs free-energy reactor so that the combustion products are obtained from chemical-equilibrium considerations. The hot combustion gases (TURBINE) are expanded through the turbine (TURB-BLK) to near-atmospheric pressure, producing shaft work and yielding the flue gas (FLUEGAS)

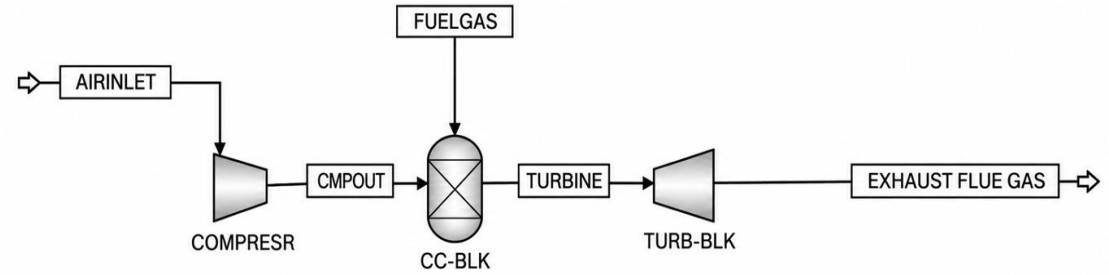

Fig 2.1: Aspen Plus process flowsheet of the gas-turbine power plant

The principal modelling assumptions adopted in the study are summarised below.

(i) The process operates at steady state with negligible heat losses to the surroundings except where a duty is explicitly specified.

(ii) The air supplied to the compressor is dry and of fixed composition, taken as 0.03 mol% CO_2, 20.95 mol% O_2, 78.09 mol% N_2 and 0.935 mol% argon.

(iii) Combustion in the combustion chamber attains chemical equilibrium, modelled by minimisation of the Gibbs free energy.

(iv) The compressor and turbine are modelled as adiabatic machines with constant isentropic and mechanical efficiencies.

RESULT AND ANALYSIS

Effect of Air-Fuel Ratio on Turbine-Inlet Temperature

Figure 3 presents the turbine-inlet temperature as a function of air-fuel ratio. It falls steeply from about 2,245 K at AF = 20 to about 954 K at AF = 100 (at pressure ratio 5). This is the fundamental mechanism by which the AF ratio controls performance and emissions simultaneously, diluting the combustion products with excess air lowers the peak temperature and thereby reduces both the available enthalpy drop (reducing specific power) and the thermal-NO formation rate (reducing emissions). The temperature at AF = 20 exceeds 2,200 K, which is far above the blade-temperature capability of modern cooled turbines (~1,650 K) and represents a physically unrealisable operating condition. The leaner cases, from AF = 61.5 upward, bring the temperature into the range consistent with uncooled industrial machines of the MS6001B class and are therefore the physically realisable operating region of the study.

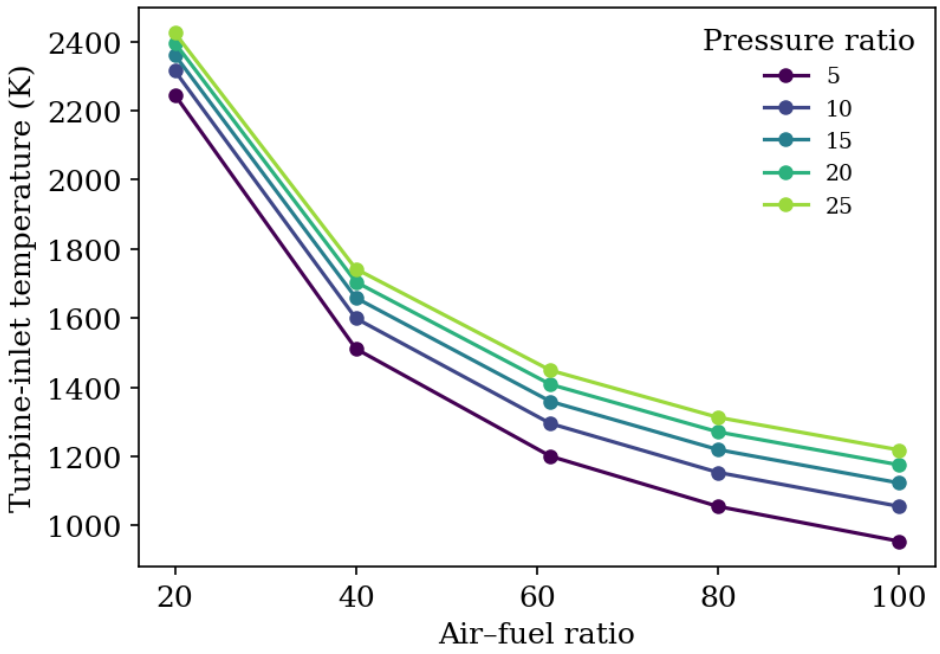

Figure 2.2. Turbine-inlet temperature as a function of air-fuel ratio at a pressure ratio of 5.

Exhaust Gas Composition and Emissions

Major Species

The exhaust composition is governed almost entirely by the air-fuel ratio and is essentially independent of pressure ratio. Table 2.1. presents the mole fractions of the five major species averaged over pressure ratio. As the mixture is leaned from AF = 20 to AF = 100, the CO_2 mole fraction falls from 7.90% to 1.77% and the H_2O fraction falls from 16.80% to 3.63% as the combustion products are progressively diluted in the larger excess-air stream. Simultaneously, the O_2 fraction rises from 2.46% to 17.01% as the fraction of air not consumed in combustion grows. N_2 and Ar remain nearly constant.

Table 2.1. Exhaust gas major-species mole fractions (averaged over pressure ratio).

AF	N_2	O_2	CO_2	H_2O	Ar
20	0.7106	0.0246	0.0790	0.1680	0.0085
40	0.7455	0.1127	0.0427	0.0883	0.0089
61.5	0.7578	0.1459	0.0283	0.0584	0.0090
80	0.7631	0.1604	0.0220	0.0452	0.0091
100	0.7667	0.1701	0.0177	0.0363	0.0091

Pollutant Emissions

Table 2.2. presents the pollutant species (CO, NO and H_2) in parts per million by mole, averaged over pressure ratio. The results illustrate the competing effects of temperature and oxygen availability on combustion products. At AF = 20 the very high flame temperature drives extensive thermal NO formation (5,990 ppm) and leaves a significant CO residual (2,457 ppm) and unburned H_2 (919 ppm), reflecting the equilibrium shift toward partial-combustion products at extreme temperatures. As the mixture is leaned, the excess oxygen completes the oxidation of CO and H_2 very rapidly; both are below 2 ppm at AF = 40 and essentially zero beyond. The NO falls steeply with decreasing flame temperature, from 5,990 ppm at AF = 20 to 111 ppm at AF = 100. These trends are qualitatively consistent with published data for diffusion-flame industrial combustors. The AF = 20 figures are physically unrealisable and the practically relevant range begins at AF ≥ 61.5.

Table 2.2 Exhaust pollutant species in ppm by mole (averaged over pressure ratio; qualitative trends only).

AF	CO (ppm)	NO (ppm)	H_2 (ppm)
20	2,457	5,990	919
40	1.9	1,833	1.2
61.5	0.02	506	0.02
80	<0.01	221	<0.01
100	<0.01	111	<0.01

CONCLUSION

A single steady-state flowsheet was built in Aspen Plus that couples the gas-turbine. The power system used a Gibbs-energy combustor and isentropic compression and expansion blocks

calibrated to the manufacturer rating of the Warri Refining and Petrochemical Company MS6001B machine. The model was driven by acquired plant data.

Turbine inlet temperature is a function of Air Fuel ratio and this is the fundamental mechanism by which the AF ratio controls performance and emissions simultaneously, diluting the combustion products with excess air lowers the peak temperature and thereby reduces both the available enthalpy drop (reducing specific power) and the thermal-NO formation rate (reducing emissions)

GT still present significant environmental challenges due to their carbon output. One of the most pressing issues is the scale of cumulative emissions. While individual GT units may have relatively lower emissions, the widespread deployment of natural gas plants means that total carbon output remains high. This is particularly problematic in regions with heavy reliance on GTs for electricity production in Nigeria and the world at large.

Exhaust composition was governed almost entirely by air–fuel ratio and was essentially independent of pressure ratio: Air–fuel ratio therefore controlled power, emissions and capture duty through the single mechanism of dilution.

The turbine of the frame machine expands the combustion products through a small number of high loading stages, each extracting a large enthalpy drop; the first-stage nozzle, which faces the full firing temperature, is the life-limiting component of the entire plant. The concept of blade cooling explains the historical trajectory of gas-turbine performance: by bleeding compressor air through internal passages and film-cooling holes in the hot-section aerofoils, modern machines sustain gas temperatures well above the melting point of the blade alloys, and every increment of admissible firing temperature purchases cycle efficiency and specific work

Turbine inlet temperature fell steeply with air–fuel ratio, at roughly 16 K per unit, from 2,245 K to 954 K. The design point of AF = 61.5 placed the machine at about 1,200 K, the metallurgical limit of the uncooled MS6001B, extracting maximum power consistent with safe blade temperature.

It is recommended that waste exhaust heat is recovered rather than rejected in the cooling train, as this is where there is great thermodynamic gain and carbon capture be integrated in the plant to reduce harmful emission.

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