SOME ASPECTS ON WET COMPRESSION – PHYSICAL EFFECTS AND MODELING STRATEGIES USED IN ENGINE PERFORMANCE TOOLS

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ABSTRACT

As the first component following the air intake, the gas turbine's compression system can encounter a significant amount of liquid water in the gas path flow during operation, affecting its thermodynamics, aerodynamics and mechanics in various ways. In order to estimate the impact of liquid water on the system behavior prior or parallel to engine testing, modeling capability for such droplet-laden flows is crucial for development tools in gas turbine industry. This paper presents a summary of the most important facts concerning compression of liquid-water two-phase flow in aircraft and stationary gas turbine engines. As an introduction to the topic some general information on phenomena involving wet compression are provided, followed by a description of the physical effects and interactions due to the presence of a liquid water phase in the compressor geometry. In a next step, the state-of-the-art modeling strategies for wet compression with respect to engine performance calculations are outlined, categorized and evaluated concerning their validity. The latter takes into account a comprehensive collection of experimental data available from scientific literature, that has formed the basis for model validation in the past. The advantages and limitations of the different approaches are discussed and ideas for further improvement of modeling are presented.

1. INTRODUCTION

Modular engine performance tools simulate the thermodynamic cycle of aircraft engines and stationary gas turbines using code-based component modules, which are linked to each other according to the specific user-defined engine configuration. The use of engine performance tools is essential in all engine life cycle phases for the reliable prediction of engine behavior at varying operating conditions, allowing evaluation of engine performance data and safe operation. The component modules of a performance tool, such as for example air intake modules, compressor or combustor modules, contain the physics of the 'real' component implemented in the form of characteristics, turbo-component maps or mathematical expressions. First engine performance tools were developed for calculating the thermodynamic engine cycle based on gaseous working fluids only, so that deviations in performance output, modeling gaps respectively, may come up, in case the engine performance is evaluated for an ingested droplet-laden gas flow. The phenomena relevant to gas turbine industry that involve the compression of two-phase flow consisting of liquid water and humid air are outlined in the following section. This paper does not consider two-phase flows carrying a solid water phase, e.g. in the form of (high-altitude) ice-crystals, griesel, graupel, hail or snow. It is emphasized that this text focuses on gas turbine engine systems with axial flow paths. Due to the strong dependency of two-phase flow effects on the specific engine geometry, system behavior of an engine with radial flow paths will differ significantly from the one of an engine with mainly axial gas path flow.

FIG 1. Condensing steam from a catapult enveloping an F-14 Tomcat on the flight deck of an aircraft carrier [a]

2. PHENOMENA AND GAS TURBINE APPLICATIONS INVOLVING COMPRESSION OF LIQUID-GAS TWO-PHASE FLOW

2.1. Water Ingestion

During operation in flight aero engines are continuously interacting with the specific environmental conditions surrounding them. Apart from variable ambient pressure, temperature and relative humidity, an engine may encounter a significant amount of liquid water in its ingested gas stream. The in-flight ingestion of liquid water may be the result of flying through clouds, heavy rain or thunderstorms [1]. In particular at low engine speeds and high aircraft velocities, a combination present during descent and landing, concentration of liquid water in the gas path flow may be hazardously increased due to contraction of the aero engine air capture stream tube relative to the entry plane of incoming liquid water droplets. This phenomenon is called scoop-effect. Liquid water may as well be ingested by an aircraft engine from tire-generated runway sprays or splash water from ground puddles while for example taxiing or accelerating for take-off [2].

Marine gas turbine engines may be affected by water ingestion due to their field of application. For instance gas turbine engines on ship-to-shore-connectors (hovercrafts) may ingest certain amounts of water when operating in inclement weather and at heavy sea conditions. Of course engine air intake design will be done as to prevent critical amounts of water, though complete avoidance of ingestion may not be achieved at all operating conditions. This statement does also apply to helicopter engines running during storm or when hovering over splashing, wavy water surfaces [3].

On aircraft-carriers flight vehicles are often accelerated to take-off speed by high-pressure steam catapults, which are integrated into the carrier flight deck. Worn catapult sealings may lead to loss of hot steam into the environment. If ambient temperatures are low, the leaking hot steam may condense, spontaneously forming small water droplets that can be ingested by an aircraft engine (see FIG 1.) [4]. A similar, important form of interaction with a liquid water phase *may* result from intake condensation during engine testing. Engines tested in a sea-level static or altitude test facility suck in clime, humid ambient air at given initial stagnation pressure and temperature level. Due to fluid acceleration, static pressure and temperature will drop along the distance to the engine entry section, so that the saturation line of humid air may be crossed. Thus, one should note that saturation pressure depends on static temperature exponentially. Essential for spontaneous, heterogeneous condensation is the presence of a sufficient number of condensation-initiating nuclei in the flow region of low static temperatures and pressures, crossing the saturation line. Such nuclei may e.g. be dust, pollen or soot particles from combustion. The time-dependent condensation process in the air intake section is accompanied by a local increase of total (and correspondingly static) temperature due to release of latent heat of evaporation. This may have an impact on tested engine performance and discrepancies between measured test input and input ‘seen’ at the engine inlet. At certain boundary conditions fluid acceleration by a tested engine may also lead to formation of tiny ice-crystals at the engine inlet section (icing). Intake condensation may potentially occur during engine acceleration while starting on ground level. Intake condensation is not a problem for flying at high altitudes, high-velocity flight conditions respectively, since here relative humidity tends to be low and static pressure and temperature do increase due to the intake ram effect [5-9]. Generally a liquid water phase entering the engine is disintegrated and evaporating while going through the engine’s intake geometry and compression system. Liquid water not being evaporated in the compression system in front of the pre-combustor diffuser may reach the combustion chamber, where it leads to local temperature reduction, affected reaction kinetics, local extinction, deteriorated fuel burn-out and increased pressure losses. If the ingested quantity of liquid water reaches a critical level, combustor flame-out and engine run-down may occur.

2.2. Water Injection

A well-known method for performance enhancement in stationary gas turbine and aero engine applications is the specific injection of liquid water in the air intake system for the purpose of working fluid inter-cooling [10]. The injected, best possibly homogeneously distributed, small-sized water droplets are intended to evaporate while passing through the compression system, thus reducing the average and exit temperature level in the compressor component. Since water has a comparatively high latent heat of evaporation, the transformation of water mass from liquid into gas state absorbs much heat energy from the hot gas phase surrounding the injected water droplets. The inter-cooling by droplet evaporation minimizes the specific work of the compression process. On the other hand, in a stationary gas turbine operating at constant maximum turbine inlet temperature, the compressor exit temperature reduction by inter-cooling allows an increased injection of fuel into the combustion chamber. Evaporated liquid water and added fuel lead to an increased gas mass flow through the turbine component with increased specific heat capacity and thus more system power output at improved thermal efficiency. An alternative to water injection in front of the compressor is the direct water injection into the combustor. Combustor water injection is less efficient than injection at the compressor inlet section and introduces some difficulty concerning engine operational stability due to combustor vibrations from discontinuous combustion and thermal shocks from instantaneous contact of hot combustor walls with cold injected water flow.

The principal idea of specific water injection into gas turbine compression systems for system power enhancement was already developed in the early years of gas turbine technology. Though, initially the application of the method was restricted entirely or used short-term due to a lack of adequate liquid phase injection technology at that time. Droplets and sheets of liquid phase exceeding a certain size limit may lead to inacceptable erosion on rotating blades and in particular the compressor’s front stages. With the developments in military aviation in the 1940ies advanced work was performed on water injection systems and corresponding testing for aero engine applications (‘wet thrust’) [11-14]. Compressor water injection (or alternatively combustor water injection) were intended to help generate temporarily increased thrust during aircraft starting at high altitude airfields and/or high ambient temperatures. For injection often a mixture of methanol and de-mineralized water was used. Mixing liquid water with methanol was intended to both provide a water tank anti-freezing agent for operation after low ambient temperature periods and to let the methanol perform as added fuel input to the combustor after evaporation. To avoid deleterious deposits and corrosive effects on compressor blades, merely distilled water with low mineral content was used. Note that this statement applies also to today's applications for gas turbine water injection.

Coming from military aircraft applications, the principle of temporary water injection was transferred to civil aviation beginning in the 1950ies and accompanied by further technological improvements. One should note that the engine water injection systems necessitated additional installation space and additional weight. The enormous developments concerning turbofan aero engines and the associated added power reserves made a further use of methanol-water injection dispensable in the early 1970ies.

Concerning the technology of water injection in the engine air intake system or at the compressor inlet section, the inserted liquid water mass flow must be controlled so that the performance of the combustion chamber is not deleteriously affected. As mentioned above, an alternative to water injection into the intake section or at the compressor inlet is the direct water injection into the engine's combustor. In this application it is not made use of a compressor inter-cooling effect, but of a local temperature reduction in the flame zone, enabling increased fuel input before reaching material limits and achieving a reduction of emissions. Besides the installation of water injection nozzles in the air intake system or at the compressor inlet, some nozzle installation configurations perform a direct injection between individual compressor stages. These configurations are of minor practical interest, for instance due to operability and maintenance aspects.

Further progress in the development of liquid phase injection technology was achieved in several industrial branches in the 1970ies to 1990ies. Because of the developments on the worldwide energy market till the mid-1990ies and associated rising energy costs, the principle of specific water injection for power boosting was 're-discovered' for stationary gas turbine applications [15-24]. Research efforts then increased for cost-effective long-term liquid water injection systems for use in stationary gas turbine entry sections. Yet ongoing nozzle (skid) development work has been focusing on best-possibly small generated droplet sizes and clearly defined injection rates of the liquid phase for efficient evaporation along the compressor flow path. Small droplet sizes are intended to avoid significant long-term erosion effects and keep maintenance costs low. As in aero engine applications, water injection in stationary gas turbines is aimed at increasing system output power in particular at high ambient temperatures. Since on the one hand more new power plants are needed in emerging markets, where climate will differ from the one in mid-temperature zones, and periods of high ambient temperatures may often be coupled with phases of highest power demands, advanced water injection technology for stationary gas turbine engines can generate significant benefits. Thus, starting around year 2000 the research work concerning water injection for stationary gas turbine applications rose drastically, as can be seen taking into account the large number of scientific publications on the topic, that are coming up until today. For water injection principles in gas turbine industry also the terms 'inlet fogging', 'inlet cooling', 'evaporative compressor cooling', 'overspray', 'high fogging' and 'low fogging', 'wet steam injection' or 'wet compression' are used to describe both the position of water injection in the gas turbine flow path and the extent of added liquid water mass flow. The use of these terms is not uniform, but it can be stated that 'evaporative compressor cooling', 'overspray', 'high fogging', 'wet steam injection' and 'wet compression' are related to a cycle process in which yet un-evaporated liquid water droplets, injected in the intake section or the compressor inlet, are present in the compressor gas stream. The most common term for the description of liquid water-gas two-phase flow in the field of gas turbine compression is 'wet compression'. The terms 'inlet fogging', 'inlet cooling' and 'low fogging' do more refer to applications in which injected liquid water disintegrates and fully evaporates prior to reaching the compressor inlet plane for the purpose of inlet air temperature reduction. Several literature sources emphasize the future potential of optimized water injection for long-term stationary gas turbine power boosting in different engine system configurations.

2.3. Wet Gas Compression

Oil and gas industry rigs may be located off-shore for drilling and transporting natural gas for energy conversion on-shore. The gas transportation from sea to land is done with long pipelines equipped with in-line compressors. Since natural gas gained from deep-sea sites is not completely dry, but interspersed with certain amounts of other hydrocarbons in liquid state, these compressors face a so-called 'wet gas' two-phase flow in operation. The latent heat of evaporation of liquid hydrocarbons is some orders of magnitude smaller compared to the one of liquid water, so that the inter-cooling effect due to evaporation during compression is less dominant. Though, the presence of a liquid phase may affect the compressor performance, so that the physical phenomena related to 'wet gas' operation should be considered in component development [25,33].

3. IMPACTS OF A LIQUID PHASE ADDED TO GAS PATH FLOW IN A THERMODYNAMIC ENGINE CYCLE

In this section the global effects of a liquid water phase on the engine components air intake, compression system and combustor as well as on overall engine performance shall be outlined. Basically, it can be stated that the operation of a gas turbine with specific water injection, using a well-defined liquid mass flow rate and nozzle configuration, differs from operation of a gas turbine system with rain-like water ingestion. Water injection aims at section-wise homogenized two-phase flow with an appropriate small-droplet-size distribution for optimization of the evaporation process and thus most effective evaporative inter-cooling along the compressor path [26-29]. Small water droplets do closer follow the gas stream passing through the compressor stages than larger droplets, which have a higher (velocity) slip. The smaller the transported droplets the less is the probability of being collected on rotating blades. In contrast to that, aero engines in inclement weather operation predominantly ingest inhomogeneously distributed, large-size droplets, which may reach the inlet plane of the compressor. The inertia of large-size droplets and the corresponding velocity slip lead to increased droplet collection efficiency, meaning enhanced film formation on blades and on the compressor casing due to centrifugation, splashing phenomena and ligament formation at rotating blade trailing edges [30-41]. All these phenomena are coupled with some sort of aero-mechanical losses. Resulting from the very short characteristic residence times of transported fluid in high-speed compressors, it is possible that some large-sized droplets and un-evaporated wall films may enter the pre-combustor diffuser or the combustion chamber.

In stationary gas turbine applications using specific **water injection** in front of the compressor, **high-fogging** skids respectively, with liquid water-to-dry air mass ratios normally being between 1% and 2% and droplet diameters typically in the range of micrometers, the following global effects on system performance have both been observed in water injection experiments and described using analytical methods:

1) The evaporation of the liquid water phase along the compressor flow path leads to an inter-cooling effect. The compressor exit gas temperature is reduced, the gaseous phase mass flow into the combustor does increase. The continuous evaporation leads to local density and thus flow velocity changes affecting compressor stage matching [38,43,44,51,60].

2) As a result of the inter-cooling effect the specific work of the compression process is reduced [52-63]. Assuming constant compressor efficiency for 'wet' operation, this means that based on the expression for specific fluid work, which is related to Gibbs' formulation, the pressure level at compressor exit is raised for a given enthalpy input [10,91]. If water injection is performed in combination with increased fuel injection to achieve a constant turbine inlet temperature, the power output of the gas turbine system is boosted at higher thermal system efficiency [10,19,38,50]. The boosting results from the facts that mass flow through the turbine will increase and that the specific heat capacity of the gas stream in the turbine section will be raised by a higher mass fraction of vapor and combustion gas in the flow path. The system's thermal efficiency is positively affected since compressor work is reduced relatively to turbine work by inter-cooling. Compressor power and turbine power both increase in absolute terms due to the increase in gas mass flow transported through the engine. Mechanical design constrains must be accounted for.

3) Due to evaporation a work load shift in the compressor to the rear stages can be observed, as the overall compressor pressure ratio increases at given work input. Beginning at the compressor inlet section, liquid phase is heated up by compression. In the rear section of the compressor gas flow density increases significantly due to temperature reduction by evaporation of liquid phase, diminishing the local axial flow velocity. Taking into account the typical form of a stage map, this means that the rear stages are throttled. The rear stages' pressure ratio is increased in comparison to operation at 'dry' conditions, while the compressor's front stages are unloaded [42-51]. The mass flow of ambient air being transported by the gas turbine engine does increase, since at high-power setting a *choked turbine* operating at an increased pressure level does demand for an increased mass flow to keep reduced mass flow constant [10,38,50,52,60]. The compressor surge line is merely insignificantly affected by water injection, if quantities of water are *low* [34,67]. A comprehensive collection of information and analyses concerning compression system stability with two-phase flow is, to the knowledge of the authors, not available in scientific literature. Inter-cooling by evaporating two-phase flow in the compression system produces an effect similar to going from one line of constant standard-reduced speed to a line of higher standard-reduced speed in the compressor map. By liquid phase change in the flow, compressor inlet temperature is *fictitiously reduced*. Correspondingly, this means higher reduced mass flow and higher compressor pressure ratio. At fixed turbine inlet temperature, gas turbine rotational speed does increase, since a larger mass flow must be transported through the engine. 'Wet' operation shifts the working line in the compressor map further toward the surge line, diminishing compressor surge margin.

4) The larger the injected water droplets the more pronounced is the erosion threat with respect to long-term 'wet' operation, in particular in the front compressor stages. The smaller the droplets in the generated water spray are intended to be, the more nozzle pressure is needed. Higher nozzle pressure is coupled with higher system investment and maintenance costs [23,29,38,64-66].

5) The aerodynamic performance of compressor blades exposed to two-phase flow, with low liquid water loading typical for gas turbine fogging, is not severely affected. Virtual blade profile thickening due to formation of liquid water films on blade geometries is negligible [31,34,67,97]. Flow angle deviation and stage mismatching result dominatingly from evaporation and the associated density change. Since injected droplets are intended to be rather small, they will closely follow the main gas stream and thus keep *added aerodynamic shear losses low*. Apart from splashing of high-speed droplets on rotating compressor blades, which results in a shift of the droplet size spectrum to even smaller diameters by droplet impact break-up, mechanical interaction between the liquid water phase and the compressor geometry is of secondary interest. It is the aim of specific water injection to make use of the thermodynamic benefits of inter-cooling, while keeping the counteracting aerodynamic and mechanical losses at a minimum. The injected quantity of liquid water is controlled as such that no significant amount of liquid phase passes into the combustion chamber [1].

6) Liquid water injection for gas turbine engines can also be performed directly into the combustion chamber to cool down the primary flame zone and to allow increased fuel injection, depending on the specific engine power setting [50,68]. The raised gas mass flow and specific heat capacity through the turbine boost the work output of the gas turbine system.

Below the global impacts for gas turbine operation involving **ingestion of large amounts of liquid water** with large-size droplet spectra are indicated as known from scientific literature. Though, 'large' amounts of liquid phase stand for injected liquid water-to-dry air-ratios by mass far above 2% and 'large' droplets are defined as those being in the range of millimeters and above. The impacts of water ingestion are of particular interest concerning aero engine applications in inclement weather. Generally speaking, the ingestion of large amounts of liquid water into the engine inlet has a deleterious effect on compression system and overall engine performance. Since the interaction of a specific engine with a liquid phase depends on its design, size and power setting, the use of the terms 'small and large droplet diameters' and 'large and small quantities of ingested water' may vary in engineering practice depending on the specific engine type. The statements that were developed for water injection on the previous page can principally be applied to water ingestion cases, too. However, while for *water injection* at the compressor inlet the *thermodynamic benefits* from inter-cooling are intended to overcome the aero-mechanical losses from two-phase flow effects, at a specific quantity and form of liquid water ingested the induced aero-mechanical losses begin to counteract the positive thermodynamic effects significantly. **Water ingestion and water injection differ in terms of weighting of thermodynamic benefits versus aerodynamic and mechanical losses**. The global impacts of ingested 'large' quantities of liquid water, large-diameter droplet-laden flows respectively, are stated as follows:

1) A liquid phase entering the engine's compression system will partially evaporate, thus reducing compressor exit temperature.

2) The aerodynamic performance of the compression system is significantly affected by the liquid phase, when quantities are 'too large'. Droplets impinging on compressor blades may then form a liquid film on them, virtually thickening and altering the blade profiles [69-72]. The blade film will eventually move toward the casing due to centrifugal force and toward the blade trailing edge in axial direction due to shear forces, possibly separating from it in form of a wavy water film ligament [32,35]. The liquid film crossing the blade tip section toward the casing may partially disintegrate as a 'liquid jet in cross-flow'. The liquid film moving along the compressor on the casing wall will block the gas flow over the rotor tips, increasing local shear forces and turbulence, but though counteracting tip clearance flow. Aerodynamic stability may be significantly negatively affected, corresponding to the flow pattern of ingested liquid phase [30,31,34].

3) Taking a look into a compressor map, the ingestion of a large quantity of liquid water has a deleterious effect on compressor performance comparable to going from a higher standard-reduced speed for 'dry' operation to a lower standard-reduced speed line for 'wet' operation. Ingested large quantities of liquid water diminish the compressed gas mass flow through a blockage effect and lead to a decreased total pressure level at the compressor exit. Extra torque must be provided by the shaft to overcome added shear forces and inertia resulting from the liquid water films on the rotor blade surfaces [67,69,91].

4) Large ingested quantities of liquid water, that do not entirely evaporate along the compression system flow path, may enter the combustor and lead to local extinction phenomena in the primary flame zone. If a critical ingestion level is reached sudden combustor flame-out may occur, which then results in engine failure and shut-down. The more liquid water reaches the combustion zone the more deleteriously affected is the combustor fuel burn-out. To hold a certain thrust level, more fuel must be injected, deteriorating specific fuel consumption [1,30,73]. Liquid water flow reaching the combustion zone does counteract potentially positive effects from partial inter-cooling.

5) For large quantities of ingested water the mechanical impacts on compressor parts will be increased, in particular due to the presence of liquid films, tip section flow perturbations and statistic droplet impingement on blades. These impacts may include blade vibrations, increased erosion and corrosion threat.

Summarizing the above information, it can be stated that gas turbine system performance is affected when operating with droplet-laden gas flow depending on the ingested amount of liquid water mass per mass of (humid) air, the specific spray droplet-size spectrum (and liquid flow pattern) as well as the characteristic engine design, size and configuration. On the one hand deliberate use of water injection may lead to thermodynamic benefits overcoming the aerodynamic and mechanical system changes, while too large quantities of liquid phase will deteriorate compressor and overall engine performance. Note that all forms of interaction between a liquid water phase and the engine components can be categorized into being related to thermodynamics, aerodynamics or mechanics. In the following section the different forms of interaction between liquid water droplets, or sheets of liquid water respectively, and specific engine components are outlined.

4. FORMS OF INTERACTION BETWEEN WATER DROPLETS, SHEETS OF LIQUID WATER AND GAS TURBINE SYSTEM COMPONENTS

A single water droplet entering the air intake of an (aircraft) gas turbine engine has a specific temperature, a slip velocity relative to the gas path flow and a representative initial diameter corresponding to its shape. Basically, due to relative motion in high-speed flow environment droplet shapes will differ from being ideally spherical. In most literature sources concerning wet compression modeling for the purpose of simplicity the assumption of ideally spherical droplets is made. In high-speed two-phase flow droplets with velocity slip are exposed to significant shear forces, in particular close to boundary layers, which lead to spontaneous droplet break-up (rupture), droplet-droplet-collisions and coagulation. These phenomena are very often neglected in terms of wet compression model analysis. One should note that at a gas turbine intake section there is always a certain droplet-size spectrum, if exposed to water injection or water ingestion, going from very small to very large droplet diameters. A well-representative characterization of droplet-size spectra for comparison of two-phase flow effects is given by the so-called Sauter mean diameter (SMD), which relates overall volume of droplets to the total droplets' surface [29,38]. What is posted for water droplets here is also principally valid for sheets of liquid water entering the gas turbine engine. FIG 2. presents the different forms of interaction of a liquid water phase with a turbofan aero engine, being also representative for other gas turbine engine configurations. Consider a single water droplet in an ingested droplet-laden gas flow that tries to follow the gas phase surrounding it due to drag force, as implied in FIG 2. for a turbofan engine intake. The larger the droplet, the more will its trajectory deviate from main flow streamlines. In the front stage of a compression system, the moving droplet may impact on the first rotating blade section (fan rotor in FIG 2. respectively). If a collision with the rotor occurs depends on the position of the droplet relative to the compressor's axis of rotation, the shaft's rotational speed, the droplet's velocity vector and the geometric dimensions of the rotor section. The rotor dimension in axial flow direction is associated with the passage time, residence time respectively, of the transported droplet going through the blade row. If no collision with the rotor blades occurs, due to gas pressure and gas temperature increase the droplet temperature rises while going over into the downstream stator passage. As for the rotor blades, a collision of the transported droplet with the stator blades is generally possible. When the droplet has passed the compressor's first stage, impacting may then occur in one of the following stages (or the high-pressure compression system as implied in FIG 2.). Though, as the temperature of the droplet continuously increases while being delivered through the engine's compression system, evaporation will occur, leading to down-sizing of the droplet and reduced velocity slip relative to the gas path flow in the rear stages. Splashing and film formation due to droplets impacting on blades do affect the overall evaporation rate significantly and must thus be accounted for when modeling compressor 'wet performance'. These and other aspects relevant to gas turbine engine operation with droplet-laden gas flow are outlined in the following.

4.1. Air intake break-up processes (aero engines)

Water droplets in high-speed gas flow are exposed to significant shear forces that can lead to spontaneous droplet break-up in case that local droplet surface tension is overcome. The break-up processes may be of a different process type and result in distinct, representative final droplet sizes depending on specific boundary conditions. Relevant break-up process types are for example hat-type, bag-type and catastrophic droplet break-up [1,69]. Due to droplet rupture occurring in engine air intakes when ingesting droplet-laden gas flow at elevated flight velocities, it is difficult to relate meteorological data concerning local rainfall rate, liquid water content and rain-droplet-size spectra to representative input droplet sizes for wet compression modeling, that are 'seen' by the engine's compression system some distance downstream the engine inlet plane. Thus, some discrepancy in calculated results is to be expected when modeling aero engine performance in flight, which has to be weighted. Heat and mass transfer are significantly dependent on droplet volume and droplet surface.

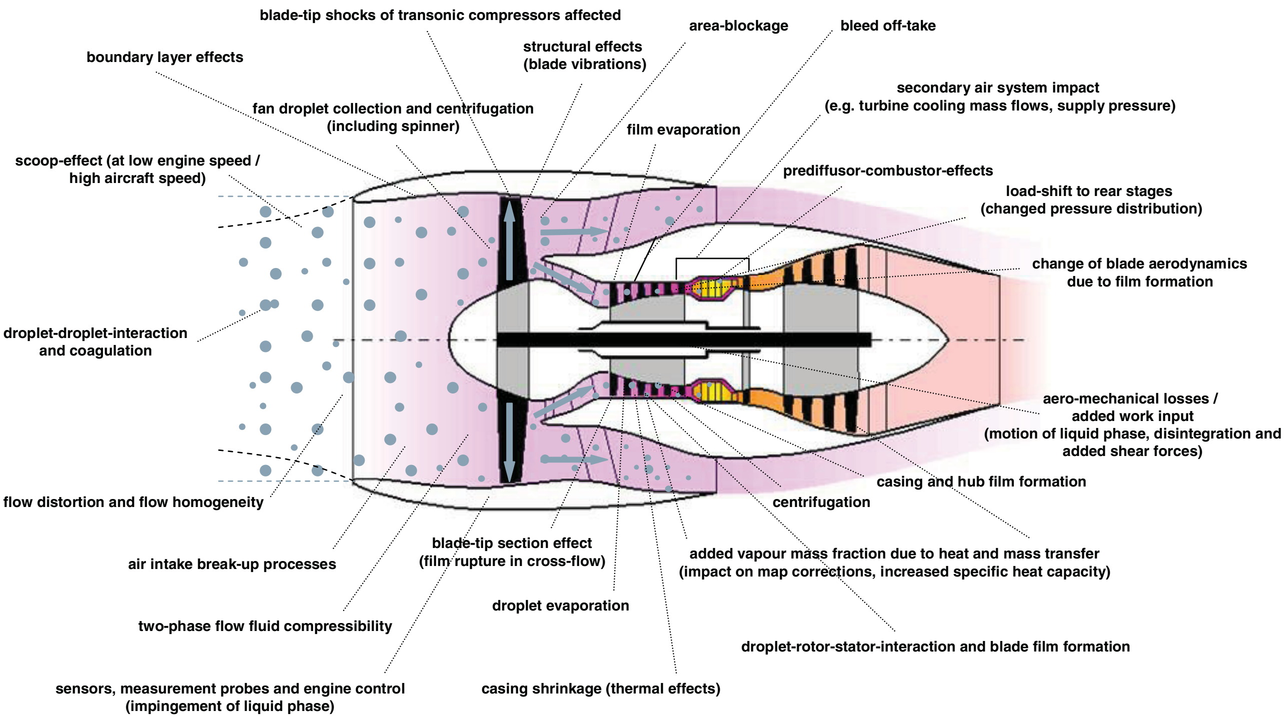

FIG 2. Forms of interaction of an unmixed turbofan engine with an ingested droplet-laden gas flow (adapted from [90])

4.2. Scoop-effect (aero engines)

Aero engines of aircrafts descending in inclement weather upon arrival are exposed to strong interaction with liquid water, since this flight condition combines high aircraft velocity with idle engine speeds. High flight velocity results in increased ram of the air capture stream tube, diminishing the ingested gas mass flow through spillage [1]. The ram effect is reinforced by the engine operating in idle setting. In such a flight condition the section area 'seen' by the air stream entering the engine due to ram is smaller than the geometrical inlet area potentially been hit by liquid water phase (see FIG 2.). The changed area ratio does affect the mass ratio of ingested liquid water to ingested gas flow toward higher numbers. This phenomenon is called 'scoop-effect' or 'scooping'. Ingestion of liquid water in idle setting is in particular hazardous due to lower gas pressure and temperature increase along the engine compression system compared to maximum continuous operation. Evaporation and centrifugation, splashing effects on structure parts and bleed extraction are reduced. In case of too low evaporation, liquid phase may enter the combustor affecting pressure-losses, reaction kinetics and fuel burn-out. If local temperature reduction reaches a critical level, combustor flame-out may occur. When running at maximum engine speed spillage is low, which means weak scooping.

4.3. Droplet-droplet-interaction and coagulation

Water droplets that are slipping through the gas path flow may collide with other moving droplets at elevated velocities. Depending on specific boundary conditions, distinct forms of interaction may then occur. Though, droplet collisions are partially-elastic impact processes. Upon impact droplets may coagulate, splash into secondary droplets or change momentum direction. In high-speed flow droplet geometry will deviate from ideal spherical shape. In most literature sources droplet-droplet-interaction and deviation from spherical shape are neglected in terms of wet compression modeling.

specific heat capacity greater than the one of dry air. Hence due to evaporation along the compressor path the gas mixture's overall heat capacity will be larger in the downstream system components.

4.5. Droplet-rotor-stator-interaction and film formation on compressor blades

Droplets entering the gas turbine engine intake may follow trajectories differing from gas path flow streamlines, so that collisions with rotating (and static) blade rows occur. Depending on specific boundary parameters, upon droplet impact and especially on rotating walls, different forms of interaction may result. Relevant impact parameters are for instance velocity slip and the direction of the impact trajectory relative to the blade wall motion, blade shape, surface roughness and surface temperature as well as gas flow properties. For a generalized analysis of droplet impact phenomena dimensionless numbers such as e.g. Weber, Reynolds, Ohnesorge and Laplace number can be introduced [1,36,39,69]. Principally four major forms of interaction upon droplet impact on a solid wall can be classified. First, colliding droplets may be hit back into the gas path flow in a partially-elastic impact process, without losing parts of their mass (*rebound effect*). Second, droplets at higher speeds may disintegrate upon impact in form of a *splashing* process into secondary droplets, which have a smaller diameter than the primary droplets. Droplet disintegration and added shear forces though absorb energy input that must be delivered by the compressor shaft. Third, droplets impinging on blades at lower relative

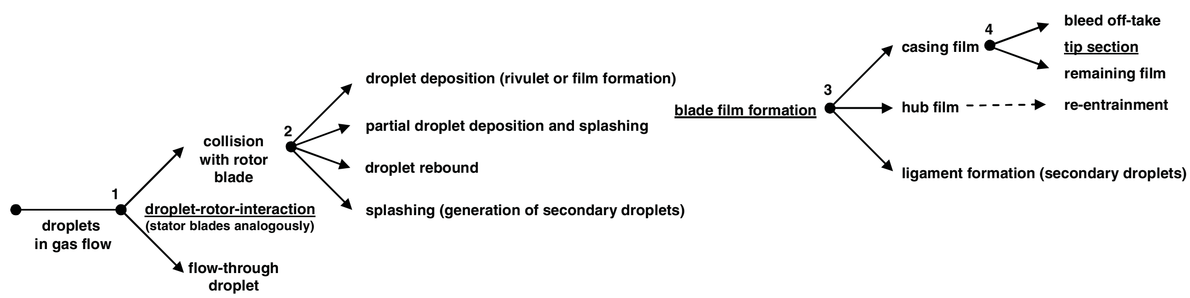

FIG 3. Forms of interaction of a droplet-laden gas flow with compressor blade rows and film formation ('droplet stream split')

4.4. Droplet evaporation

Droplets entering the gas turbine compression system are exposed to a pressure and temperature rise along the gas flow path, thereby continuously heating up by convection and thermal radiation. Heat transfer is strongly reinforced by velocity slip between the droplets and surrounding gas flow. The heat transfer to a droplet is coupled with a mass transfer from the liquid phase to the gaseous phase. Evaporation of liquid water consumes a comparably high amount of evaporation enthalpy, latent heat of evaporation respectively. The energy transfer needed for evaporation to occur does reduce the gas phase temperature. It should be noted, that accounting for some assumptions the total characteristic evaporation time for a single droplet in hot flow is approximately proportional to the square of its initial diameter [26,38]. Droplet size thus is an important measure for heat and mass transfer. Water vapor has a

speeds may be decelerated as such to form a thin liquid water film on the rotor blade surface (*droplet deposition*). This liquid film moves away from the compressor shaft axis toward the tip section and casing wall due to centrifugal force and in axial direction due to shear effects with the gas path flow. The motion of the film is counteracted by blade surface friction, which is increased by local film corrugation from statistically impacting droplets. The more liquid water is ingested, the larger will be the average blade film thickness. If ingestion reaches a critical limit 'virtual blade profile-thickening' occurs, affecting the profile aerodynamics and flow stability. As a consequence of blade profile modification stage-wise pressure rise and stage stall limits may significantly decrease. As fourth form of interaction, *partial splashing* can occur. This phenomenon is characterized by partial deposition and partial splashing of an impacting droplet. When having been centrifuged toward the casing, part of the rotor blade film is lead over into a casing wall film moving in axial flow direction. This film is driven by local shear forces. The other part of the rotor blade film may reach the blade's trailing edge. There the liquid film may separate from the blade, generating wavy water film structures, so-called ligaments. Interacting with the surrounding gas flow, these ligaments eventually disintegrate into secondary droplets. The secondary droplets resulting from ligament break-up are, as a rule, larger than the secondary droplets from droplet-blade splashing.

The previous statements mainly addressed droplet-rotor-interaction. Though, droplet collisions with stators do occur as well. Basically, on stator blade rows the same forms of interaction can be observed as on rotor blades, but the disposition to splashing phenomena is strongly reduced due to the lower relative velocities. The deposition of droplets on stators does also lead to formation of hub films moving in axial direction. Due to shear forces and engine geometry steps parts of the hub films will be re-entrained by the surrounding gas stream. FIG 3. displays the characteristic 'droplet stream split' of ingested droplet-laden flow in an engine's compressor. Droplet-stator-interaction though was attributed to split position 3 (hub film formation), since it is less dominant concerning splashing and liquid film formation in comparison to droplet-rotor-interaction and thus is less important for wet compression modeling.

4.6. Blade-tip section, film evaporation and casing effects

The centrifugal force due to shaft rotation transports rotor blade water films toward the blade-tip sections. Here the radial films interact with the tip clearance gas cross-flow. Local swirl leads to partial film disintegration. Moreover, the liquid film between casing and blade-tips being transported along the compressor tends to block the tip area so that the rotating blades have to overcome the additional friction when stirring through the wetted zone close to the casing. The aerodynamic performance of the blade-tip section may be deleteriously affected. While passing along the compressor, with rising temperature and pressure the liquid water casing film will continuously evaporate. On the one hand, due to film evaporation vapor mass flow does increase along the compressor path, accompanied by a cooling effect at the casing wall. As a result, casing shrinkage will occur to some extent. The benefit of a diminished tip clearance is though counteracted by increased aero-mechanical losses resulting from film disintegration, additionally induced turbulence and friction related to wet blade stirring. It is emphasized that film evaporation is less effective than droplet evaporation. Therefore, in case of large quantities of liquid water entering the compression system, significant parts of the casing film may potentially reach the combustor. The tip section is a crucial part of the compressor component in terms of operational stability, in particular when operating in liquid water two-phase flow, as has been shown in recent experiments [34]. It is emphasized that in the majority of scientific publications on wet compression analysis, which is the focus of this paper, the importance of the tip section zone is pointed out, but not covered by modeling [36].

4.7. Secondary air system, bleeds and turbine component, fan and bypass flow (aero engines)

Liquid water casing films are moving along the compressor path driven by shear forces. At positions where the casing does integrate bleed ports larger amounts of liquid water are pressed out of the compressor geometry, thus blocking the port area for gas flow. If the bleed port is connected to the bypass duct, amounts of liquid will leave the duct un-evaporated due to short residence times in high-temperature regions (see FIG 2.). Extracted two-phase bypass bleed flows will mix with the bypass stream and leave the engine over the bypass nozzle, if the engine configuration is unmixed. Some bleed ports in the compressor provide cooling air supply for the high- and low-pressure turbine components. Also these ports can take on significant amounts of water in liquid state from the casing film, which partially evaporate until reaching the hot turbine section. Pressure losses in the bleed lines will increase due to the presence of two-phase flow. As stated, small droplets evaporating in the compression system generate an inter-cooling effect affecting the pressure distribution along the compressor component. Local pressure levels in gas turbine operation with small amounts of ingested liquid water will thus differ from (off-)design values intended for 'dry' operation [38]. Generally speaking, altered pressure distribution due to the presence of a liquid water phase will affect the cooling air mass flows and thus overall engine performance. If engine components are exposed to unaccounted liquid water, local temperature changes will be the consequence. Note that aircraft customer bleeds and supply pressure bleed ports (e.g. for the bearings) may also be exposed to certain amounts of liquid water. With respect to aero engine applications, in particular the fan has a special function when liquid water is ingested by a turbofan engine. Due to the rotational motion of the fan component, in two-phase flow significant shares of the liquid phase will collide and be collected by the fan blades at elevated speeds. The principal forms of fan interaction with liquid phase are the same as for (high-pressure) compression system rotor blades stated above. Depending on fan speed, a dominant share of the formed liquid water blade film is centrifuged to the bypass duct, while the smaller share of liquid phase enters the engine core (fan-filtering function). The total amount of liquid water 'seen' at the turbofan engine entry section is not equal to the amount entering the engine's gas generator. In this context, fan, spinner and splitter design are important aspects. If operating in the transonic range, blade-tip shocks will be affected by the presence of two-phase flow [74].

4.8. Prediffusor-combustor-effects

If - by evaporation and bleed port extraction in the engine core compressor - the amount of liquid phase is not reduced as to achieve 'dry' fluid flow in front of the combustor, this may lead to significant local temperature reduction in the combustion zone. As a consequence, local pressure losses and incomplete fuel burn-out will increase, reaction kinetics and pollutant emissions will be affected [1,68,69,73].

4.9. Change in gas compressibility, heat and mass transfer

In engine performance calculations the behavior of turbo components is often described using component maps based on Mach similarity. Mach similarity covers the similarity of velocity triangles and flow angles referring to axial and radial Mach numbers, reduced mass flow and reduced speed respectively. These maps can either be referred to fully- or quasi-dimensionless numbers [1,76,91]. Though, the specific map of a compressor is valid only for a given set of inlet reference conditions, for which it was derived, incorporating the assumption of a constant isentropic exponent. At compressor operating conditions differing from the map reference conditions, correction procedures have to be applied. If ambient humidity (vapor phase only) changes relative to the map reference, a so-called 'humidity correction' must be applied, accounting for the specific gas constant and isentropic exponent being modified by a varied ambient air vapor mass fraction [75-78]. To be exact, though 'full similarity' of Mach numbers cannot be achieved, since isentropic exponent is basically not a constant, as assumed in map derivation. Humidity alters engine performance noticeably [1,79]. If the fluid stream entering the engine gas path is an evaporating two-phase flow, 'gas compressibility' will be affected on the one hand because of the mere presence of the droplets inserted into the gas phase and, in the rear stages of the compressor, by 'seeing' higher vapor content in the gas flow due to mass transfer from liquid into gaseous phase [80,81]. Liquid water droplets themselves may principally be regarded as incompressible. Due to the presence of an evaporating two-phase flow in the compressor, local 'gas transport properties' will change. Accordingly, local Reynolds numbers, affecting compressor performance in form of another map correction procedure, will change as well. Summarizing the above aspects, it can be stated that for a two-phase flow the compressor map derived from 'dry' conditions assuming Mach similarity is no longer valid, but can be taken as basis for wet compression modeling as long as considered quantities of liquid water are not 'too large'. Note that humidity correction must also be applied for the evaluation of the turbine component map.

4.10. Area-blockage, boundary layers and aero-mechanical losses resulting from droplet motion and disintegration

Liquid fluid being transported in an engine's compression system exerts a blockage effect on gas path flow. Although the volume fraction of the inserted water droplets or sheets of liquid film may be small compared to the gas flow volume, liquid fluid will act as additional source of frictional resistance owing to shear forces on phase boundaries. Frictional forces contribute to energy dissipation, which must be compensated for by work input to the compression system. Droplets impacting on rotating blades and forming corrugated liquid films must be accelerated to some speed of rotation, thus inducing extra torque demand for the compressor shaft. When being abandoned from the rotor blade rows, this liquid water will eventually hit static compressor parts again to dissipate the gained kinetic energy. Parts of the blade films and gas path flow droplets will disintegrate, thus consuming thermo-mechanical energy to overcome surface tension. Increased shear forces and droplet rupture will as well occur in the gas path flow boundary layers to rotating and static compressor parts, in which velocity gradients are large. The aero-mechanical losses due to the presence of a liquid phase in the compression system may overcome the thermodynamic benefits from evaporative inter-cooling at a certain quantity of liquid water with corresponding droplet sizes.

4.11. Structural effects, inlet distortion and flow inhomogeneity

When water droplets are present in the delivered gas turbine compressor flow, some will hit rotating blades at high relative speeds in form of a statistical process. Upon droplet collision rotor blades will deform and experience impulses acting against their direction of rotation, though adding extra torque demand. These collision phenomena are locally concentrated effects, depending on the mass of liquid water droplets per time unit hitting a specific blade spot. Rotor blades operating in two-phase flow may experience elastic flexural vibrations related to droplet impingement and water film stirring. Due to these vibrations blade angles of attack can alter slightly from regular off-design 'dry' operation, aerodynamically inducing blade flutter. As stated, due to water ingestion erosion threat may be increased in particular in the fan and the front stages of the gas turbine compression system. Owing to evaporation, splashing phenomena, centrifugal force and lowered relative velocity due to drag, the rear stages of the compression system will be less affected by aero-mechanical interaction with the liquid phase. In state-of-the-art modeling strategies for wet compression flexible structural parts are not considered. The compression system is then regarded as having fixed geometry.

Other important aspects concerning the analysis of gas turbine operation with liquid water are inlet distortion and two-phase flow section homogeneity all along the compressor flow path. Even specific injection of liquid water into an engine intake using best-quality fogging equipment for stationary gas turbines is principally a statistical process that never is fully repeatable. Relevant parameters such as liquid water mass injected per time unit, generated droplet-size spectra, local droplet temperatures, spatial droplet collection and the degree of intake droplet break-up will vary in a way in any circumstance. Unsteadiness and flow inhomogeneity parameters gain additional relevance when ingesting rain and fog under atmospheric conditions. It can be expected that the presence of two-phase flow will influence distortion effects and flow inhomogeneity along the compressor path to some extent, taking into account the large number of forms of interactions between droplet-laden flow with engine systems stated above [82,92].

4.12. Scaling of performance changes and test equipment

Apart from physical boundary conditions, in scientific literature it is emphasized that the impact of two-phase flow on engine performance is strongly depending on the specific gas turbine engine configuration, engine design and characteristic system dimensions, e.g. fan diameter, compressor length and area-ratio related to bypass ratio, if turbofan aero engines are considered. In particular the fan, spinner and splitter design are decisive for determining the quantity of liquid phase entering the engine core [1]. Evaporation is a transient process, so that liquid flow residence times in sections of elevated temperature are coupled with characteristic length scales. Small-size gas turbine engines will perform differently compared to large-size engines if being exposed to two-phase flow. Weighting and assessment of the strongly non-linear thermodynamic, aerodynamic and mechanical effects due to the presence of a liquid phase in the gas flow must be done for each gas turbine engine model individually. To be able do such analyses for 'wet' engine operation, appropriate modeling strategies must be derived for specific engine components, in particular the engine's compression system and the combustor, to reliably predict physical parameters in engine performance context. This paper focuses on two-phase flow modeling in axial flow compressors for engine performance calculations. Note that if computed engine performance output is compared to experimental data, sensors and measurement probes in the gas path may have been affected by droplet and liquid film impingement (potential interaction with the control system) [83]. For some test applications thus specially shielded measurement equipment has been designed in the past.

5. MODELING STRATEGIES FOR WET COMPRESSION SIMULATION IN PERFORMANCE CALCULATIONS

In a comprehensive literature review on wet compression modeling strategies for engine performance calculations, a variety of different calculation methods could be identified. These methods can principally be categorized into three major method groups based on their degree of theoretical modeling as follows.

<u>Empiricism-based methods</u> describe engine interaction with two-phase flow based on empirical correlations, which may be supplemented by fundamental thermodynamic theory, without in-depth modeling and analysis of component performance with an injected or ingested liquid phase [84,95-97]. These methods, which are related to extensive testing, can be found in flight handbooks and aero engine guidelines in form of charts, tables or mathematical formula e.g. from the 1950ies. These are mostly valid only for a single type of gas turbine engine. Detailed aspects, as for example ingested droplet sizes, are not covered by this modeling strategy due to high costs of testing for correlations. Empiricism focuses on simple-to-handle correction factors for thrust and fuel consumption owing to a processed liquid water mass flow. Input to the calculation then is 'dry' gas turbine engine performance at given ambient conditions.

<u>Thermo-analytical methods</u> form the second major method group for wet compression analysis in performance context. In these methods, based on a number of inputs such as an estimated axial pressure distribution along the compressor length, or compression rate respectively, a set of output parameters is derived involving thermodynamic equations. The resulting model parameters are then for instance specific work of compression and compressor exit temperature. The models perform in iterative steps and do either include equilibrium-based or non-equilibrium evaporation models. If 'dry'-operation compressor pressure distribution, e.g. from a stage-stacking scheme, is used as input for a thermo-analytical method, computed output can be applied for correcting 'dry' gas turbine compression system data for 'wet' operation. Compared to the first method group, thermo-analytical methods perform with a higher degree of modeling. A disadvantage of them is the requirement of an input pressure distribution, since actually compressor pressure rise will be affected by inter-cooling and aero-mechanical losses along the gas path. Primary purpose of thermo-analytical methods is a first-hand study of impacts of parameter changes on relevant system quantities in water injection applications.

The third group for wet compression modeling strategies encompasses <u>detailed numerical methods</u>. These approaches perform with the highest degree of theoretical modeling. They do include more forms of interaction between liquid phase and engine components and focus on applications in context of engine performance tools. Most models in this method group were developed in recent years as research efforts on water injection for power boosting of stationary gas turbines have been steadily increasing. The detailed numerical methods can be divided up in <u>five sub-categories</u>, depending on their way of treating wet compressor calculation as a part of engine performance simulation. These sub-categories are outlined in the following sections.

Note that in all the sub-category models the compressed gas phase is treated as mixture of perfect gases, although vapor mass fraction increases significantly due to evaporation [91]. In the following, the advantages, limits and discrepancies in the detailed numerical modeling strategies shall be outlined by referring to exemplary sub-category model versions from scientific literature. Model versions within a sub-category may thus differ in detail.

5.1.1. Enthalpy-entropy-approach

Enthalpy-entropy-approaches perform a step-wise calculation of wet compression employing a stage-stacking scheme and considering the formulation of an entropy change due to heat and mass transfer. FIG 4. describes the calculation procedure presented in [59] by Sexton et al.. In a first step enthalpy increase per stage (i) is determined based on 'dry' compression with an Euler work equation and stage angle data. Assuming constant isentropic stage efficiency, an evaporation model, which is coupled with an entropy balance considering liquid water, vapor and dry air, is introduced to adjust the thermodynamic condition at the stage outlet to operation with two-phase flow. In [62] the (simple) entropy balance is replaced by an entropy model incorporating several sub-steps, which do cover distinct thermodynamic and aero-mechanical effects of the two-phase flow. Having taken into account the entropy change, in [59] then the calculation scheme proceeds with the determination of the pressure increase in stage (i) based on Gibbs' relation and the subsequent computation of stages (i+1). In this manner, for a given input data set of e.g. compressor reduced speed, water amount injected and initial representative droplet size a 'wet operation' compressor map can be derived, which may then be used as input to engine performance simulation tools.

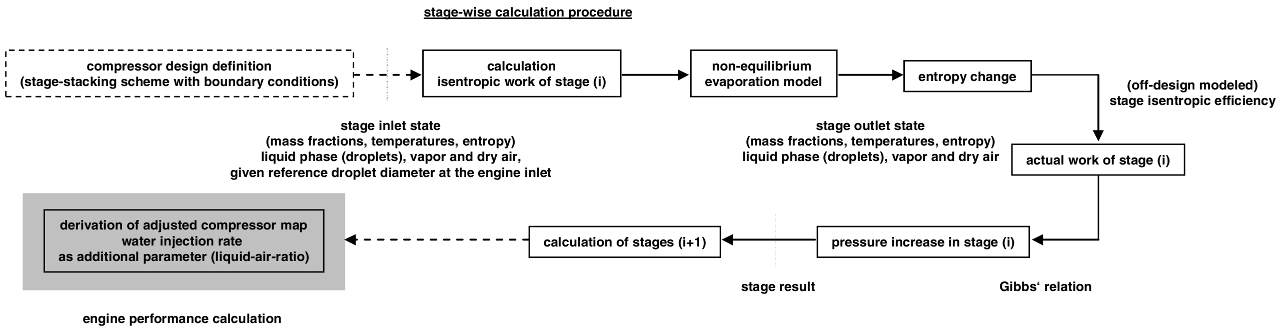

FIG 4. Schematic diagram of the enthalpy-entropy-approach from [59]

Enthalpy-entropy-approaches are in a way a form of inverted thermo-analytical calculation methods, since pressure distribution is a *result* of enthalpy(-entropy-)input. For the calculation of stage enthalpy increase, exact information on compressor blade geometries are needed, what may be disadvantageous in terms of engine performance calculations aimed at direct use of compressor maps. The derivation of 'wet operation' compressor maps for specific compressor geometries may take certain computation time, considering the possible variety of input parameters for map generation. The approach outlined in [59] focuses on evaporative inter-cooling at a given reference droplet diameter of 10 microns at the engine inlet, neglecting aero-mechanical forms of interactions of the liquid phase with the compressor.

discretization width for high-quality model results [98-101]. To cover most turbulent kinetic energy in high-speed flow and droplet motion, fine discretization is needed. The finer the discretization the more computational resources are needed to run simulations within appropriate periods of time. A source of uncertainty for sure is the selection of an adequate two-phase flow model for simulation including liquid phase interaction with walls upon collision, coagulation, film formation as well as heat and mass transfer. The fundamental idea of the CFD-adapted performance calculation is to run several CFD two-phase flow simulations of a (full) compressor geometry at varying operating conditions to gain correction factors for 'wet' compressor map total pressure ratio, efficiency and reduced mass flow. These factors can be used as input to

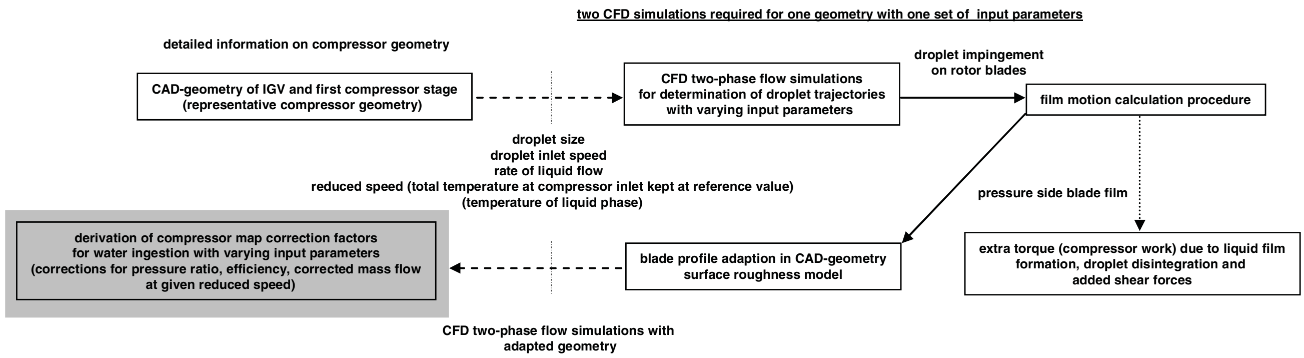

FIG 5. Schematic diagram of the CFD-adapted performance calculation from [69]

5.1.2. CFD-adapted performance calculation

CFD-adapted performance calculation takes advantage of two-phase flow modeling capability of computational fluid dynamics software tools (abbrev. CFD). Principally, as computational resources do steadily increase, it is possible to generate numerical meshes from full compressor geometries. On the mesh physical equations covering the engineering task can be discretized, boundary conditions be defined and solutions be derived with iterative solution procedures. For the simulation of two-phase compressor flow, model equations as gained from Euler-Euler or Euler-Lagrange approaches can be used. Limitation concerning CFD applicability to full compressor flow simulation is induced by required

engine performance calculation tools. Though, the gained map correction factors depend on the specific compressor geometry, the selected two-phase flow model, and the defined boundary conditions for the CFD simulation as the chosen turbulence model, evaporation modeling, grid width distribution and numerical solution procedures. Map corrections may be derived for distinct droplet-size spectra or reference Sauter mean diameters, droplet temperatures, varying ingested liquid water mass flows, radial inlet distributions, inlet slip velocities and different flow patterns of liquid water ingested. Because of the large number of potentially varied (CFD) parameter inputs to the flow simulation, the number of 'needed' simulations for derivation of map correction factors allowing 'good engineering judgment' is rather large. For practical applications, parameter variations have to be somehow performance prediction is limited due to its enormous demand for computational resources.

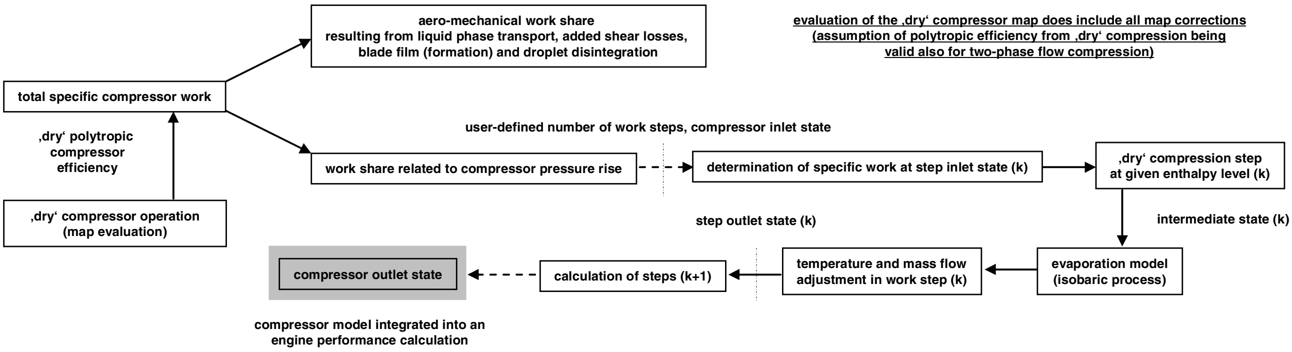

FIG 6. Schematic diagram of a parallel-stream-approach using overall compressor maps similar to the scheme from [90]

restricted. To keep computation time low, in [69] Nikolaidis performs two-phase CFD flow simulations for a J79-GE-17 compressor's first stage with variable inlet guide vane row, accounting for a single representative droplet diameter, varying liquid water flows and compressor rotational speeds. It is assumed that the simulated compressor stage geometry is representative for the full compressor geometry of a J79-GE-17 engine. Nikolaidis developed a scheme to incorporate aerodynamic compressor performance changes due local film formation on rotor blades by impacting droplets in his correction factor generating CFD procedure. As implied by FIG 5., in [69] first an initial two-phase flow CFD simulation of the stage geometry is performed with given input parameters at a constant rotational speed (fixed reference total temperature at compressor inlet). Droplets impacting on specific rotor blade spots are counted in simulated time as to feed the film motion calculation procedure. This procedure derives information on the quasi blade profile alteration due to film formation at the considered blade spots (virtual profile thickening). Additionally some information on local film motion (shear forces), added blade surface roughness and extra torque demand due to film formation are derived. With the input data from virtual blade profile thickening, the compressor stage CAD-geometry (CFD mesh respectively) is adapted to incorporate aerodynamic profile changes resulting from liquid films on the rotor. With the adapted mesh the two-phase CFD simulation of the compressor geometry is re-run. Having varied a reasonable number of simulation *input parameters* such as e.g. representative droplet diameter (SMD), velocity slip at the component inlet plane, droplet temperature and the ratio of liquid-to-gas-phase by mass, then simulation output is a set of correction factors to be coupled with compressor map reading in an engine performance calculation tool. Nikolaidis' simulations allow for evaporation effects. Though it must be noted that evaporation in a single stage compressor model will not be representative for evaporation along the whole compressor geometry. Since the analysis in [69] focuses on rain ingestion phenomena with low evaporation and dominant aerodynamic compressor performance changes due to large ingested quantities of liquid water, this assumption can be introduced. The applicability of this method in practice to gain engine-specific two-phase flow

5.1.3. Parallel-stream-approach using compressor maps

A further sub-category for detailed numerical modeling of two-phase flow compressor performance is given by methods based on a parallel-stream-approach using overall compressor maps. In this method type liquid and gas phase flow are virtually considered separately from each other. As shown in FIG 6. for a calculation scheme similar to the one defined by Kurzke in [90], first 'dry' compression is calculated at given power setting using an overall compressor map including all map corrections to derive 'dry' polytropic compressor efficiency. It is assumed that this compressor efficiency value is also valid for 'wet' compressor operation. Then the total specific work of the compressor at the given ('dry') operating point is divided up into a work share related to compressor pressure rise (pure thermodynamics) and aero-mechanical *losses* induced by droplet transport, droplet-wall-interaction and added shear forces in the gas path flow. Formulas to describe the aero-mechanical losses can be found in [90,91]. 'Dry' polytropic compressor efficiency and the specific compressor work share related to pressure rise are taken as basis for the 'wet' compressor calculation covering evaporation of liquid phase. The 'wet' calculation proceeds in a user-defined number of compressor work steps, each having two sub-steps. As first sub-step, *partial 'dry' compression* of the gas phase in step (k) is performed with the enthalpy delta from compressor work and the initial gas condition in (k) as an input. The liquid phase is ignored for the compression process. As second sub-step, an isobaric evaporation process employing a specific evaporation model is considered. With an enthalpy and mass flow balance, temperature and mass flow change of liquid and gas phase in (k) are then calculated. In the following the scheme proceeds with steps (k+1) until enthalpy steps add to the total compressor work related to compressor pressure rise (thermodynamics). The number of work steps is not necessarily related to the number of compressor stages.

5.1.4. Generalized performance curves coupled with shape factoring and evaporation modeling

Another detailed numerical wet compression calculation method is described with generalized performance curves coupled with shape factoring and evaporation modeling in context of stage-stacking schemes. FIG 7. below presents the calculation method proposed by Bagnoli, Bianchi, Melino et al. in [45]. Each compressor stage is described by a stage performance map (phi-psi-form) as independent calculation module in the engine performance configuration. Stage maps can be derived from given geometries in different ways. When exact compressor stage blade geometries are not known, phi-psi

between stage inlet and exit. In other versions of the general performance curve-based detailed modeling approach, stage-wise evaporation occurs at the stage exit only. Some model versions cover more forms of interaction than others, though stage map representation is regarded as not being affected by e.g. droplet impacting, film formation and altered aerodynamics [44].

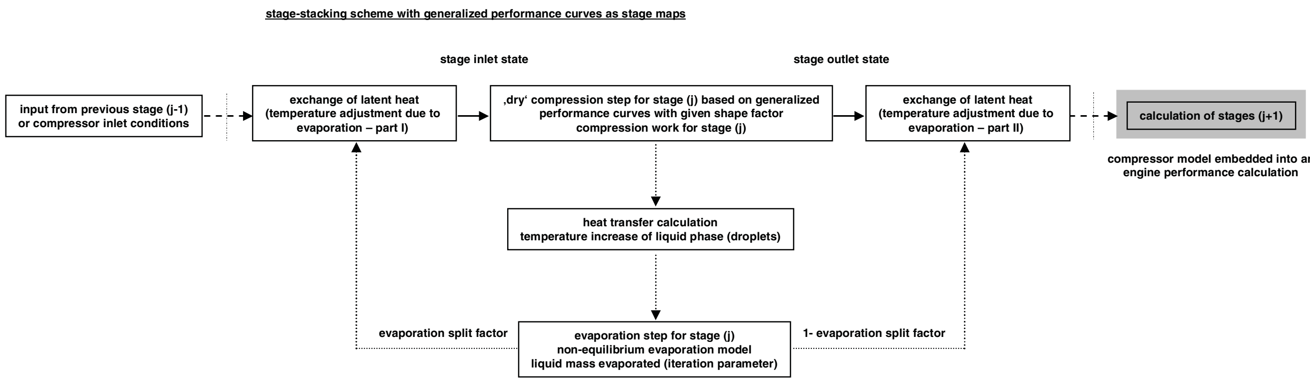

FIG 7. Schematic diagram of the generalized performance curve approach with shape factoring and evaporation modeling from [45]

map representation can be used accounting for a so-called shape factor that models aerodynamic discrepancies to a general reference stage with an empirical approach [85,86]. Note that 'dry' *compressor stage performance representation* with generalized curves and shape factoring is exposed to a certain degree of uncertainty, since the value of the stage shape factor must be somehow determined, if not estimated or guessed. In the scheme shown in FIG 7. as a first step 'dry' stage operation (j) is calculated, resulting in a temperature and pressure increase at given shape factor. With the thermodynamic condition at the stage inlet and the newly computed one at stage outlet, a reference state is determined to be input for an evaporation model routine. The result of the evaporation routine is a quantity of evaporated liquid water for stage (j). This liquid water is virtually split into a quantity evaporating in front of the stage and a quantity evaporating at the stage's outlet plane. Accounting for an enthalpy balance and with the quantity of liquid water evaporating in the front part, the input temperature to the compression in stage (j) is adjusted. Subsequently compression based on the stage performance map with given shape factor is re-done. The stage exit temperature is then adjusted for the other part of liquid evaporating. Then a new reference state is derived as input for the evaporation model, in turn affecting compression. This iterative procedure is performed for stage (j) until the change of mass of liquid water does no longer exceed a specific residual value. Output quantities from stage (j) are then transferred to stage (j+1) in the stage-stacking scheme. The split of the evaporation process between the front and outlet section of stage (j) is done as to virtually separate evaporation from the heat transfer between the gas and the liquid water phase. Some model sensitivity is introduced by the selection of the factor splitting liquid water evaporation

5.1.5. Velocity-triangle modification procedure

A highly-iterative type of detailed numerical wet compression modeling methods is classified by velocity-triangle modification procedures. Velocity-triangle-based methods incorporate exact geometric blade design data into a mean-line, blade-wise calculation scheme (upgraded stage-stacking scheme), which is part of an engine performance model. Application of this method focuses on exact knowledge of flow triangle modification due to various two-phase flow effects in the rotor and stator sections, going from the compressor's front stage to the outlet stage. FIG 8. describes the velocity-triangle modification procedure by Leonardo, Tsuchiya & Murthy from [28,87] in very abstracted form (Purdue WINCOF code). In the model first a 'dry' calculation is performed for stage (g), with rotor blade and stator blade considered separately from each other. Subsequently highly-iterative and complex velocity-triangle-correcting steps are incorporated, e.g. considering droplet motion and droplet impingement on blades, film formation and droplet rebound, added shear effects and pressure losses, ligament break-up processes as well as heat and mass transfer effects. In [28,87] different calculation *modes* for distinct droplet size classes are defined, since stage performance will change depending on the size (and quantity) of liquid water droplets inserted. As stage-wise calculation proceeds, due to droplet-size varying two-phase flow effects the large-droplet calculation mode will eventually develop into the small-droplet calculation mode. Another very detailed version of velocity-triangle modification for wet compression modeling in high-fogging context is defined by Khan & Wang in [47-49]. The large number of iterations in their modeling approach becomes obvious considering the definition of their calculation scheme in three published papers. Velocity-triangle

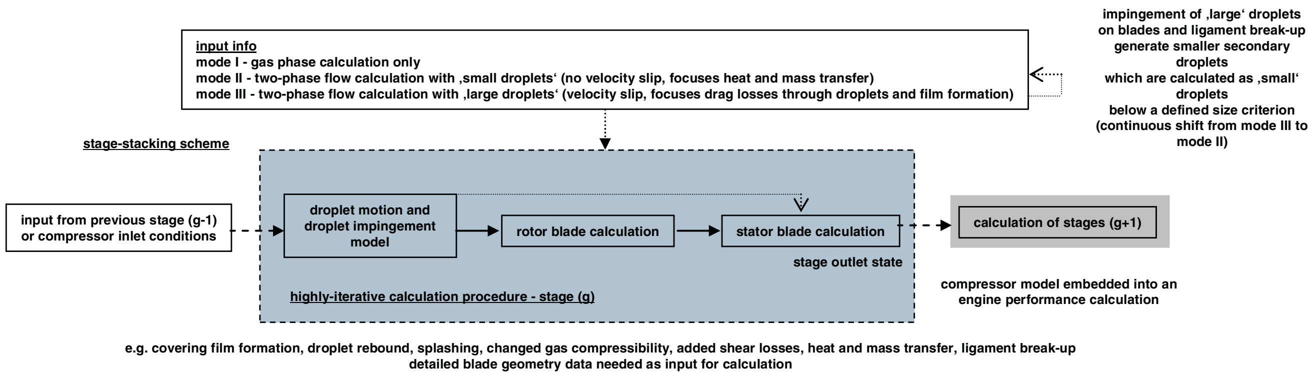

FIG 8. Schematic diagram of the velocity-triangle-modification procedure from [28,87]

modification is also implied by Matz, Kappis, Cataldi et al., who couple their calculation method with non-equilibrium evaporation, droplet splashing and blade film formation as well as a flow angle deviation adder [21,38,65]. Note the hints concerning secondary air system flow changes in [21]. Focus of the approaches outlined in [21,38,47-49,65] is the calculation of wet compression in terms of water injection for stationary gas turbine engines, whereas the Purdue WINCOF code addresses the simulation of wet compression in terms of rain ingestion into aero engines (turbofan engines) [28,87-89].

5.2. Droplet Motion

A side-aspect of special interest in terms of detailed wet compression modeling is **the description of the motion of droplets** when travelling along the compressor gas path. Droplet motion is coupled with velocity slip and thus heat and mass transfer processes as well as ‘droplet stream split’ according to FIG 3., if impingement processes are considered. In literature three distinct modeling approaches to cover droplet motion are presented:

1) Droplet motion without slip relative to gas path flow (‘small’ droplets in particular)
2) 1D-droplet motion with slip relative to gas path flow in mean-line direction [38]
3) (Quasi-)3D-droplet motion according to Lagrange equations or an analogous model [27,36,40,87]

From method 1) to method 3) modeling complexity does increase. As an example, in [36] Zhluktov, Bram & De Ruyck determine ‘droplet stream split’ corresponding to FIG 3. by using trial particle motion analysis based on Lagrange equations. It remains questionable whether the effort of such complex 3D-droplet motion modeling is justified by more accurate results in comparison to 1D-motion modeling or a simple guess for velocity slip.

In literature some information on weighting of the forms of interaction between liquid phase and the compression system has been provided. It has been indicated that e.g. representative droplet size (spectra), splashing effects, blade and casing film formation as well as bleed pressure supply change and non-equilibrium evaporation are crucial modeling factors for adequate wet compression simulation.

6. MODEL EVALUATION AND EXPERIMENTAL DATA FROM SCIENTIFIC LITERATURE

In TAB 1. in the appendix different versions of the detailed numerical wet compression models, that have been found in the comprehensive literature review, are presented. The model versions’ assignments to a certain modeling sub-category are indicated by the Roman letters standing at the model author names and changed table line colors. Additionally, important remarks concerning the approaches’ focus of modeling, their specific advantages and limitations are given. The evaluation of model version advantages and limitations is linked with an assessment of three mutually-coupled model attributes:

- Physical and mathematical *complexity*
- *Usability*, referring to use in engine performance tools with direct evaluation of overall compressor maps or stage characteristics
- *Validity*, referring to modeling assumptions and gained experimental knowledge to give proof to them

Please check the model evaluation key presented in the appendix of this paper.

TAB 2. provides a summary of the experimental data found in scientific literature, that have formed the basis for wet compression analysis and model validation in the past. The tabulated remarks give hints on the respective tested compressor or whole gas turbine engine system, the injected liquid-water-air-ratios, droplet sizes, measured parameters as well as made observations. Note that only few experimental data are available in scientific literature concerning ‘wet’ compressor operation in tested whole gas turbine engine systems. Though, academic research work has focused on investigating the impacts of droplet-laden gas flow on compressor performance apart from whole engine performance analysis. It has also been found that there’s little detailed information and few experimental data concerning compressor stability aspects, like motion of surge line due to the presence of an ingested two-phase flow. In this context, the research work by Williams in [31,34] is a highlight. Some publications concerning compressor stability address CFD analyses of blade aerodynamics in droplet-laden flow and experimental analyses of two-phase flow effects around non-rotating blade cascades [32,35,74,102].

As can be seen from TAB 1., there is no comprehensively validated detailed numerical wet compression modeling strategy that fulfills the criterion of high applicability in gas turbine engine performance tools. Thus, the term 'comprehensively' is related to testing scenarios with largely varying input parameters such as liquid water injection rate, characteristic droplet size, liquid phase flow pattern as well as different engine speeds and ambient conditions. The term 'high applicability' refers to model use in gas turbine performance simulation tools with compressor component representation preferably in form of an overall compressor map. Consider that setting-up a well-defined stage-stacking scheme may be time-consuming, so that wet compression modeling with such schemes may not achieve relevance in performance engineering practice, when quick two-phase flow performance predictions shall be generated for a specific system. Numerical wet compression modeling approaches demanding detailed blade geometry input may be disadvantageous for use in map-based performance calculation programs, since, if not given, blade geometry must be approximated [51,59]. This for sure introduces uncertainty. The same statement applies to wet compression modeling approaches involving stage performance maps, performance curves respectively [45,46]. For both the parallel-stream-approach using compressor maps and CFD-adapted performance calculation a sole literature source could be found in available databases.

7. CONCLUSIONS

Summarizing the information provided on the preceding pages, it can be stated that in the past a large number of different numerical methods for wet compression calculation were derived in context of engine performance simulations. In this paper these methods were categorized and information concentrated as to serve as a starting point for future wet compression modeling research for stationary gas turbine and aircraft engine applications. It has been found that there is no comprehensively validated detailed numerical wet compression modeling strategy that fulfills the criterion of high applicability in terms of gas turbine engine performance tools. The distinct forms of interaction of an injected or ingested liquid water phase with gas turbine engine components such as the air intake, the compression system and the combustor are complex and need further in-depth analysis, both experimentally and theoretically. This statement does particularly concern low-pressure compressor (fan) and high-pressure compressor operation in two-phase flow as these are decisive (aero) engine components. Some topics of particular interest concerning safe engine operation with two-phase flow have yet not been covered by published experimental results. Further research work is needed concerning operational stability aspects like for instance compressor surge margin degradation, deterioration of aerodynamic blade performance in highly-loaded two-phase flow and structural effects. Such analyses could well-supplement future wet compression modeling efforts. Comparing the model descriptions from the list of referenced publications of this paper, it can be stated that most detailed numerical wet compression modeling methods are based on a *step-wise* calculation scheme or stage-stacking scheme, respectively. In these procedures though very often an initial 'dry' compression calculation step is followed by a 'correction step', adjusting initial output for two-phase flow effects. Considering the tabulated data in the appendix of this paper, it can be seen that 'large'-droplet and 'small'-droplet size application cases have been differentiated in most wet compression tool developments up-to-now. In terms of aero engine performance calculations, programs must be able to cope with boundary conditions referring to either case to be able to predict engine performance with high quality.

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REFERENCES

[1] NATO-RTO-AGARD Advisory Report No. 332. Recommended Practices for the Assessment of the Effects of Atmospheric Water Ingestion on the Performance and Operability of Gas Turbine Engines. Propulsion and Energetics Panel Working Group 24, September 1995.

[2] Parikh P., Hernan M., Sarohia V. Quantitative Determination of Engine Water Ingestion. AIAA-86-0307. AIAA 24th Aerospace Sciences Meeting, January 1986.

[3] Erhard W., Gabler R., Preiss A. et al. Monitoring and Control of Helicopter Engines at Abnormal Operating Conditions. RTO AVT Symposium on 'Design Principles and Methods for Aircraft Gas Turbine Engines' (publish. RTO MP-8), May 1998.

[4] Hale A., Klepper J., Hurwitz W. A numerical capability to analyze the effects of water ingestion on compression system performance and operability. GT2005-68480. ASME Turbo Expo 2005, June 2005.

[5] Spencer J.H., Archer D.C. The effect of condensation within an aircraft inlet duct on installed turbofan engine performance. Phil.: Nav. Air Prop. Test Center 9.1-9.10, 1971.

[6] Blake J.C. Effects of condensation in the JT9D turbofan engine bellmouth inlet. AIAA-75-1325. AIAA/SAE 11th Propulsion Conference, September/October 1975.

[7] Zerkle R.D., Colley W.C., Doel D.L. Analysis of moisture condensation in engine inlet ducts. AIAA/ASME Symposium on Particulate Laden Flows in Turbomachinery, 1982.

[8] Young J.B. Condensation in jet engine intake ducts during stationary operation. 93-GT-5. International Gas Turbine and Aeroengine Congress and Exposition (ASME), May 1993.

[9] Potter J., Tatam R.P. Optical condensation measurement in gas turbine engine inlets. SPIE Vol. 3172 0277-786X/97. SPIE Conference on Optical Technology in Fluid, Thermal and Combustion Flow, 1997.

[10] Münzberg. Flugantriebe – Grundlagen, Systematik und Technik der Luft- und Raumfahrtantriebe. Springer Verlag Berlin, 1972 (in German).

[11] Wetzel I.T., Jennings B.H. Water spray injection of an axial flow compressor. Proc. Midwest. Power Conf. 376-380, April 1949.

[12] Hamrick J.T., Beede W.L. Some Investigations With Wet Compression. ASME Paper 1952-SA-15, 1952 (prepared for NACA, Lewis Flight Propulsion Laboratory).

[13] Moult E.S. Water Injection – What it is – and what is does. De Havilland Gazette No. 76, 1953.

[14] Triebnigg H. Leistungssteigerung von Luftfahrttriebwerken durch Wasser-Methanoleinspritzung. VDI Luftfahrttechnik Bd. 1 p. 30-35, Juni 1955 (in German).

[15] Jahnke H. Das Betriebsverhalten von Axialverdichtern bei Wassereinspritzung am Verdichtereintritt. Diss. Technische Universität Dresden, März 1968 (in German).

[16] Weichert M. Wassereinspritzung in den Verdichter von Gasturbinen zur Leistungssteigerung und Reduktion von Stickoxid-Emissionen. Report BMFT-FB-T / 82-075 (Technische Universität Berlin, Prof. Gašparovic), May 1982 (in German).

[17] Utamura M., Kuwahara T., Murata H. et al. Effects of intensive evaporative cooling on performance characteristics of land-based gas turbine. PWR-Vol. 34, 1999 Intern. Joint Power Generation Conference Vol. 2 Power (ASME), 1999.

[18] Nuding J., Johnke T. Increasing GT Power and Efficiency Through Wet Compression – Conference Presentation. Power-Gen Conference Europe, 2002.

[19] Bhargava R., Meher-Homji C.B. Parametric analysis of existing gas turbines with inlet evaporative and overspray fogging. GT2002-30560. ASME Turbo Expo 2002, June 2002.

[20] Jolly S., Cloyd S. Performance enhancement of GT 24 with wet compression. Power-Gen International, December 2003.

[21] Cataldi G., Güntner H., Matz C. et al. Influence of high fogging systems on gas turbine engine operation and performance. GT2004-53788. ASME Turbo Expo 2004, June 2004.

[22] Bhargava R.K., Bianchi M., Melino F. et al. Gas turbine fogging technology: A state-of-the-art review Part II: Overspray fogging – analytical and experimental aspects. GT2005-68337. ASME Turbo Expo 2005, June 2005.

[23] Chaker M.A., Meher-Homji C.B. Gas turbine power augmentation by inlet air fogging and wet compression – technology overview and operating experience. ISROMAC12-2008-20165, 12th Int. Symp. On Transp. Phen. And Dyn. Of Rot. Machinery, February 2008.

[24] Higuchi S., Koganezawa T., Horiuchi Y. et al. Test results from the advanced humid air turbine system pilot plant – Part I: Overall performance. GT2008-51072. ASME Turbo Expo 2008, June 2008.

[25] Hundseid Ø., Brenne L., Bakken L.E. et al. Wet gas performance of a single stage centrifugal compressor. GT2008-51156. ASME Turbo Expo 2008, June 2008.

[26] Quandt E. Water Droplet Evaporation in Air During Compression in a Gas Turbine Engine. Technical Memorandum NSWCCD-TM-80-96/06, Machinery Research and Development Directorate. Carderock Division, Naval Surface Warfare Center, June 1996.

[27] Tsuchiya T., Murthy S.N.B. Water Ingestion into Jet Engine Axial Compressors. AIAA-82-0196. AIAA 20th Aerospace Sciences Meeting, January 1982.

[28] Leonardo M., Tsuchiya T., Murthy S.N.B. PURDU-WINCOF – A Computer Code for Establishing the Performance of a Fan-Compressor Unit with Water Ingestion. NASA CR-168005, prepared for NASA Lewis Research Center by Purdue University, 1982.

[29] Chaker M., Meher-Homji C.B., Mee T. Inlet fogging of gas turbine engines – Part A: Fog droplet thermodynamics, heat transfer and practical considerations. GT2002-30562. ASME Turbo Expo 2002, June 2002.

[30] Warwick W.R. Effect of water ingestion on J79-GE-8 altitude performance. ATL-SP-4 63-ENV-5. U.S. Naval Air Turbine Test Station, Trenton NJ, 1963.

[31] Day I., Freeman C., Williams J. Rain ingestion in axial flow compressors at part speed. GT2005-68582. ASME Turbo Expo 2005, June 2005.

[32] Ulrichs E., Joos F. Experimental investigations of the influence of waterdroplets in compressor cascades. GT2006-90411. ASME Turbo Expo 2006, May 2006.

[33] Grüner T.G., Bakken L.E., Brenne L. et al. An experimental investigation of airfoil performance in wet gas flow. GT2008-50483. ASME Turbo Expo 2008, June 2008.

[34] Williams J. Further effects of water ingestion on axial flow compressors and aeroengines at part speed. GT2008-50620. ASME Turbo Expo 2008, June 2008.

[35] Eisfeld T., Joos F. Experimental investigation of two-phase flow phenomena in transonic compressor cascades. GT2009-59365. ASME Turbo Expo 2009, June 2009.

[36] Zhluktov S., Bram S., De Ruyck J. Injection of water droplets in an axial compressor. Experimental Heat Transfer, Fluid Mechanics and Thermodynamics (p. 1415/1420). Edizioni ETS Pisa, 2001.

[37] Monteyne L. Simulatie van waterinjectie in gas-compressoren. Thes. Vrije Universiteit Brussel, Academiejaar 2006-2007 (in Flemish).

[38] Matz C., Kappis W., Cataldi G. et al. Prediction of evaporative effects within the blading of an industrial axial compressor. GT2008-50166. ASME Turbo Expo 2008, June 2008.

[39] Williams J. Water ingestion in jet engines. PhD Thes. University of Cambridge, 2003.

[40] White A.J., Meacock A.J. Wet compression analysis including velocity slip effects. GT2010-23793. ASME Turbo Expo 2010, June 2010.

[41] Grigor'yants R.R., Zalkind V.I., Zeigarnik Y.A. et al. Special Features of the Behavior of the Liquid Phase in High-Speed Compressors of Aircraft Engine-Derivative Gas-Turbine Units and Their Impact on Characteristics and Effectiveness of 'Wet Compression'. Thermal Engineering Vol. 54 No. 4 p. 310-318, 2007.

[42] Ludorf R.K., Tronbøl T.H. , Elder R.L. et al. Stage rematching as a result of droplet evaporation in a compressor. 95-GT-194. International Gas Turbine and Aeroengine Congress and Exposition (ASME), June 1995.

[43] Horlock J.H. Compressor performance with water injection. GT2001-0343. ASME Turbo Expo 2001, June 2001.

[44] Roumeliotis I., Mathioudakis K. Evaluation of interstage water injection effect on compressor and engine performance. GT2005-68698. ASME Turbo Expo 2005, June 2005.

[45] Bagnoli M., Bianchi M., Melino F. et al. Development and validation of a computational code for wet compression simulation of gas turbines. GT2006-90342. ASME Turbo Expo 2006, May 2006.

[46] Sanaye S., Rezazadeh H., Aghazeynali M. et al. Effects of Inlet Fogging and Wet Compression on Gas Turbine Performance. GT2006-90719. ASME Turbo Expo 2006, May 2006.

[47] Wang T., Khan J.R. Overspray and Interstage Fog Cooling in Compressor Using Stage-Stacking Scheme – Part 1: Development of Theory and Algorithm. GT2008-50322. ASME Turbo Expo 2008, June 2008.

[48] Wang T., Khan J.R. Overspray and Interstage Fog Cooling in Compressor Using Stage-Stacking Scheme – Part 2: Case Study. GT2008-50323. ASME Turbo Expo 2008, June 2008.

[49] Khan J.R., Wang T. Overspray Fog Cooling in Compressor using Stage-Stacking Scheme with Non-Equilibrium Heat Transfer Model for Droplet Evaporation. GT2009-59590. ASME Turbo Expo 2009, June 2009.

[50] Roumeliotis I., Mathioudakis K. Evaluation of water injection effect on compressor and engine performance and operability. Applied Energy 87 (2010) 1207-1216, 2010.

[51] Montalvo-Catano G., O'Brien W. Performance modeling of a power generation gas turbine with wet compression. GT2011-46584. ASME Turbo Expo 2011, June 2011.

[52] Hill P.G. Aerodynamic and Thermodynamic Effects of Coolant Injection on Axial Compressors. The Aeronautical Quarterly Part 4 (14) p. 331-348, November 1963.

[53] Zheng Q., Sun Y., Li S. et al. Thermodynamic analyses of wet compression process in the compressor of gas turbine. GT2002-30590. ASME Turbo Expo 2002, June 2002.

[54] Härtel C., Pfeiffer P. Model analysis of high-fogging effects on the work of compression. GT2003-38117. ASME Turbo Expo 2003, June 2003.

[55] Petr V. Simple and regenerative GT with wet compression. ISROMAC10-2004-028, 10th Int. Symp. On Transp. Phen. And Dyn. Of Rot. Machinery, March 2004.

[56] Kang J., Cha B., Yang S. Thermodynamic and Aerodynamic Meanline Analysis of Wet Compression in a Centrifugal Compressor for a Microturbine. Intern. Symp. On Air Breathing Engines ISABE-2005-1269, 2005.

[57] Hoang N.H., Hasegawa Y., Kubo T. et al. Fundamental Study on Water Injection Cooling System of Centrifugal Compressor for Micro Gas Turbine. Proc. Of Intern. Symp. On EcoTopia Science 2007, ISETS07, 2007.

[58] Kim K.H., Ko H., Perez-Blanco H. Analytical modeling of wet compression of gas turbine systems. Applied Thermal Engineering 31 (2011) 834-840, 2011.

[59] Sexton M.R., Urbach H.B., Knauss D.T. Evaporative compressor cooling for Nox suppression and enhanced engine performance for naval gas turbine propulsion plants. 98-GT-332. International Gas Turbine and Aeroengine Congress and Exhibition (ASME), June 1998.

[60] White A.J., Meacock A.J. An evaluation of the effects of water injection on compressor performance. GT2003-38237. ASME Turbo Expo 2003, June 2003.

[61] Sexton W.R., Sexton M.R. The effects of wet compression on gas turbine engine operating performance. GT2003-38045. ASME Turbo Expo 2003, June 2003.

[62] Abdelwahab A. An Investigation Of The Use Of Wet Compression In Industrial Centrifugal Compressors. GT2006-90695. ASME Turbo Expo 2006, May 2006.

[63] Kim K.H., Perez-Blanco H. An assessment of high-fogging potential for enhanced compressor performance. GT2006-90482. ASME Turbo Expo 2006, May 2006.

[64] Bettocchi R., Morini M., Pinelli M. et al. Set up of an experimental facility for the investigation of wet compression on a multistage compressor. GT2010-22451. ASME Turbo Expo 2010, June 2010.

[65] Ripken M. Evaluation of evaporative compressor cooling (ECC) mean-line model using an additional wet compression dataset; Evaluation of compressor blade erosion. Thes. Technische Universität Dresden, 2008.

[66] Khan J.R., Wang T. Simulation of Inlet Fogging and Wet-compression in a Single Stage Compressor Including Erosion Analysis. GT2008-50874. ASME Turbo Expo 2008, June 2008.

[67] Roumeliotis I., Mathioudakis K. Water injection effects on compressor stage operation. GT2006-90427. ASME Turbo Expo 2006, May 2006.

[68] Hendricks R.C., Shouse D.T., Roquemore W.M. Water Injected Turbomachinery. NASA/TM-2005-212632, ISROMAC10-2004-039, March 2005.

[69] Nikolaidis T. Water ingestion effects on gas turbine engine performance. PhD Thes. Cranfield University, October 2008.

[70] Nikolaidis T., Teixeira J.A., Pilidis P. et al. Water film formation on an axial flow compressor rotor blade. GT2008-50137. ASME Turbo Expo 2008, June 2008.

[71] Das K., Hamed A.A., Basu D. Droplet trajectories and collection on fan rotor at off-design conditions. GT2006-91214. ASME Turbo Expo 2006, May 2006.

[72] Williams J., Young J.B. Movement of deposited water on turbomachinery rotor blade surfaces. GT2006-90792. ASME Turbo Expo 2006, May 2006.

[73] Laing P., Murthy S.N.B. Gas Turbine Prediffuser-Combustor Performance During Operation with Air-Water Mixture. DOT/FAA/CT-93/52 Interim Report, August 1994.

[74] Yang H., Zheng Q., Luo M. et al. Wet Compression Performance of a Transonic Compressor Rotor at its Near Stall Point. J. Marine Sci. Appl. (2011) 10: 49-62, 2011.

[75] Grabe W., Bird J. Humidity Effects on Gas Turbine Performance. Technical Report TR-ENG-003 / NRC No. 30241 National Research Council Canada, Division of Mechanical Engineering, Engine Laboratory, December 1988.

[76] Volponi A.J. Gas turbine parameter corrections. 98-GT-347. International Gas Turbine and Aeroengine Congress and Exhibition (ASME), June 1998.

[77] Mathioudakis K., Tsalavoutas A. Uncertainty reduction in gas turbine performance diagnostics by accounting for humidity effects. GT2001-0010. ASME Turbo Expo 2001, June 2001.

[78] Kyritsis V.E., Pilidis P. An analytical approach for gas turbine parameter corrections. GT2008-50716. ASME Turbo Expo 2008, June 2008.

[79] Bryant T.L. Factors of Temperature and Humidity in Relation to Aircraft Performance. Section F, Geography. Proceedings of the Oklahoma Academy of Science Vol. 45 p. 193-197, 1965.

[80] Bursik J.W. Pseudo-sonic velocity and pseudo-Mach number concepts in two-phase flows. NASA Technical Note NASA-TN-D-3734 (George C. Marshall Space Flight Center Huntsville, Ala., December 1966.

[81] Nguyen D.L., Winter E. R. F., Greiner M. Sonic velocity in two-phase systems. International Journal of Multiphase Flow, Volume 7 Issue 3 p. 311-320, June 1981.

[82] Sereda S.O., Gel'medov F. Sh., Muntyanov I.G. Experimental Study of the Effect of Water Injection into the Inlet Channel of a Multistage Axial Compressor on Its Characteristics. Thermal Engineering Vol. 51 No. 5 p. 409-414, 2004.

[83] Murthy S.N.B., Ehresman C.M. Water ingestion simulation: Test needs. Intern. Symp. On Air Breathing Engines ISABE 89-7120, 1989.

[84] Wilcox E.C., Trout A.M. Analysis of thrust augmentation of turbojet engines by water injection at compressor inlet including charts for calculating compression processes with water injection. National Advisory Committee for Aeronautics NACA Report 1006, 1951.

[85] Spina P.R. Gas turbine performance prediction by using generalized performance curves of compressor and turbine stages. GT2002-30275. ASME Turbo Expo 2002, June 2002.

[86] Bhargava R.K., Bianchi M., Melino F. et al. Influence of compressor performance maps shape on wet compression. GT2008-50761. ASME Turbo Expo 2008, June 2008.

[87] Tsuchiya T., Murthy S.N.B. Effect of water on axial flow compressors – Part I Analysis and predictions. Technical report AFWAL-TR-80-2090/PI, Air Force Wright Aeronautical Laboratories / Purdue University, June 1981.

[88] Tsuchiya T., Murthy S.N.B. Effect of water on axial flow compressors – Part II Computational program.

Technical report AFWAL-TR-80-2090/PII, Air Force Wright Aeronautical Laboratories / Purdue University, June 1981.

[89] Murthy S.N.B., Tsuchiya T., Ehresman C.M. et al. Water ingestion into axial flow compressors – Part III Experimental results and discussion. Technical report AFWAL-TR-80-2090/PIII, Air Force Wright Aeronautical Laboratories / Purdue University, October 1981.

[90] Kurzke J. GasTurb11 – Design and Off-Design Performance of Gas Turbines (Manual). Also see gasturb.de by J. Kurzke, status 04/01/2012.

[91] Walsh P.P., Fletcher P. Gas Turbine Performance 2nd Edition. Blackwell Science Ltd (Blackwell Publishing) Oxford, 2004.

[92] Tsuchiya T., Murthy S.N.B. Axial-compressor flow distortion with water ingestion. AIAA-83-0004. AIAA 21st Aerospace Sciences Meeting, January 1983.

[93] European Aviation Safety Agency (EASA) – Certification Specifications for Engines CS-E Amendment 1, 10th Dec 2007. Ingestion of Rain and Hail (CS-E 790) p. 64-66. (equivalent to FAR 33.78)

[94] Zheng Q., Wang G., Sun Y. et al. Experiment on water spraying internal cooling within the compressor of gas turbine. 97-GT-158. International Gas Turbine and Aeroengine Congress and Exhibition (ASME), June 1997.

[95] Ellisman C. Flight investigation of thrust augmentation of a turbojet engine by water-alcohol injection. National Advisory Committee of Aeronautics NACA RM E7G29, September 1947.

[96] Ross A.O., Huppert M.C. Analytical determination of effect of water injection on power output of turbine-propeller engine. National Advisory Committee of Aeronautics NACA RM E9H17, November 1949.

[97] Baron B., Dowman H.W., Dackis W.C. Experimental investigation of thrust augmentation of axial-flow-type 4000-pound-thrust turbojet engine by water and alcohol injection at compressor inlet. National Advisory Committee of Aeronautics NACA RM E7K14, July 1948. See also some other research publications on naca.cranfield.ac.uk, status 04/01/2012.

[98] Spurk J.H., Aksel N. Strömungslehre – Einführung in die Theorie der Strömungen 6.Aufl. Springer Verlag Berlin, 2006 (in German).

[99] Frohn A., Roth N. Dynamics of droplets. Springer Verlag Berlin, 2000.

[100] Matysiak A. Euler-Lagrange Verfahren zur Simulation tropfenbeladener Strömung in einem Verdichtergitter. Diss. UniBw Hamburg, April 2007 (in German).

[101] Groll R., Jakirlić S., Tropea C. Comparative study of Euler/Euler and Euler/Lagrange approaches simulating evaporation in a turbulent gas-liquid flow. Int. J. Numeric. Meth. Fluids 59:873-906, 2009.

[102] Eisfeld T. Experimentelle Untersuchung der Aerodynamik einer mit Wassertropfen beladenen Luftströmung in einem ebenen Verdichtergitter. Diss. UniBw Hamburg, November 2011 (in German).

[a] DoD Photography by Photographer's Mate Airman Ryan O'Connor, U.S. Navy, February 22nd, 2005 (released).

APPENDIX

EVALUATION KEY

Evaluation in TAB 1. is done as to match the individual methods' modeling and performance qualities to the 'dot groups' below to the best of knowledge. It is emphasized that method assignment to a certain 'dot group' does not mean that strictly all criteria as stated on the right side are fulfilled, though most of them are met for the specific model attribute. Generally speaking, in terms of engine performance calculations it is the aim to have an advanced, detailed numerical wet compression modeling strategy with high usability that is based on a comprehensive set of validation data. This means that 'best-possible' wet compression modeling would be attributed two 'dots' concerning model complexity, three 'dots' for usability and three 'dots' concerning model validation to experimental data.

Physical and mathematical *complexity*

●●● In-depth modeling of two-phase flow effects in combination with highly-iterative mathematical solution procedures and/or complex coupling with a stage-stacking scheme, possibly separation into rotor and stator calculation or large and small droplet modes.

●● Advanced modeling of two-phase flow effects in combination with iterative mathematical solution procedures and/or coupling with a stage-stacking scheme.

● Basic modeling of two-phase flow effects - focus on selected impacts – with a low number of iterative mathematical solution procedures, no coupling with a stage-stacking scheme.

Usability

●●● High usability for engine performance calculations with direct evaluation of overall compressor maps, minimum geometry data needed as input, method performs at low added computational costs, minimum time input for setting up a performance model.

●● Medium usability for engine performance calculations with direct evaluation of overall compressor maps, some geometry data needed as input (approximations), method performs at medium computational expenses, time input for setting up a performance model.

● Reduced usability for engine performance calculations with direct evaluation of overall compressor maps, detailed (blade) geometry data needed as input, computationally expensive solution procedures, increased time input for setting up a performance model, possibly gaps in method documentation in open literature.

Validity

●●● Comprehensive validation database, several gas turbine operating points or different power settings with wet conditions, varying injection rates, droplet size distributions, pos. measurements along the compressor gas path, modeling of individual two-phase flow effects assessed and supplemented by experimental data.

●● Medium validation database, some gas turbine operating points or different power settings with wet conditions, some varied boundary parameters, pot. two-phase flow effects modeling assumptions supplemented by experimental data.

● Low or no validation database indicated, modeling relies mainly on assumptions, expected trends with wet operation covered.

<u>TAB 1.</u> **Author and References / Model Sub-Category**	**Validation to Experimental Data / Calculated System**	**Remarks**	**Evaluation**
I Sexton, Urbach, Knauss (Virginia Military Institute Lexington, VA / Carderock Division of the Naval Surface Warfare Center Annapolis, MD) 1998 [59]	No validation to experimental data, compressor map analysis, shaft power and thermal efficiency of gas turbine cycle / GE LM2500, approximated 16-stage compressor with NACA blade profiles to derive compressor map for wet engine performance calculation	Focus wet compression in terms of water injection, comparison of wet system performance to dry system performance, simplified engine model, wet compressor calculation based on enthalpy-entropy-formulations including non-equilibrium evaporation model (equal-enthalpy-steps), no velocity slip, blade geometry input needed	Complexity ●● Usability ● Validity ●
I Sexton, Sexton (Bechtel Power Corporation Frederick, MD / Virginia Military Institute Lexington, VA) 2003 [61]	No validation to experimental data, parametric study for various inlet conditions and fogging conditions / GE LM2500, approximated 16-stage compressor with NACA blade profiles (see line above)	Focus wet compression in terms of water injection, update model to [59] with added engine inlet duct calculation and particle-size distribution input, no velocity slip, blade geometry input needed	Complexity ●● Usability ● Validity ●
I White, Meacock (Cambridge University) 2003 [60]	No validation to experimental data, comparison of compressor calculation results only / generic 12-stage compressor	Focus wet compression in terms of water injection, non-equilibrium evaporation modeling with assumption of constant polytropic efficiency, compressor calculation not incorporated into an engine performance calculation, general trends, iterative approach dry step followed by an evaporation step, does include differentiation in aerodynamic and thermodynamic entropy rise, no velocity slip, blade geometry input needed	Complexity ●● Usability ● Validity ●
I Abdelwahab (Praxair Inc. Tonawanda, NY) 2006 [62]	No validation to experimental data, comparison of compressor calculation results only / one-stage centrifugal flow compressor and 4-stage air separation plant compressor (Main Air Compressor)	Focus wet compression in terms of water injection, non-equilibrium model, analysis of polytropic efficiency change, compressor calculation not incorporated into an engine performance calculation, general trends, stage enthalpy step coupled with evaporation model and entropy production model, no velocity slip, blade geometry input needed	Complexity ●● Usability ● Validity ●
I Kim, Perez-Blanco (National Institute of Technology Geongbuk, Korea / The Pennsylvania State University, University Park, PA) 2006 [63]	No validation to experimental data, comparison of compressor calculation results only / generic compressor according to [60]	Focus wet compression in terms of water injection, adapted version of White & Meacocks' 2003 wet compression model, adapted numerical treatment and different evaporation time modeling, no velocity slip, blade geometry input needed	Complexity ●● Usability ● Validity ●
I White, Meacock (Cambridge University) 2010 [40]	No validation to experimental data, comparison of compressor calculation results only / generic 12-stage compressor	Updated version of the 2003 wet compression model, incorporation of quasi-3D-droplet motion and thus velocity slip, droplet collision and splashing, film formation on casing and hub as well as film evaporation modeling, blade geometry input needed	Complexity ●●● Usability ● Validity ●

II Nikolaidis (Cranfield University) 2008 [69]	Validation of results to Day, Freeman, Williams [31], Warwick [30], sub-models validated to individual scientific literature sources / J79-GE-8 turbojet engine	CFD-adapted performance calculation accounting for virtual blade-profile thickening, surface roughness change and relative motion between film and gas stream, for one speed of rotation and liquid water rate (at given droplet properties) two CFD simulations of the compressor geometry are needed, simplification in form of simulating one axial flow compressor stage with variable inlet stator geometry, two-phase flow modeling and CFD input are sources of model uncertainty, derivation of correction factors for engine performance calculations based on compressor maps, combustor performance change considered, detailed compressor geometry input needed for CFD-mesh-generation, focus is wet compression in terms of water ingestion (rain), negligible rate of evaporation	Complexity ●●● Usability ● Validity ●●
III Kurzke (GasTurb11) 2007 [90]	No information on validation to experimental data provided, use of commercial software in industrial practice	Calculation in equal-enthalpy-steps based on given compressor map, two-step-approach (dry calculation followed by an evaporation step), assumption of constant 'dry' polytropic efficiency for wet compression, all liquid water evaporated in front of the combustor following a compressor, does not cover centrifugal effects and droplet-wall-interaction, does include consideration of aero-mechanical losses with an expression from [91], no detailed geometry input needed	Complexity ● / ●● Usability ●●● Validity ●
IV Bagnoli, Bianchi, Melino et al. (University of Bologna / University of Ferrara) 2006 [45]	Validation of results to Utamura, Kuwahara, Murata et al. [17] (engine power variation and cycle efficiency vs. injected liquid water mass flow), comparison of some code output to [60] / GE F9E, approximated 17-stage compressor to match with dry compressor performance	Focus is wet compression in terms of water injection, stage-stacking-scheme with phi-psi-stage modeling (shape factor), non-equilibrium evaporation model referring to [54], gas temperature change due to evaporation in single stage virtually divided up into temperature adjustments before and after compression (factor), 'dry' compression (and thus sensible heat transfer to liquid water phase) separated from latent heat transfer, stage calculation iteration until evaporated liquid mass from non-equilibrium model converges	Complexity ●● Usability ● / ●● Validity ●●
IV Sanaye,Rezazadeh, Aghazeynali et al. (Iran University of Science and Technology / Tehran Regional Electric Company) 2006 [46]	Validation of results to Bhargava, Meher-Homji [19] / GE 9171E, approximated 17-stage compressor to match with gas turbine system's dry performance	Focus is wet compression in terms of water injection, non-equilibrium evaporation based on model given in [29], assumption of constant polytropic stage efficiency, compressor off-design calculation with phi-psi-stage modeling (shape factor) according to [85], adjustment of gas state due to evaporation taking into account an entropy expression, no velocity slip	Complexity ●● Usability ● / ●● Validity ●●
IV Roumeliotis, Mathioudakis (National Technical University of Athens) 2005 / 2010 [44,50]	No validation to experimental data, comparison of single-shaft-engine output using generic and 15-stage Tornado-engine compressor	Focus is wet compression in terms of water injection (up to 2% overspray stated), compressor map evaluation based on map zooming procedure with forward and backward stage-stacking-scheme evaluation with phi-psi-stage maps, two-phase flow modeling taking into account enthalpy and entropy formulations, no velocity slip, general trends	Complexity ●●● Usability ● Validity ●

V Tsuchiya, Murthy, Leonardo (Purdue University, West Lafayette, IN / Aero Propulsion Laboratory at Wright-Patterson Air Force Base, OH / NASA Lewis Research Center, OH) 1982 [27,28,87-89] PURDU-WINCOF Code, PURDU-WICSTK Code	Validation of results to experimental data generated with a 6-stage axial research compressor derived from T63 engine (axial part of Allison 250 compressor) / T63 research compressor	Focus is wet compression in terms of water ingestion (rain), compressor tested for mean droplet sizes 90 microns and 600 microns up to 8% liquid water mass fraction in entry flow, stage-stacking scheme (blade-wise calculation) employing analysis of velocity triangles, exact blade geometry input needed, code differentiates in large-size and small-size droplet class calculation, two main calculation steps: 1. calculation of blade row performance non-evaporating two-phase flow 2. blade exit state correction including e.g. centrifugal force, droplet-blade interaction, heat and mass transfer, use of non-equilibrium evaporation model, droplet velocity slip for large droplets considered based on quasi-3D droplet motion, no significant evaporation observed in the tested compressor (rain ingestion test case)	Complexity ●●● Usability ● Validity ●● / ●●●
V Zhluktov, Bram, De Ruyck (Vrije Universiteit Brussel) 2001 [36]	No validation to experimental data, parametric study / generic compressor geometry based on Allison 501-KB7 compression system, 1-stage booster (DFVLR) and 14-stage core compressor (BBC/SULZER)	Focus is wet compression in terms of water injection, correlations for incidence and deviation angles as well as pressure losses, complex models for droplet-wall interaction including splashing, film formation and boundary layer effects, quasi-3D droplet motion for the calculation of liquid mass flow to the casing and droplet-blade interaction (trial particles), approximated compressor geometry, general trends, analyses of the progress of evaporation along the axial length of the compressor, stage-stacking scheme (blade-wise calculation)	Complexity ●●● Usability ● Validity ●
V Hale, Klepper, Hurwitz (Aerospace Testing Alliance, Arnold Air Force Base, TN / Pratt & Whitney Aircraft Engines, East Hartford, CT) 2005 [4] AEDC MLC/MPC	Validation of results to experimental data generated with an 8-stage P & W - MAN Turbo FT8 engine low-pressure compressor / FT8 LPC derived from JT8D aircraft engine	Focus is wet compression in terms of water injection, calculation is performed as stage-stacking scheme (blade-wise) in two steps, an initial dry calculation is modified by the multi-phase code to account for heat and mass transfer, correlations, evaluation of test data incorporates modeling of the impact of the test cell with respect to pre-evaporation / condensation in front of the tested engine's inlet plane, larger discrepancies between code output and experimental data, additional fan performance analysis, parametric study, exact geometry input needed, equilibrium evaporation model	Complexity ●● / ●●● Usability ● Validity ● / ●●
V Matz, Kappis, Cataldi et al. (ALSTOM Switzerland Ltd, Baden) 2004 / 2008 [21,38,65]	Validation of results to experimental data gained at ALSTOM's test center in Switzerland and at customer sites / 50 Hz and 60 Hz ALSTOM gas turbine units (resp. GT24/GT26 with ALFog high fogging system)	Calculation scheme based on analysis of velocity triangles, detailed calculation method info not published [65], does include e.g. non-equilibrium evaporation model, droplet-wall interaction, splashing, film evaporation and aerodynamic flow angle correction, adjusted bleed calculation to account for bleed pressure change along the compressor geometry, wet compression model tuned to reference GT data set, explicit validation of pressure distribution levels between compressor stages (load-shift and inter-stage analysis), relative droplet motion along the meridional coordinate only, geometry input needed (flow angle deviation)	Complexity ●● / ●●● Usability ● / ●● Validity ●● / ●●●
V Wang, Khan (ECCC, University of New Orleans, LA) 2008 / 2009 [47-49]	No validation to experimental data, parametric study / generic 8-stage compressor in gas turbine engine cycle	Focus is wet compression in terms of water injection, calculation is performed as stage-stacking scheme (blade-wise), comparison of non-equilibrium and equilibrium evaporation modeling, highly-iterative procedure, see hints on specific work of compression and phi-psi-stage modeling (shape factor), exact blade geometry input needed, assumption of constant equivalent polytropic efficiency, no slip for equilibrium model, 10% velocity slip for non-equilibrium evaporation assumed [66]	Complexity ●●● Usability ● Validity ●

V Montalvo-Catano, O'Brien (Virginia Polytechnic Institute and State University, Blacksburg, VA) 2011 [51]	Validation of results to experimental data, analysis of the impacts of evaporation model selection / GE LM2500, approximated 16-stage compressor with NACA blade profiles as presented in [61] to match with gas turbine system's dry performance, see ref. Jolly S., Cloyd S., Hinrichs J. Wet compression adds power, flexibility to aeroderivative GTs. Power, Vol.149 No.4 p. 52 – 57, May 2005	Focus is wet compression in terms of water injection, calculation is performed as stage-stacking scheme (blade-wise), evaporation is assumed to occur in the rotor sections only, gas turbine performance output is significantly changed by evaporation model selection, droplet-size distribution assumed to be between 3 microns and 35 microns (spec. evaporation model type rain vs. water injection), paper focuses on droplet evaporation, no modeling of side-effects such as droplet-wall interaction or film formation	Complexity ●● / ●●● Usability ● Validity ● / ●●

TAB 1. Versions of detailed numerical wet compression models in scientific literature (status 2012) – Roman letters and distinct line colors show differentiation into sub-model categories

TAB 2. **Author and References**	**Tested System**	**Remarks**
Baron, Dowman, Dackis (NACA, Flight Propulsion Research Laboratory, Cleveland, OH) 1948 [97]	GE/Allison J35 (TG-180) with 11-stage axial flow compressor	Performed water-alcohol-injection test program at sea-level conditions with three distinct liquid phase injection systems at the compressor inlet, varying injection rates and alcohol concentrations - including pure water flow, graphs for static-thrust-augmentation, fuel flow, compressor discharge pressure, air flow, compressor discharge temperature as well as radial temperature profiles at compressor exit for constant rotor speed and exhaust gas temperature, hints on centrifugal effects and casing film formation for higher liquid flows, info rubbing from 4^{th} to 9^{th} stage for two injection system cases (film evaporation), discrepancy between thrust augmentation and characteristics of the injection spray (evaporation rate dependency on droplet temperature, gas temperature, initial diameter and ambient relative humidity), liquid water injection up to approx. 6.5% liquid to (humid) air by mass
Wetzel, Jennings (Northwestern Technological Institute, Evanston, IL) 1949 [11]	War-time development program test compressor with 11 blade rows	Compressor tested while operating at constant speeds, nozzle pressure ranges from 150 psi to 300 psi, liquid water injection up to 1% liquid to (humid) air by mass, in most test cases full evaporation along the compressor, compressor has wet walls at some operating conditions, reduction of specific compressor work, no evidence of erosion (short testing time)
Hamrick, Beede (NACA, Lewis Flight Propulsion Laboratory, Cleveland, OH) 1952 [12]	Several centrifugal flow compressor impellers	Centrifugal compressor tested while operating at constant speeds, highlights impacts of mineralized water, special instrumentation against droplet impingement, compressor tested up to 5.5% liquid-to-air by mass, recognized total pressure ratio increase, but also gain of required specific compressor work wet vs. dry operation (aero-mechanical performance changes counteract thermodynamic benefits from wet compression)
Warwick (U.S. Naval Air Turbine Test Station, Trenton, NJ) 1963 [30]	J79-GE-8 altitude water ingestion tests ATF-like testing station	Data for compressor discharge temperature, compressor discharge pressure (pressure ratio), flame-out limits, compressor casing temperatures, clearances, time-history of flame-out event, fuel schedule, different water injection nozzle configurations applied, maximum water injection up to 8% liquid-to-air-by mass, changing the injection pattern for the liquid water stream had significant impact on engine performance and flame-out limits (spray ring vs. large-hole wall spray nozzles)
Hill (Massachusetts Institute of Technology, Cambridge, MA) 1963 [52]	Two unspecified GE turbo-shaft engines of different size and pressure ratio	Data points for compressor work reduction and total mass flow increase due to wet compression, maximum liquid water injection approx. 1% liquid-to-dry-air by mass (coolant flow liquid water, standard inlet temperature as input to evaporation variable)
Jahnke (Technische Universität Dresden) 1968 [15]	3-stage axial flow test compressor with variable stator vane at the compressor inlet	Test data for operation at constant speeds, gas temperature reduction, un-evaporated liquid water at the compressor casing behind the stages, compressor exit pressure reduction with wet compression, increase in total mass flow, stage work distribution affected by evaporating liquid phase (load-shift to the rear stages), reduced compressor efficiency with wet compression (added aero-mechanical losses, as stated in document), change of compressor map similar to [31,34], water injection up to 1.4% liquid-to-total-air by mass, some hints on instrumentation

Tsuchiya, Murthy (Purdue University, West Lafayette, IN / Aero Propulsion Laboratory at Wright-Patterson Air Force Base, OH / NASA Lewis Research Center, OH) 1982 [27]	6-stage axial flow research compressor derived from T63 engine (axial part of Allison 250 compressor)	Compressor performance tested for mean droplet sizes 90 microns and 600 microns up to 8% liquid water mass fraction in entry flow, data points for mass flow rates, total temperature ratios at constant speeds, blockage-effect indicated, accumulation of liquid phase at the compressor casing, no significant evaporation detected in tests (focus is on the impacts of rain ingestion)
Weichert (Berlin Institute of Technology) 1982 [16]	Single-stage radial flow compressor (in Turbomeca Marbore VI engine) with a pressure ratio of 3.5	Data points taken at constant reduced speeds, shift of operating line to lower pressure ratios and lower reduced mass flows with wet compression, some hints on instrumentation and emission reduction, optical measurement concept, aero-mechanical losses counteracting thermodynamic benefits from evaporation (low evaporation here), favoured droplet sizes in the range of 20 microns, specific water injection rate corresponds to that droplet size (system-specific/nozzle-specific rate), approx. injected 3% liquid-to-air by mass
Utamura, Kuwahara, Murata et al. (Hitachi Ltd.) 1999 [17]	Simple cycle GE F9E gas turbine (MAT cycle) with 17-stage axial flow compressor	Data concerning engine power boost, thermal cycle efficiency, fuel flow, compressor efficiency with wet compression, time-history following injection of liquid phase, injection up to 0.7% liquid-to-air by mass, some hints on compressor washing and long-term effects, favoured droplet sizes in the range of 10 microns
Bhargava, Meher-Homji (Universal Ensco Inc., Houston, TX / Mee Industries Inc., Monrovia, CA) 2002 [19]	Trend-indicating plots for 67 gas turbines, net output, overspray and inlet fogging, field data for GE 9171E simple cycle gas turbine	Most relevant field data such as e.g. compressor discharge temperature, compressor discharge pressure, air mass flow, compressor work, efficiency, heat rate and net output
Sereda, Gel'medov, Muntyanov (Moscow Mechanical Engineering Enterprise Salyut / PI Baranov Central Institute of Aircraft Engine Building, Moscow) 2004 [82]	14-stage Lyulka AL-21F3 axial flow compressor	Some experimental data concerning transformation of temperature and pressure fields in a compressor flow path due to moisture evaporation, non-uniformities, water injection up to 2.5% liquid-to-air by mass, water droplet sizes are in the range of 20 microns, compressor characteristics and stage performance, impact of injection pattern on compressor performance changes
Hale, Klepper, Hurwitz (Aerospace Testing Alliance, Arnold Air Force Base, TN / Pratt & Whitney Aircraft Engines, East Hartford, CT) 2005 [4]	8-stage P & W – MAN Turbo FT8 engine low-pressure compressor / FT8 LPC derived from JT8D aircraft engine	Some data points for static pressure rise in tested LPC with wet compression, varying injected amount of liquid water
Day, Freeman, Williams (Whittle Laboratory, Cambridge / University of Bath) 2005 [31]	4-stage laboratory compressor with inlet guide vane	Experiments with four different types of water injection nozzles (51 microns to 1.5 mm diameter straight-jet type injection), compressor pressure rise characteristics with up to 17% injected liquid-to-air by mass, below 4% injected water has negligible effect on pressure rise characteristic (compare to certification testing guidelines [93]), analysis of droplet size impact, torque increase, stall susceptibility and stability aspects, some hints on impacts of centrifugation and liquid flow pattern along the compressor, film formation, flow velocity distribution changes along the compressor blade geometries, aerodynamic efficiency change
Roumeliotis, Mathioudakis (National Technical University of Athens) 2006 [67]	One-stage axial flow test compressor with a pressure ratio of 1.57, tests conducted up to 50% nominal speed (max. speed 18000 rpm)	Tests for wet compressor stage operation at low speeds (idle-descent), water injected up to 2% liquid-to-air by mass, impact of wet compression on stage map (higher pressure ratio, slightly higher mass flow), low evaporation test case, no significant changes in pressure rise coefficient and flow coefficient, no impact on stall margin, no significant change of blade aerodynamics, increase in compressor torque and thus consumed compressor power for constant speed operation

Williams (University of Oxford) 2008 [34]	4-stage laboratory compressor with inlet guide vane as in 2005 [31]	Supplementary work to previous paper on water ingestion, in-depth analysis of the impacts of water injection patterns on the compressor characteristics, water injected up to 20.6 % liquid-to-air by mass, dynamic similarity, analysis of wet compression effects on components following the compressor, whole engine operational aspects, some additional hints on stability, surge line and importance of the tip section
Matz, Kappis, Cataldi et al. (ALSTOM Switzerland Ltd, Baden a. oth.) 2008 [38], also see [21]	50 Hz and 60 Hz ALSTOM gas turbine units (resp. GT24/GT26 with ALFog high fogging system)	Generation of calibration data for wet compression modeling, gas turbine inlet gas mass flow increase for 50 Hz unit at 80% relative humidity, decrease in gas mass flow at 100% relative humidity with wet compression (condensation, coagulation, liquid flow pattern, blockage-effect), measurement of some bleed supply pressures along the compressor length (good match to code output, aim in-depth analysis of wet compression stage performance, liquid phase physics) for tested 50 Hz units, some data concerning radial temperature distribution and consumed compressor power, liquid water penetration check (stain, erosion) for 60 Hz unit with photographs
Bettocchi, Morini, Pinelli et al. (University of Ferrara / E.R.S.E. SpA SSG, Milano) 2010 [64]	WECOS system: Allison 250-C18 turbo-shaft engine, six axial stages plus one centrifugal stage	Some preliminary results gained from the wet compression test facility, SMD from 16 microns to 34 microns at 50 bar injection nozzle pressure, operation at constant speeds with one or two water injectors, operation with up to 1.1% liquid-to-air by mass, measured compressor performance maps, increased torque, time-history for testing, compressor casing deformation observed due to injector opening (casing shrinkage), focus of test facility research work will be on long-term operation and blade erosion

TAB 2. Experimental data on wet compression that have formed the basis for model validation and effect analysis in the past – lines in gray represent data referring to water ingestion, lines in white mark data concerning water injection applications