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Multi-energy Microgrid Planning Considering Heat Flow Dynamics

Miguel Heleno, *Member, IEEE* and Zhengwei Ren

Abstract—This paper presents a multi-energy microgrid optimal planning method, considering the intra-hour dynamics of the heating system as constraints of the energy dispatch, and consequently of the sizing of the Combined Heat and Power units. The fixed-mass flow rate method and finite difference technique are applied to linearize the dynamic constraints. Moreover, to determine the values of the mass flow rates, a critical scenario representing the slowest heat dynamic process is put forward. Finally, a realistic microgrid case study is presented to illustrate the effect of considering the heat flow dynamics in the overall solution.

Index Terms—Microgrids, Multi-energy, Planning, Combined Heat and Power.

NOMENCLATURE

A. Sets and Indices

t	Hourly time period (h).
τ	Minute time period (min).
i, j	Start and end nodes of the pipe.
Ω_j	The set of nodes connected to j .
L_p	A certain loop in the heating system.
Ω_{L_p}	The set of all loops in the heating system.
Π_j	The set of pipes connected to node j .

B. Parameters

CE_t	Electricity price (\$/kWh) at time t .
CG_t	Gas price (\$/kWh) at time t .
CI	Annualized capital investment.
OM	Annual operation and maintenance cost.
η_H	Heat efficiency of the CHP.
$P_{j,t}^d$	Active electricity load at node j in time t .
$Q_{j,t}^d$	Reactive electricity load at node j in time t .
r_{ij}, x_{ij}	Resistance and reactance of line ij .
$P_{i,t}^{b,max}$	Limit of active power purchases.
$Q_{i,t}^{b,[min,max]}$	Limit of reactive power purchases.
$U^{[min,max]}$	Bounds of the squared voltage.
I^{max}	Maximum of the squared current.
$\bar{m}_{j,t}$	Injected mass flow rate at node j in time t .
$\bar{m}_{ij,t}$	Mass flow rate in pipe ij at t .
$\bar{m}_{l,t}$	Mass flow rate in pipe l at t .
$T_i^{sup,min}$	Minimum supply temperature at source.
$T_j^{sup,min}$	Minimum supply temperature at load.
k_l	Resistance coefficient of pipe l .
$H_{j,t}^d$	Heat load at node j in time t .
c_w	Water specific heat.

α_{H2E}	Heat to electricity ratio of CHP unit.
ρ	Density of the water (kg/m^3).
R	Pipe thermal resistance per length (mK/W).
λ_{ij}	Overall heat transfer coefficient of the pipe $i - j$ per unit length (W/mk).
k_w	Thermal conductivity of water (W/mK).
A	Cross-sectional area of the pipe (m^2).
L_{ij}	Length of the pipe ij (m).
φ_{ij}	Steady-state heat loss factor for pipe ij .
$\Delta\tau$	Intra-hour time step (min).
$\Delta T^{ex,max}$	Heat exchangers maximum temperature range.

C. Variables

P_t^b	Electricity bought from external grid (kWh).
H_t^g	Heat generation from the CHP unit (kWh).
S^H	Heat capacity of CHP unit (kW).
$P_{ij,t}$	Active power flow on line ij .
$Q_{ij,t}$	Reactive power flow on line ij .
$P_{i,t}^b$	Active power bought from the external grid at node i in time t .
$Q_{i,t}^b$	Reactive power bought from the external grid at node i in time t .
$P_{i,t}^g$	Active power generation of CHP unit at node i in time t .
$Q_{i,t}^g$	Reactive power output of CHP unit at node i in time t .
$I_{ij,t}$	Squared current of line ij at time t .
$U_{i,t}$	Squared voltage at node i in time t .
$T_{l,t}^{sup}$	The supply temperature of pipe l at node j .
$T_{l,t}^{ret}$	The return temperature of pipe l at node j .
$T_{i,t}^{sup}$	The supply temperature at node i (for pipe ij).
$T_{j,t}^{sup}$	The supply temperature at node j (for pipe ij).
$T_{i,t}^{ret}$	The return temperature at node i (for pipe ij).
$T_{j,t}^{ret}$	The return temperature at node j (for pipe ij).
$T_{a,t}$	The ambient temperature at time t .
$\Delta T^{so,max}$	The maximum temperature difference at source.
N_S	The number of heat exchangers connected to heat source (integer variable).
$H_{i,t}^g$	CHP heat generation at node i in time t .

I. INTRODUCTION

MICROGRIDS are a flexible way of deploying and integrating distributed energy resources (DERs) into local energy systems. In particular, when comprising different energy vectors, multi-energy microgrids (MEMGs) introduce a powerful layer of coordination across the entire chain of

local energy distribution with significant potential to decrease energy losses and operation costs.

Planning and design of microgrids aims at determining the optimal portfolio, size and location of DERs for a microgrid network, in order to supply the heat, gas and electricity loads in a cost-effective and reliable manner. Previous studies have investigated the microgrid planning problem in various aspects, focusing on the distributed generation sizing and controls placement in a single microgrid site [1]–[3] or in a group of networked microgrids [4]. For a single microgrid, the planning problem finds the optimal combination of renewable and conventional energy sources, considering the economic and environmental targets [2], [5]. For networked microgrids, the planning problem takes into account the operational benefits and challenges of a coordinated response of many interconnected microgrids in a distribution system [4].

Due to the complexity of the design of electricity microgrids, and despite some non-linear optimization approaches presented in the literature [6] [7], design tools used by the industry, such as DER-CAM [8] and Microgrid Design Toolkit [9], are based on mixed-integer linear programming (MILP) formulations. These formulations have evolved over the last decade to address new challenges of design, including the time resolution of the dispatch [10], the role of thermal loads in the optimal sizing [11], the need to account for internal network topology [12], the optimal operation of multiple energy vectors [12], [13], the N-1 security when planning microgrids in remote locations [14], or aspects related to resilience against natural disasters [5], [15], [16]. Additional research has focused on renewable generation uncertainty [17]–[20] via stochastic programming models that consider intra-day variability of renewable sources. Other sources of uncertainty, including load or market prices, are considered in [3] in a bi-level robust optimization model while in [21] a similar model is applied in the context of provisional microgrids to enable larger deployment of renewable distributed generation. In [22] a scenario-based approach to address uncertainty in this context of multi-node isolated microgrids was proposed.

Multi-energy microgrids are an effective way to enhance energy efficiency. In the design of these systems, similar models and methods used in electricity microgrids are extended to other energy carriers, such as heat, cooling and gas [22]–[24]. However, the combination of different resources, in particular the detailed model of the coupling characteristics among energy networks, is a new challenges to be solved [23], [24]. In fact, although recent developments on microgrids planning comprise advances in multi-energy modeling, they are still primarily focused on electricity and rely on relatively simple representations of the heating systems, for example considering energy losses proportional to the energy flow [12], [14] or as a linear function of the distances between the nodes [25]. Physical proprieties, such as temperatures, mass flow and comfort of the building occupants are either not considered or modeled by non-linear equations [26], [27] with limited applicability in planning problems.

Only in the context of larger multi-energy systems, i.e. outside the microgrid domain, it is possible to find steady-state models of the heat flow, in particular linearized around fixed

mass flow rate (fixed-MFR) [28], [29]. These models, together with power flow equations, can impose ramping constraints to the Combined Heat and Power (CHP) generators [30] providing more realistic operational constraints to the planning problem. However, a simple adaptation of these models to the MEMG domain may not be accurate enough, as microgrids - with smaller sizes and more aggressive dispatch policies - can entail significant inter-temporal variability in the demand and heat flows. Therefore, representing the dynamics of the heating system (at intra-hour level) in microgrids becomes critical to the security and comfort of the consumers. These dynamics are composed of hydraulic processes (that take seconds to minutes) [31], but mostly of thermal processes (that can take several minutes to achieve the steady state) [32], and whose effect may affect the feasibility of the energy dispatch policies and consequently the MEMG planning. To address this gap, this paper presents a MEMG optimal planning method considering the intra-hour dynamics of the heating system [33], [34] as constraints of the energy dispatch and the capacity sizing of the microgrid heat sources, formulating an overall MEMG planning model as a mixed-integer SOCP problem [35], [36]. The contributions of this paper are threefold:

- First, we propose a new MEMG planning model that considers the intra-hour heat propagation as a constraint of the hourly steady-state heat flow. By applying the finite difference technique [34] and the fixed-MFR method, we represent the dynamics of the heat flow as boundaries of the steady-state operation and show the impact on the microgrid design.
- Second, the critical conditions of the fixed-MFR method are explored to allow a secure planning in a linearized form. The MFR is chosen to satisfy all hydraulic constraints of the heat model. Different from [28], this scenario defines the minimum allowable MFRs for heat prorogation and exchange, capturing the longest time to respond to heat load variations while keeping the reliability of the hydraulic system.
- Third, this model is applied to a realistic microgrid planning problem and impact of the heat dynamic models is evaluated together with the performance of the fixed-MFR conditions.

The rest of the paper is organized as follows: Section II provides the steady-state MEMG planning model; section III presents the multi-timescale heat dynamic constraints and discusses the fixed-MFR critical conditions; section IV shows results while Section V highlights the conclusion.

II. STEADY STATE PROBLEM FORMULATION

This section presents the fundamental MEMG planning model, considering steady-state representations of both heat and electricity systems. Here, we present the model with the DER technologies that are relevant to the methodology proposed. Expanding this model to accommodate additional DERs is a straightforward exercise and a direct application of comprehensive methods in the literature, such as [12], [14]. Therefore, for the sake of clarity of the contributions, this paper will focus only on coupling technologies.

A. Objective Function

The planning model aims to find the optimal capacity of the generators, the supply and return temperatures, and the number of heat exchangers that minimize the total investment and operation cost of the microgrid. Equation (1) divides this objective in three parts: cost of electricity purchased from the external grid ($\sum_t CE_t \cdot P_t^b$); cost of natural gas consumed by the gas-fired CHP unit ($\sum_t CG_t \cdot \eta_H^{-1} \cdot H_t^g$); and the annualized fixed investment of the CHP unit $(CI + OM) \cdot \alpha_{H2E}^{-1} \cdot S^H$.

$$\min \sum_t^{8760} (CE_t \cdot P_t^b + CG_t \cdot \eta_H^{-1} \cdot H_t^g) + (CI + OM) \cdot \alpha_{H2E}^{-1} \cdot S^H \quad (1)$$

B. Electricity System Constraints

To describe the power flow constraints, the branch flow model is adopted. Equations (2) to (3) describe the active and reactive power flow balance at node j . Equations (4) to (5) refer to the power balance at node i . Equations (6) describes the relationship between the voltages of the start and end node of the line $i - j$. The relation among line current, line power, and bus voltages are described by (7). The limits for active and reactive power purchased from the external grid are given by (8) and (9), and the limits of bus voltage and line power are given by (10) and (11).

$$P_{j,t}^d = \sum_{\{\cdot\} \in \Omega_j} (P_{\{\cdot\}j,t} - I_{\{\cdot\}j,t} \cdot r_{\{\cdot\}j}) \quad (2)$$

$$Q_{j,t}^d = \sum_{\{\cdot\} \in \Omega_j} (Q_{\{\cdot\}j,t} - I_{\{\cdot\}j,t} \cdot x_{\{\cdot\}j}) \quad (3)$$

$$\sum_{\{\cdot\} \in \Omega_i} P_{i\{\cdot\},t} = P_{i,t}^b + P_{i,t}^g \quad (4)$$

$$\sum_{\{\cdot\} \in \Omega_i} Q_{i\{\cdot\},t} = Q_{i,t}^b + Q_{i,t}^g \quad (5)$$

$$U_{j,t} = U_{i,t} - 2(r_{ij} \cdot P_{ij,t} + x_{ij} \cdot Q_{ij,t}) + I_{ij,t} \cdot (r_{ij}^2 + x_{ij}^2) \quad (6)$$

$$I_{ij,t} = ((P_{ij,t})^2 + (Q_{ij,t})^2) / U_{i,t} \quad (7)$$

$$P_{i,t}^{b,min} \leq P_{i,t}^b \leq P_{i,t}^{b,max} \quad (8)$$

$$Q_{i,t}^{b,min} \leq Q_{i,t}^b \leq Q_{i,t}^{b,max} \quad (9)$$

$$U_{i,t}^{min} \leq U_{i,t}, U_{j,t} \leq U_{i,t}^{max} \quad (10)$$

$$0 \leq I_{ij,t} \leq I_{ij,t}^{max} \quad (11)$$

To solve the branch flow model, the (7) is transformed into inequality constraint as (12) using the second-order cones, which results in the relaxed SOCP model (2)-(6), (8)-(12).

$$I_{ij,t} \cdot U_{i,t} \geq ((P_{ij,t})^2 + (Q_{ij,t})^2) \quad (12)$$

This relaxation is proven to give results with physical meaning when limited by the *DistFlow* equations [37], [38] and it has been successfully applied in the context of DER placement, e.g. [39]. Thus we consider $\tilde{U}_{i,t}, \tilde{U}_{j,t}, \tilde{P}_{ij,t}, \tilde{Q}_{ij,t}$ as ancillary variables of the *DistFlow* model, where $U_{0,t}$ and $\tilde{U}_{0,t}$ are the

squared voltage of slack buses in SOCP model and *DistFlow* model respectively.

$$U_{0,t} = \tilde{U}_{0,t} \quad (13)$$

$$\tilde{U}_{j,t} = \tilde{U}_{i,t} - 2(r_{ij} \cdot \tilde{P}_{ij,t} + x_{ij} \cdot \tilde{Q}_{ij,t}) \quad (14)$$

$$\tilde{U}_{i,t}, \tilde{U}_{j,t} \leq U^{max} \quad (15)$$

It should be noted that by integrating (13)-(15) into the original optimization model, the physical feasibility of the results can be guaranteed. In other words, when the optimization model has the characteristics of the assumptions in [36], the equality in constraint (12) can be satisfied, and the relaxation is exact. When the upper voltage is binding, equations (13)-(15) guarantee the physical meaning but the solution for the original problem can be sub-optimal.

C. Heating System Steady State Model

Fig.1 shows a schematic illustration of heating system with the heat source and load connected by the supply and return pipes. The water is first heated in the source and pressurized by the pump, then it flows into the supply pipe, and finally reaches the load side, carrying heat to satisfy the demand. Here, both supply and return pipes are denoted as pipe ij , with i representing the source node and j for the load node. m_{ij} is the mass flow rate within the pipes. Thus, for the supply pipe, the mass flows from node i to node j , while for return pipe, the mass flows from j to i . T_i^{sup} and T_j^{sup} refers to the temperatures in different pipes. T_i^{sup} is the supply temperature of water at the source outlet, and the T_j^{sup} is the supply temperature at the load inlet. T_i^{ret} is the return temperature at the load outlet, and T_j^{ret} is the return temperature of water at the source inlet.

Fig. 1. A schematic of the heating system.

The steady-state representation of the heating system comprises both hydraulic and thermal models. The first describes the mass flow in the pipes and depends on complex factors, such as the pressure of circulation pumps or the friction factor of the pipes. In this work, this process is simplified by adopting a feasible solution for the mass flow circulation that satisfies the the following constraints as in [31] :

$$\sum_{\{\cdot\} \in \Omega_j} \bar{m}_{\{\cdot\}j,t} = \bar{m}_{j,t} \quad (16)$$

$$\sum_{l \in L_P} k_l \cdot \bar{m}_{l,t} \cdot |\bar{m}_{l,t}| = 0 \quad (17)$$

where $\bar{m}_{\{\cdot\}j,t}, \bar{m}_{j,t}, \bar{m}_{l,t}$ are within a time step t . The values of $\bar{m}_{\{\cdot\}j,t}, \bar{m}_{j,t}, \bar{m}_{l,t}$ are determined by the proposed scenario,

which will be discussed later. Equation (16) represents the continuity of flow, namely, the mass flow that enters into a node equals to that leaves the node. Equation (17) is the loop pressure equation. The set of all loops in the heating system is denoted as Ω_{L_p} , and $L_p \in \Omega_{L_p}$ is a certain loop of heat pipes. So, the loop pressure equation indicates that the sum of the head losses on loop L_p equals to 0.

The thermal model is expressed by (18)-(28), which is given by [31]. It includes the temperature mixture equations in the pipes intersection (18) (19), the heat exchange at the load and source (20) (21) as well as the steady state loss functions (22) (23), which is described as a function of the temperatures at the start and end of each pipe. Equation (24) constraints the temperature difference at the loads, while (25) determines the capacity of the heat source, in terms of the heat exchangers required. For each node, the mass flow temperature in the supply pipe should be higher than that in the return pipe, as expressed in (26). The upper and lower bound for the mass flow temperature in the supply pipe should be controlled, as given by (27) (28). Noticing that water is used as the mass in the pipes, thus the upper temperature limit is set as 100°C.

$$\sum_{l \in \Pi_j} (\bar{m}_{l,t} \cdot T_{l,t}^{sup}) = 0 \quad (18)$$

$$\sum_{l \in \Pi_j} (\bar{m}_{l,t} \cdot T_{l,t}^{ret}) = 0 \quad (19)$$

$$H_{j,t}^d = c_w \cdot \bar{m}_{ij,t} \cdot (T_{j,t}^{sup} - T_{j,t}^{ret}) \quad (20)$$

$$H_{i,t}^g = c_w \cdot \bar{m}_{ij,t} \cdot (T_{i,t}^{sup} - T_{i,t}^{ret}) \quad (21)$$

$$T_{j,t}^{sup} - T_{a,t} = (T_{i,t}^{sup} - T_{a,t}) \cdot \varphi_{ij} \quad (22)$$

$$T_{i,t}^{ret} - T_{a,t} = (T_{j,t}^{ret} - T_{a,t}) \cdot \varphi_{ij} \quad (23)$$

$$T_{j,t}^{sup} - T_{j,t}^{ret} \leq \Delta T^{ex,max} \quad (24)$$

$$T_{i,t}^{sup} - T_{i,t}^{ret} \leq N_S \cdot \Delta T^{ex,max} \quad (25)$$

$$T_{a,t} \leq T_{i,t}^{ret} \leq T_{j,t}^{ret} \leq T_{j,t}^{sup} \leq T_{i,t}^{sup} \leq 100 \quad (26)$$

$$T_i^{sup,min} \leq T_{i,t}^{sup} \quad (27)$$

$$T_j^{sup,min} \leq T_{j,t}^{sup} \quad (28)$$

D. Coupling Units

The coupling unit adopted in this paper is the gas-fired CHP. Equation (29) presents the relation between the heat and electricity output by the α_{H2E} ratio, while (30) limits CHP output.

$$H_{i,t}^g = \alpha_{H2E} \cdot P_{i,t}^g \quad (29)$$

$$0 \leq H_{i,t}^g \leq S^H \quad (30)$$

III. INTEGRATING MULTI-TIMESCALE HEAT DYNAMIC CONSTRAINTS

This section integrates the intra-hour dynamics of the heat flow into the steady-state MEMG planning model.

A. Modelling the Multi-timescale Heat Dynamics

In order to model the multi-timescale characteristics of the heat dynamics, we consider a single pipe $i - j$, which can be described by the following equation [33]:

$$\begin{aligned} \rho c_w A \frac{\partial T(x, \tau)}{\partial \tau} + m_{ij}(x, \tau) \cdot c_w \frac{\partial T(x, \tau)}{\partial x} \\ = k_w A \frac{\partial^2 T(x, \tau)}{\partial x^2} + \frac{T_a - T(x, \tau)}{R} \end{aligned} \quad (31)$$

where the four terms of the equation refer, respectively, to the internal energy of the fluid, the enthalpy flux of the fluid, the heat conduction in the fluid, and the heat conduction from the ambient environment. However, assuming that the heat propagation takes significantly longer than the hydraulic dynamics processes, we can assume $m_{ij}(x, \tau)$ to be constant in t : $\bar{m}_{ij,t}$. The value and the impact of the $\bar{m}_{ij,t}$ in the results of the overall planning model is discussed later in this section. Also, the heat conduction within the fluid can be neglected [33]. Thus, we can rewrite (31) as:

$$\rho c_w A \frac{\partial T_{j,\tau}}{\partial \tau} + \bar{m}_{ij,t} \cdot c_w \frac{\partial T_{j,\tau}}{\partial x} = \frac{T_a - T_{j,\tau}}{R} \quad (32)$$

where $\frac{\partial T_{j,\tau}}{\partial \tau}$ is the partial derivative of mass flow temperature of node j at time τ , with respect to the time. $\frac{\partial T_{j,\tau}}{\partial x}$ is the partial derivative of mass flow temperature of node j at time τ , with respect to the length. Equation (32) is a nonlinear partial difference equation. To solve it, finite difference technique is applied to the partial derivatives as:

$$\frac{\partial T_{j,\tau}}{\partial x} = \frac{T_{j,\tau} - T_{i,\tau}}{L_{ij}} \quad (33)$$

$$\frac{\partial T_{j,\tau}}{\partial \tau} = \frac{T_{j,\tau} - T_{j,\tau-1}}{\Delta \tau} \quad (34)$$

where $\Delta \tau$ is the intra-hour time step. The relation between τ (minute-level) and t (hourly-level) is illustrated in Fig. 2. The hourly time period $t_0 + 1$ is divided into $\tau, \tau + 1, \tau + 2, \tau + 3, \dots$, with a fixed intra-hour time step $\Delta \tau$. Here, we divide one hour into four $\Delta \tau$. $\tau - 1$ is the last $\Delta \tau$ in t_0 . This means that the last temperature of t_0 in node j (T_{j,t_0}) can be written in τ as $T_{j,\tau-1}$. Analogously, for time $t_0 + 1$, T_{j,t_0+1} corresponds to $T_{j,\tau+3}$. By substituting (33) and (34) into (32) we obtain

Fig. 2. Multi-timescale of thermal dynamic process (a - Heat load in hour t ; b - Temperatures in minute τ)

the linearized heat dynamic constraints as:

$$\underbrace{\left(\frac{\rho c_w A}{\Delta \tau} + \frac{\bar{m}_{ij,t} c_w}{L_{ij}} + \frac{1}{R}\right)}_{a_1} T_{j,\tau} = \underbrace{\frac{\rho c_w A}{\Delta \tau}}_{a_2} T_{j,\tau-1} + \underbrace{\frac{\bar{m}_{ij,t} c_w}{L_{ij}}}_{a_3} T_{i,\tau} + \underbrace{\frac{T_a}{R}}_{a_4} \quad (35)$$

where the coefficients and the constant term in (35) are denoted by a_1 , a_2 , a_3 , and a_4 respectively. It should be mentioned that there are one supply pipe and one return pipe between node i and j . In this single pipe example, i is connected to the heat source, while j is connected to the heat load. So, (35) exists both for supply and return pipes (for return pipes, replace i, j by j, i to represent the direction of the mass flow).

In practice, the supply temperature at heat sources $T_{i,t}^{sup}$ and the return temperature at loads $T_{j,t}^{ret}$ can be controlled, and they can determine the values of the supply temperature at loads $T_{j,t}^{sup}$ and the return temperature at sources $T_{i,t}^{ret}$. Considering a constant set point control during t , the supply/return temperatures at the loads/sources can be described by the curves in Fig. 2 (b): curve I represents $T_{i,t}^{sup}$, curve II represents $T_{j,t}^{sup}$, curve III represents $T_{j,t}^{ret}$ and curve IV represents $T_{i,t}^{ret}$. Thus, dividing steady state period t into n dynamic periods represented by the set $[\tau, \dots, \tau + \varphi \dots \tau + (n-1)]$, it is possible to describe the temperature in each dynamic period φ by (36), while ensuring the synchronism between t and $\tau + \varphi$ by (37).

$$\begin{aligned} T_{j,\tau+\varphi} &\triangleq f_\varphi(T_{j,t_0}, T_{i,t_0+1}) \\ &= (a_1^{-1} a_2)^{\varphi+1} T_{j,t_0} \\ &+ \sum_{k=0}^{\varphi} (a_1^{-1} a_2)^k \cdot a_1^{-1} a_3 \cdot T_{i,t_0+1} \\ &+ \sum_{k=0}^{\varphi} (a_1^{-1} a_2)^k \cdot a_1^{-1} a_4, \quad \forall \varphi \in [0, n-1] \end{aligned} \quad (36)$$

$$T_{j,t_0+1} = T_{j,\tau+(n-1)} \quad (37)$$

Equations (36)-(37) formulate the linearized heat dynamic model for supply pipe $i-j$. For return pipe, replace i, j in (36)-(37) by j, i respectively.

B. MEMG Planning Model with Multi-timescale Heat Dynamic Constraints

As shown in Fig.2 (b), curves II and IV vary with time τ , while curve I and III vary with time t . Thus, the heat exchange equations should be revised as:

$$H_{j,t}^d = c_w \cdot \bar{m}_{ij,t} \cdot \left(\frac{\sum_{\varphi=0}^{n-1} T_{j,\tau+\varphi}^{sup}}{n} - T_{j,t}^{ret} \right) \quad (38)$$

$$H_{i,t}^g = c_w \cdot \bar{m}_{ij,t} \cdot \left(T_{i,t}^{sup} - \frac{\sum_{\varphi=0}^{n-1} T_{i,\tau+\varphi}^{ret}}{n} \right) \quad (39)$$

where (38) describes the energy balance at load side, which indicates that the area of ABCD in Fig. 2 (a) equals to the area of EFGH in Fig. 2 (b) multiplied by $c_w \cdot \bar{m}_{ij,t}$. Equation (39) describes the energy balance at source side, which means that the area of JKLM multiplied by $c_w \cdot \bar{m}_{ij,t}$ equals to the output of the heat source in hour t .

Constraints are revised for the temperature difference at the loads and the heat exchangers at the sources, as follows:

$$T_{i,t+1}^{sup} - \frac{\sum_{\varphi=0}^{n-1} f_\varphi(T_{i,t}^{ret}, T_{j,t+1}^{ret})}{n} \leq N_S \cdot \Delta T^{ex,max} \quad (40)$$

$$\frac{\sum_{\varphi=0}^{n-1} f_\varphi(T_{j,t}^{sup}, T_{i,t+1}^{sup})}{n} - T_{j,t+1}^{ret} \leq \Delta T^{ex,max} \quad (41)$$

Furthermore, the supply temperatures at heat sources and return temperatures at loads in the two consecutive hours t are also constrained based on (36) and (37) as:

$$T_{j,t+1}^{sup} = (a_1^{-1} a_2)^n \cdot T_{j,t}^{sup} + \sum_{k=0}^{n-1} (a_1^{-1} a_2)^k \cdot a_1^{-1} a_3 \cdot T_{i,t+1}^{sup} \quad (42)$$

$$+ \sum_{k=0}^{n-1} (a_1^{-1} a_2)^k \cdot a_1^{-1} a_4 = f_{n-1}(T_{j,t}^{sup}, T_{i,t+1}^{sup})$$

$$\begin{aligned} T_{i,t+1}^{ret} &= (a_1^{-1} a_2)^n \cdot T_{i,t}^{ret} + \sum_{k=0}^{n-1} (a_1^{-1} a_2)^k \cdot a_1^{-1} a_3 \cdot T_{j,t+1}^{ret} \\ &+ \sum_{k=0}^{n-1} (a_1^{-1} a_2)^k \cdot a_1^{-1} a_4 = f_{n-1}(T_{i,t}^{ret}, T_{j,t+1}^{ret}) \end{aligned} \quad (43)$$

Besides, it is also important to guarantee, for every single intra-hour period, that the supply temperature is always higher than the return temperature, both at the source (44) and loads (45).

$$T_{a,t+1} \leq f_\varphi(T_{i,t}^{ret}, T_{j,t+1}^{ret}) \leq T_{i,t+1}^{sup} \leq 100 \quad \forall \varphi \in [0, n-2] \quad (44)$$

$$T_{a,t+1} \leq T_{j,t+1}^{ret} \leq f_\varphi(T_{j,t}^{sup}, T_{i,t+1}^{sup}) \leq 100 \quad \forall \varphi \in [0, n-2] \quad (45)$$

Finally, the planning model for MEMG with heat dynamic constraints can be expressed as:

$$\begin{aligned} \min \quad & \sum_t^{8760} (CE_t \cdot P_t^b + CG_t \cdot \eta_H^{-1} \cdot H_t^g) \\ & + (CI + OM) \cdot S^H \cdot \alpha_{H2E}^{-1} \end{aligned}$$

s.t.

Electricity : (2) – (6), (8) – (15)

Heat steady state : (18), (19), (26) – (28)

Heat dynamics : (38) – (45)

Coupling constraints : (29), (30)

In summary, in this new version of the MEMG planning model, the heat exchange equations, and the temperature difference equations for loads and sources are improved by integrating intra-hour temperature constraints as additional constraints of the hourly temperature. The model is finally presented as a MISOCP problem, which can be solved by a commercial solver.

C. Defining the Critical Scenario based on Fixed-MFR

As mentioned above, the fixed-MFR method is used to linearize the thermal model of the heating system. In the literature, e.g. [28], this linearization is done using an arbitrary feasible MFR. In this paper, we fix the mass flow rates at their minimum values with the objective of representing a scenario where the heat dynamic process is at its lowest rates,

while heat exchangers are working at their boundary condition with full capacity, forcing the largest temperature difference between supply and return pipes. Thus, when fixing the MFR, it is required to consider:

- The minimum allowable MFR of heat exchangers. According to (20) and (21) it is given by:

$$m_{ij,t}^{ex,min} = \frac{H_{j,t}^d}{c_w \cdot \Delta T^{ex,max}} \quad (46)$$

where $m_{ij,t}^{ex,min}$ denotes the minimum allowable MFR of heat exchangers connecting to pipe $i - j$ at time period t .

- The minimum allowable MFR of pipes. According to (46), when the heat loads are small, MFRs are also small. In this scenario, even supplied with the highest temperatures, and due to the heat loss, $T_{j,t}^{sup}$ may still be too low to satisfy the load demand. So, there should also exist a minimum allowable MFRs for pipes to limit the heat loss, given by:

$$m_{ij,t}^{p,min} = \frac{\lambda_{ij} \cdot L_{ij}}{c_w \cdot \ln(T_i^{sup,min} - T_{a,t}) / \ln(T_j^{sup,min} - T_{a,t})} \quad (47)$$

Equation (47) is obtained from the heat loss function of the steady state model, which describes the relation between $T_{i,t}^{sup}$, $T_{j,t}^{sup}$ and $\bar{m}_{ij,t}$. By fixing the value of $T_{i,t}^{sup}$, $T_{j,t}^{sup}$ is monotonically increasing with $\bar{m}_{ij,t}$, as shown in Fig. 3. So for heat pipe $i - j$, with minimum allowable MFR and lowest value of $T_{i,t}^{sup} = T_i^{sup,min}$, there should be $T_{j,t}^{sup} \geq T_j^{sup,min}$ in order to supply the load. For heat exchangers working at the largest temperature difference, $T_{j,t}^{sup}$ can be approximated by $T_j^{sup,min} = T_{a,t} + \Delta T^{ex,max}$.

Therefore, the MFR at time period t is given by:

$$\bar{m}_{ij,t} = \max(m_{ij,t}^{ex,min}, m_{ij,t}^{p,min}) \quad (48)$$

Fig. 3. Minimum mass flow rates

D. Final Remarks

It is important to stress that detailed aspects of computational fluid dynamics, including modeling of internal pipe structures (such as elbows), are out of the scope of this work. Instead, the methodology presented in this section adopts a quasi-dynamic representation of the heat networks, which uses the steady-state model of the hydraulic process and a

dynamic model of the thermal process. This representation allows to impose important dynamic security conditions into the steady state model of the MEMG design without losing the convexity and the linear proprieties of the optimization model. Moreover, it enhances the representation of microgrid heat network in the economic design, by imposing intra-hour constraints, without additional decision variables. Thus, from a computational perspective, this model does not compromise the scalability of MEMG design in comparison to the existing methods from the literature.

IV. CASE STUDY

A. Parameters Setup

Fig. 4 present a MEMG, adapted from [12], with 5 nodes and 4 lines in the electricity network (denoted as n_e), and 4 nodes and 3 lines in the heat network (denoted as n_h). The main electricity and heat networks parameters were taken from [12]: for example, the electricity conductors resistance r_{ij} and reactance x_{ij} were considered to be 64 and 1.4 p.u./m, respectively. The parameters of the heat pipes were considered to be the following: diameter $D = 0.286m$; $\lambda = 0.2W/mK$; and $R = 5mK/W$.

Regarding the temperature parameters of the model, the lower bound for the supply temperatures of the source and the load were assumed to be $T_i^{sup,min} = 60^\circ C$ and $T_j^{sup,min} = 40^\circ C$, respectively. The ambient temperature assumption was $T_{a,t} = 20^\circ C$. The intra-hour time step is $\Delta\tau = 15$ minutes.

The electricity and heat loads of each building in the microgrid were obtained from an open repository from the US Department of Energy [40]. The full building load dataset for the region of San Francisco in the Typical Meteorological Year 3 (TMY3) can be found in [41]. Each building name in Fig. 4 corresponds to a building label in the dataset. As done in other microgrid planning applications with the similar datasets [12], [14], 24 typical design days (1 weekday + 1 weekend profile per month) were constructed by averaging the hourly data.

A gas-fired CHP unit is connected to node 3, which consumes natural gas to generate heat and electricity. The objective is sizing the CHP unit as well as heat exchangers considering investment and operational costs as described in (1). The prices of electricity (peak and off-peak time-of-use rates) as well as the natural gas are given in Table I. To better illustrate the method proposed, it was assumed that the microgrid is not allowed to sell electricity back to the main grid. This assumption avoids an oversizing of the CHP due to electricity sells and allow the results to focus on the specific contributions of this paper around the heat model. The CHP unit investment cost is \$1150/kW, the O&M costs are \$200/kW per year and the lifetime is 15 years. The annualized capital investment (ACI) is calculated as follows:

$$\begin{aligned} ACI &= \frac{Capital \ Investment \times Discount \ Rate}{1 - (1 + Discount \ Rate)^{-n}} \\ &= \frac{1150 \times 0.03}{1 - (1 + 0.03)^{-15}} = \$96.3316/kW \end{aligned} \quad (49)$$

Fig. 4. MEMG case study.

TABLE I
PRICES OF ELECTRICITY AND NATURAL GAS

Time period	Off-peak	Peak
	1-8,18-24	9-17
Electricity price (\$)	0.207	0.226
Natural gas price (\$)	0.034	0.034

B. Investment Results

The optimization results are presented in Table II, where capital investment, O&M investment, and operational costs are given. The table shows two comparative cases: Case I considers only the steady-state heat model and represents an existing methodology for MEMG design; Case II considering the dynamic heat constraints in the MEMG planning and represents the methodology presented in this paper. The source capacity, the number of equipment required and the maximum temperature difference at source ($\Delta T^{so,max}$) are given in Table III.

The results show that the CHP capacity needed in case I is 2.94 MWh, while in case II is 3.78 MWh. This means that when considering dynamic constraints in the heat flow, the required CHP capacity increases significantly. This leads to a more expensive solution of capital investment, O&M cost, and natural gas consumption. However, the amount of energy purchased from the external grid decreases, since more electricity can be generated by the CHP. Finally, the maximum temperature difference at the source is 69.89°C for case I and 80°C for case II. As $\Delta T^{ex,max}$ is set as 20°C , the number of heat exchangers does not change with the cases. For both cases, 8 heat exchangers are required, including 4 for each load and 4 at the heat source.

C. Impact of Heat Dynamics on MEMG Planning

To analyze the effect of heat dynamics on the MEMG planning solution, a comparison between case I and II are shown using a typical weekday profile. Fig.5 (a) and (b) show the supply and return temperatures of the two cases while Fig.5 (c) and (d) present the electricity generated and purchased from the grid in the two cases. It can be seen that in case II, the system works at lower temperatures compared to case I. However, the average temperature difference of case II is larger than case I. The accumulated heat source generation during the day is 21523kWh for case I, and 34895kWh for case II. So, by integrating the dynamic constraints, more heat generation is required by the CHP, which also reduces the need for electricity purchases. To further address how heat dynamics

Fig. 5. Temperature comparison between case I and II.

impact the generation, we substitute the steady state results (case I) into the dynamic model (case II). Fig.6 (a) presents the load profile of the school at node 2, while Fig.6 (b) shows the original steady state temperatures $T_{3,t}^{sup}$, $T_{2,t}^{sup}$, $T_{2,t}^{ret}$ and $T_{3,t}^{ret}$, which represent the supply temperature at source, supply temperature at load, return temperature at load, and return temperature at source, respectively. By substituting these, e.g. $T_{3,t}^{sup}$ and $T_{2,t}^{ret}$, into the dynamic model we obtain the revised temperatures as $T_{2,t}^{sup,rv}$ and $T_{3,t}^{ret,rv}$.

As shown in the Fig.6, when there is an increase of the heat load at $t = 7h$ (a), the $T_{2,t}^{sup}$ and $T_{3,t}^{ret}$ also rise significantly (b). However, due to the heat dynamic constraints, both $T_{2,t}^{sup,rv}$ and $T_{3,t}^{ret,rv}$ fail to increase as much as the other temperatures. Therefore, with current $T_{3,t}^{sup}$ and $T_{2,t}^{ret}$, the load demand cannot be satisfied. The solution is either to increase $T_{3,t}^{sup}$ or decrease $T_{2,t}^{ret}$, which in any case would require a larger capacity of the heat source.

Thus, this comparative analysis between current MEMG design methodologies (Case I) and the method proposed (Case II) shows that, after a sudden increase in heat load, the solution given by the first approach is unable to satisfy the load at the intra-hour time scale. This reinforces the relevance of the methodology proposed and the need to guarantee a security conditions of the heat flow dynamics in the design phase.

TABLE II
COMPARISON OF INVESTMENT BETWEEN STEADY-STATE AND DYNAMIC MODELS

	Cost	Case I: Steady-state Heat Model	Case II: Dynamic Heat Model
Fixed Cost (\$/year)	Capital Investment	171278	219657
	O&M Cost	350618	451061
Variable Cost(\$/year)	Cost of Electricity Bought from External Grid	3898029	3365720
	Cost of Fuels by CHP	387919	677319

TABLE III
PARAMETERS OF EQUIPMENT

Equipment	Capacity(kWh)/Number/Temperature($^{\circ}C$)	
	Case I: Steady-state Model	Case II: Dynamic Model
CHP unit	2.94×10^3	3.78×10^3
Heat exchangers	8	8
$\Delta T^{so,max}$	69.89	80

Fig. 6. Case I response to a sudden increase of the heat load.

D. Critical Mass Flow Rate Conditions

In section III, the method proposed to fix MFR is used to design the critical scenario for planning the MEMG. Considering that MFRs have a direct impact on source capacity and temperatures, multiple optimizations were run for different MFRs to illustrate the robustness of the criteria presented in (46)-(48). Table IV shows the results of this analysis, where m_0 are the MFRs given by (48), they are increased proportionally. S^H is the heat source capacity, and the $\Delta T^{so,max}$ is the maximum temperature difference between supply and return pipes at the heat source. It can be seen that as MFRs rise, $\Delta T^{so,max}$ decreases in both cases. Accordingly, the number of heat exchangers decreases from 8 to 6 in case I, and from 8 to 7 in case II. Moreover, the S^H varies differently in case I and II. It is dependent on the MFRs and the temperature difference at the heat source, instead of $\Delta T^{so,max}$. Therefore, the designed MFRs in Section III produce the largest heat source capacity, number of heat exchangers, and temperature difference. It should also be noted that the reliability of the hydraulic system is guaranteed, since mechanical structures tend to be more stable under lower MFRs.

Table IV also shows that the investment, fuel and electricity cost reflect to this capacity variation. Especially, with more fuels consumed by the CHP unit, less electricity is required from the external grid.

V. CONCLUSION

This paper presents a MEMG optimal planning method, considering the intra-hour dynamics of the heating system as constraints of the energy dispatch and consequently of the capacity sizing of the CHP units. To obtain the heat dynamic constraints, this paper further studies and reveals the relation among intra-hour temperatures and inter-hour temperatures. Since this relation in its original form does not allow to maintain the convexity of the optimization model, we adopted a quasi-dynamic representation of the heat networks, which uses the steady-state modeling of the hydraulic process and dynamic modeling of the thermal process. Therefore, a fixed-MFR method is used to simplify it as a mixed-integer linear programming problem. Considering that MFRs have a direct impact on the planning results, a critical scenario is defined to set the MFRs as the minimum allowable value and guarantee the security of the design.

In the case study, tests are carried out on a 5-node microgrid with the proposed model, and compared with a traditional approach considering only steady-state models. The results show that when heat flow dynamics is neglected in the MEMG optimal design phase, the size of the coupling units (such as CHP) tend to be underestimated affecting the overall economics of the microgrid investments. In contrast, when the security conditions of the heat dynamics are added to the model, the assets are sized in a more adequate and realistic manner. In the case presented, this also impacted the economics of the electricity dispatch and the MEMG benefited from the increased CHP capacity, since less electricity is needed from the external grid. Thus, the results prove the relevance of the proposed model and its ability to better capture intra-hour operational conditions in the planning phase while providing a more realistic estimation of the microgrid design costs.

Finally, repeated tests on several levels of MFRs show that MFRs have a significant influence on the CHP sizing capacity. Meanwhile, it also verifies that the designed critical scenario possesses the largest heat source capacity, number of heat exchangers, and temperature difference between supply and return pipes. Future studies should explore alternative ways of designing these critical scenarios within the set of feasible MFRs solutions that satisfy the hydraulic constraints.

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TABLE IV
PLANNING RESULTS USING MULTIPLE MFRS

	MFRs	Overall investment (\$)	S^H (kWh)	Cost of fuels (\$)	Cost of electricity (\$)	$\Delta T^{so,max}$ ($^{\circ}C$)	Heat exchangers	
							load	source
Case I	m_0	4.8078×10^6	2.9407×10^3	3.8792×10^5	3.8980×10^6	69.8883	4	4
	$2.0m_0$	4.8008×10^6	2.9499×10^3	3.9728×10^5	3.8803×10^6	46.9921	4	3
	$3.0m_0$	4.7977×10^6	2.9531×10^3	4.0104×10^5	3.8732×10^6	34.8354	4	2
	$4.0m_0$	4.7962×10^6	2.9547×10^3	4.0308×10^5	3.8693×10^6	27.5890	4	2
	$5.0m_0$	4.7952×10^6	2.9557×10^3	4.0436×10^5	3.8669×10^6	22.8146	4	2
Case II	m_0	4.7138×10^6	3.7831×10^3	6.7732×10^5	3.3657×10^6	80	4	4
	$2.0m_0$	4.6354×10^6	3.7563×10^3	7.8675×10^5	3.1827×10^6	79.0133	4	4
	$3.0m_0$	4.6046×10^6	3.6490×10^3	8.0019×10^5	3.1574×10^6	79.0929	4	4
	$4.0m_0$	4.5825×10^6	3.5258×10^3	7.9100×10^5	3.1662×10^6	78.9221	4	4
	$5.0m_0$	4.5653×10^6	3.4242×10^3	7.8723×10^5	3.1711×10^6	78.6771	4	3

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