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Optimal Control of Offshore Indoor Climate

Zhenyu Yang and Andrea Valente

Abstract— **An optimal indoor climate control is very critical to manned offshore platforms in terms of onboard staffs' comfort, safety and the production efficiency as well. Due to the harsh weather condition and severe spatial limitations, offshore indoor climate control is much more challenging than any on-ground situations. This paper presents a preliminary investigation of optimal offshore indoor climate control for an oil-and-gas offshore platform by employing some advanced control technique - Model Predictive Control (MPC). The Single-Zone concept is adopted for modeling the offshore indoor climate, based on the thermal energy and mass balances. A constraint MPC solution is developed, and compared with a PID solution, which is derived from the existing control system. The simulation results show huge potential to significantly improve the performance of the existing system using optimal control techniques.**

I. INTRODUCTION

A recent study shows that the energy consumption of buildings accounts up to 40% of energy used worldwide, and buildings lead to 21% of all greenhouse gas emissions [10]. In recent decades, there has been extensive studies on the development of efficient buildings [4], [7], mathematical modeling of indoor climate [1], [3], [5], and optimal control of buildings' ventilation/HVAC systems [8], [12], [13], [14] etc. The literature survey reveals that almost all existing studies on energy-efficient buildings focus on the on-ground situations. Little work has been done for offshore platforms [2]. Even offshore HVAC standards are recent, like the ISO 15138 (2000) and Danish offshore HVAC technical standard [6], launched in the last decade. Most offshore HVAC systems designs have heavily relied on trial and error methods [2]. Due to the fact that the offshore buildings need to withstand harsh sea weather, severe spatial limitations and different ventilation principles, the development of an offshore indoor climate/HVAC system is far more challenging and complicated than typical on-ground situations. Nevertheless, the efficiency and reliability of offshore HVAC systems is critical to the onboard staffs' working/living conditions, platform's production activities, staff and facilities safety.

This paper is based on a project cooperated with a Danish company to study the feasibility of improving the HVAC system for a manned offshore platform as shown in Fig.1. In recent years the existing onboard HVAC system showed some problems in maintaining the indoor air pressure for building areas. Thereby the objective of the joint project is to investigate some improvements of this HVAC system performance using some advanced control strategies, along with some necessary hardware updates. This paper reports our investigation of this problem by using constraint Model Predictive Control (MPC) strategy.

Fig. 1. The Considered platform locates in the North Sea

Some preliminary investigation discovered two main problems of the current HVAC system [9]: (i) The pressurization fans are not powerful enough to handle harsh weather conditions; (ii) The deployed PID controller for the HVAC system has limited capability to handle rapidly changing and harsh operating conditions. Thereby, besides recommending bigger and more powerful pressurization fans, the advanced control strategy named constraint MPC is also investigated in our project.

The motivations for considering the MPC technique for the concerned system are:

- The indoor thermal dynamic often has slow time responses due to the relatively large time constants, while the harsh offshore weather conditions could generate rapidly changing disturbance to the considered offshore HVAC system. Thereby, some combination of feedback and feedforward/prediction control mechanism is required;
- The MPC is a kind of adaptive and on-line optimal solution, it can be more robust and flexible than typical PID controllers with respect to hostile operating conditions;
- The MPC solution can explicitly take care of constraints on system inputs, outputs and states in a systematical way. this is very important for offshore systems, from a safety perspective; and
- The MPC technique originally rooted and grew up in industrial process applications. Furthermore, some existing investigations [12], [13] for on-ground indoor climate control have indicated that there is a huge

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potential for using the MPC technique for efficient indoor climate control. It would be quite interesting to check how to extend these results/methods to handle offshore indoor climate situations.

After a mathematical model for a typical offshore indoor climate and its HVAC system is derived using the Single-Zone concept [1], [13], a constraint MPC controller is developed and afterwards compared with a PID solution derived from the existing HVAC control system. Our simulation results clearly show that the MPC performance is much better than the PID's, in terms of faster time responses, less overshoots and more robustness to disturbance.

The rest of the paper consists of a brief introduction to the offshore HVAC system and its operating principles in Section II. Section III proposes a mathematical model for a typical offshore indoor thermal dynamics based on Single-Zone concept. Section IV discusses the development of constraint MPC and shows some simulation results and comparison with a developed PID solution, and finally we conclude the paper in Section V.

II. OFFSHORE HVAC SYSTEM

Due to specific operating conditions and critical safety requirements, the configuration of an offshore HVAC system is much more complicated than that of any on-ground systems. Moreover, the offshore ventilation principle is also different from that of on-ground cases: the former follows the overpressure (pressurization) principle, while the later follows the low-pressure (natural ventilation) principle.

A. Offshore HVAC Configuration

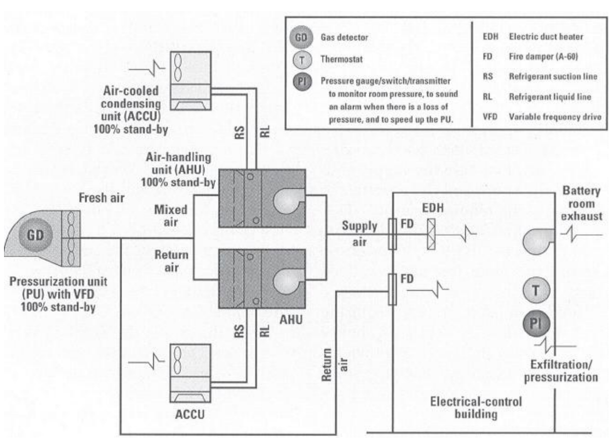

Fig. 2. Typical Offshore HVAC for Electrical-Control Building [2]

The buildings on the offshore platform are usually classified as living (accommodation) quarters, control rooms, workshops and storage buildings etc.. On the typical platform, the living quarters are located under the helideck and the corresponding HVAC system usually is located between the helideck and the living buildings in a skid-mounted way. As shown in Fig.2, a typical HVAC system consists of three general units arranged in some redundant way, i.e., the Pressurization Unit (PU), the Air-Cooled Condensing Unit (ACCU), and the Air-Handling Unit (AHU). The PU normally comprises of a set of variable-speed fans with their variable frequency drivers. The PU provides over-pressured fresh air to the AHU. A typical AHU diagram consists of a number of moisture separators and filters, fire dampers, shut-off dampers, heating/cooling coils, fans and non-return dampers. Each part has its clear standard specifications and functionalities, and more details can be found in [6], [11]. The ACCU provides cooling functionality to the air flowing through AHU. The fire damper is required at the inlet of each air intake of concerned quarter.

B. Offshore Ventilation Principle

On-ground ventilation systems usually follow the (inside) low-pressure principle so that either natural ventilation [1], [3], [5], or forced ventilation or both (often named hybrid ventilation [4], [7]) can be employed. Offshore ventilation follows the (inside) over-pressure principle. The main reason for this is safety, i.e. to prevent any potential flammable gas entering the buildings, which could cause fires or asphyxiation problems. Normally, the indoor pressure should be kept of 25-50 Pa over the outside ambient pressure when all operable exterior openings are closed and all mechanical exhaust systems operate normally [2], [11]. Both temperature and humidity inside the buildings should be maintained within an acceptable range even when the pressurization is lost due to sudden opening of exterior doors/windows [6].

C. Considered HVAC System

The considered buildings on the platform are classified into three parts: Accommodation Quarter (AQ), Control Room (CR) and Kitchen Quarter (KQ). The AQ consists of a number of rooms separated by doors, and each door has an air vent at the bottom to insure equal pressure in all AQ rooms. KQ consists of a number of partially-separated zones as shown in Fig.3. All zones have their own inlet air canals which supply air flows to the corresponding areas. Normally, one duty fan in the AHU is in operation to supply fresh air to all inlet canals. At the terminal of each inlet canal, there is an adjustable shutter to regulate the amount of fresh air into each room/zone.

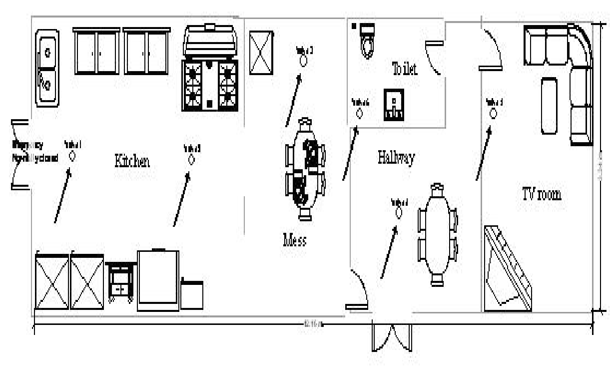

Fig. 3. Ventilation Inlet Allocations in Kitchen area [9]

TABLE I
SYSTEM PARAMETER & VARIABLES

Symbol	Interpretation (Unit)
A_{in}	inlet opening area (m^3)
A_{out}	outlet opening area (m^3)
C_{air}	heat capacity coeff. of air (J/KgK)
m_r	air mass in the room (Kg)
$\dot{m}_r$	indoor air mass changing rate (Kg/sec)
Q_{dist}	thermal disturbance (J/sec)
Q_{in}	heat trans. rate into room via inlet (J/sec)
T_r	indoor temperature (^oC)
T_{atm}	(outside) ambient temperature (^oC)
T_{in}	temperature inside the inlet canal (^oC)
V_{wind}	wind speed (m/sec)
V_{room}	room volume (m^3)
ρ_r	indoor air density (Kg/m^3)
ρ_{atm}	ambient air density (Kg/m^3)
ρ_{in}	inlet canal air density (Kg/m^3)
P_r	indoor air pressure (Pa)
P_{atm}	ambient air pressure (Pa)
P_{in}	inlet canal air pressure (Pa)
q_{in}	volumetric flow rate via inlet (m^3/sec)
q_{out}	volumetric flow rate via outlet (m^3/sec)

III. MATHEMATICAL MODELING

The dynamical model of the HVAC system is considered. Given that the current project stage mainly focuses on conceptual development, and we will consider potential industrialization at later stages, we introduce the following simplifying assumptions: (1) A single room/zone equipped with an adjustable inlet air-valve and an adjustable outlet air-valve is assumed, i.e. the AHU and the ventilation terminals are simplified to be virtual air-valves; (2) The modeling of indoor thermal dynamic follows the Single-Zone concept [1], [4], i.e. the indoor temperature/pressure is assumed to be uniformly distributed throughout the zone; (3) The pressure inside the inlet air-canal is assumed to be constant, which can be achieved by a proper pressurization control loop; (4) The ambient temperature is assumed to be below the required indoor temperature. This is often the case for the North Sea climate. The symbols used in the model and their meanings are listed in the Table.1.

A. Indoor Thermal Dynamic

The indoor thermal dynamic can be modeled according to thermal energy and air mass conservations. From the thermal energy perspective, there is

$$C_{air}m_r\dot{T}_r = \dot{Q}_{rad} + \dot{Q}_{in} + \dot{Q}_{out} + \dot{Q}_{dis} + \dot{Q}_{dist}, \quad (1)$$

where $\dot{Q}_{in}$ denotes the heat transfer rate into the room via inlet and it can be estimated by

$$\dot{Q}_{in} = C_{air}\dot{m}_{in}(T_{in} - T_r). \quad (2)$$

If the flow rate q_{in} can be estimated/measured, then $\dot{m}_{in}$ can be estimated by $\dot{m}_{in} = \rho_{in}q_{in}$. Similarly, $\dot{Q}_{out}$ represents the heat transfer rate through the outlet and it can be modeled as

$$\dot{Q}_{out} = C_{air}\rho_r q_{out}(T_{atm} - T_r), \quad (3)$$

where the indoor air density can be estimated by $\rho_r = \frac{m_r}{V_{room}}$.

$\dot{Q}_{rad}$ in eq.(1) is the thermal energy contributed from the radiator(s) and electric duct heater(s) and it can be modeled as

$$\dot{Q}_{rad} = h_{rad}A_{rad}(T_{rad} - T_r), \quad (4)$$

where T_{rad} is the radiator surface temperature and it is assumed constant in the following, A_{rad} is the radiator surface area, and h_{rad} is the average convection coeff. for the entire surface. We refer to [14], [13] for more detail modeling of a heating systems.

The thermal dissipation through the buildings is denoted as $\dot{Q}_{dis}$ in eq.(1) and it is estimated by

$$\dot{Q}_{dis} = K_{wall}A_{wall}\frac{1}{L_{wall}}(T_{atm} - T_r), \quad (5)$$

where K_{wall} is the average conductivity coeff. of the wall material, A_{wall} is the total building surface area where the thermal dissipation could happen and L_{wall} is the thickness of the surrounding wall. The thermal influences from the solar radiation, leakage through the buildings, radiations from staffs and any inside appliance (e.g., kitchen ovens, batteries and computers in control room etc) are simply modeled as one thermal disturbance term Q_{dist} in (1).

According to the mass balance principle [4], [7], there is

$$\dot{m}_r = \rho_{in}q_{in} - \rho_r q_{out}. \quad (6)$$

It is obvious that (2), (3) and (6) depend on the flow rate q_{in} and q_{out}, which can be obtained by modeling the inlet and outlet valves, respectively.

B. Inlet and outlet Valve Models

Both the inlet and outlet valve performances follow the natural ventilation principle under the assumption that the inlet canal pressure is maintained by the pressurization control loop.

Following flow dynamic theory, the air flow through the inlet valve can be estimated through

$$q_{in} = c_{in}A_{in}\sqrt{2(\frac{P_{in}}{\rho_{in}} - \frac{P_r}{\rho_r})}. \quad (7)$$

Within the vicinity of the valve, the approximation $\rho_{in} = \rho_r$ can be reasonable, thereby (7) can be simplified as

$$q_{in} = c_{in}A_{in}\sqrt{\frac{2(P_{in} - P_r)}{\rho_{in}}}. \quad (8)$$

The only controllable part in (8) is the opening area A_{in}, which has an operable range $0 \leq A_{in} \leq A_{in}^{max}$.

The outlet has a similar format as (8), taking the potential wind into consideration:

$$q_{out} = c_{out}A_{out}\sqrt{\frac{2\Delta P_{out}}{\rho_r}}, \quad (9)$$

where the differential pressure over the outlet valve consists of two factors: (i) the differential pressure induced by possible outside wind/gust [7], and (ii) the differential pressure due to thermal buoyancy [4]. According to [12], [13], ΔP_{out} can be estimated by

$$\Delta P_{out} = \begin{array}{l} \frac{1}{2}C_w\rho_{atm}V_{wind}^2 + P_r - P_{atm} \\ +\rho_r g(H_{npl} - H_{out})\frac{T_r - T_{atm}}{T_r}, \end{array} \quad (10)$$

where C_w is a specific pressure coefficient related to wind direction [13], H_{out} is the equivalent height of the outlet center to the reference pressure level and H_{npl} is the natural pressure level [3], [5].

C. Indoor Pressure Model

The ideal gas equation is use to predict the indoor pressure based on the mass of indoor air and indoor temperature, i.e.,

$$P_r = R_a \frac{m_r}{V_r} T_r, \quad (11)$$

where R_a is the ideal gas coefficient.

D. Entire System Model

The entire system diagram is illustrated in Fig.4. The controlled pressurization block is also illustrated in dash-line at the top. At this stage in the project, the only controllable input is A_{in}, all other external inputs are regarded as external uncontrollable inputs. Nevertheless, this model can be naturally extended to have more controllable inputs. The modeling of pressurization unit can be extended from the modeling of the fan units in our previous work [13]. The outputs of the system are the indoor temperature T_r and pressure P_r.

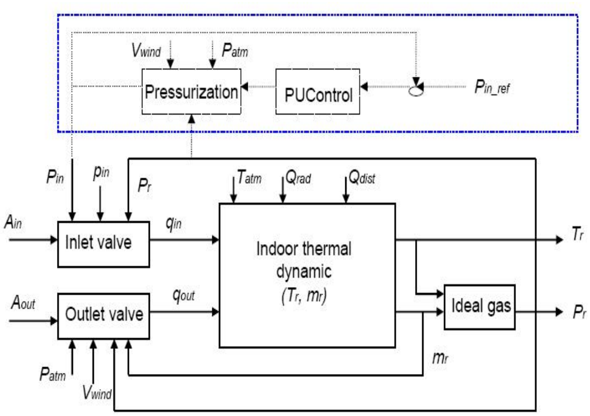

Fig. 4. Diagram of a Single-Zone Model of Offshore Indoor Climate

E. System Parameters and Steady-state Performance

The system parameters can be estimated either through designed experiments [3], [13], or calculated through relevant theory [2], [5], [14]. Due to some practical and safety limitations and the fact that the pressurization fan setup on the platform is due to be changed, most of system parameters we currently use are calculated from measured raw data. More testing data are expected in the future to validate the mathematical model and estimate system parameters.

According to the guidelines of the Danish Offshore HVAC Standards [6], the required indoor temperature should be between 18-22 degrees Celsius. The following simulation chooses the steady-state temperature as $T_r^{ss} = 22C^o$. The indoor pressure is expected to be 50 Pa above the steady-state outside (ambient) pressure, i.e., $P_r^{ss} = 50 + P_{atm}$. If $P_{atm} = 10^5$ Pa, then $P_r^{ss} = 1.0005 \times 10^5$ Pa. If the air temperature in the air inlet canal is assumed as $9C^o$, which is the average year-round outside temperature in Denmark, and the current pressurization unit normally provides 1.0075×10^5 Pa of pressure, then the air density inside the inlet canal ρ_{in} can be calculated, based on ideal gas assumption (11).

Regarding to the steady-state inlet/outlet valve positions, we suppose there is no leakage of air except through outlet valve. The offshore working regulation requires that the indoor air should be changed six times per hour in order to keep its quality [6]. This means the steady-state air inflow rates should have the capability: $q_{in}^{ss} = q_{out}^{ss} = 6V_{room} = 1680m^3/h$ under the condition that there are no disturbances. Once the steady-state values $q_{in}^{ss}, P_{in}^{ss}, P_r^{ss}$ and ρ_{in}^{ss} are known, the steady-state opening area of the inlet valve position can be calculated according to (8). For our consideration, it is $A_{in}^{ss} = 0.0114m^2$ in the following simulation. The steady-state position of the outlet valve can be calculated according to (9) under assumption that $V_{wind}^{ss}, \rho_{atm}^{ss}, P_{atm}^{ss}, T_{atm}^{ss}$ are known/measured.

The simulation of the obtained nonlinear system is illustrated in Fig.5. The initial value A_{in} of the inlet valve is smaller than the steady-state value, thereby the mass of air M_r and indoor pressure P_r decrease from the initial (steady-state) values, while the indoor temperature T_r increases over $22C^o$. When A_{in} is changed to be larger than A_{in}^{ss}, then M_r, P_r and T_r change to the opposite direction. Finally, the system reaches in the steady-state performance.

IV. DEVELOPMENT OF CONSTRAINT MPC

The state-space model-based MPC design technique is employed for the control development. Some tradeoff has been made with respect to the model precision, computation loads, complexity of optimization problems and practical implementation issues.

A. Linearized Model

A linearized model is obtained and used for MPC development at an equilibrium point with $T_r^{ss} = 22^oC$, $M_r^{ss} = 165.3kg$, $T_{amp} = 9^oC$, $P_{in} = 1.0075 \times 10^5 Pa$ and $A_{in}^{ss} = 0.0114m^2$. There are two reasons for linearizing the model: (i) The linear model will be used to construct a Kalman Filter (KF) to estimate the system state M_r based on measured pressure P_r and T_r, even though an Extended Kalman Filter (EKF) could be used directly based on the original nonlinear system. Normally EKF requires much more computational resource due to the online computation of Jacobian matrices; (ii) A linear prediction model would make it much easier and require less computational load to solve a constraint optimization problem.

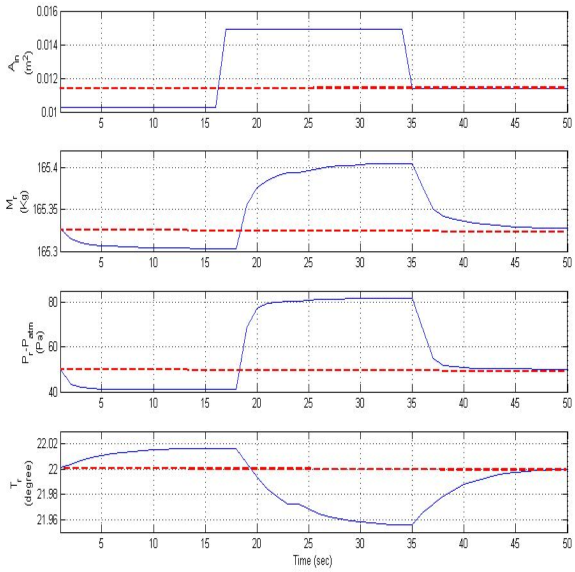

Fig. 5. Nonlinear System Response (Dash-lines: steady-state performance)

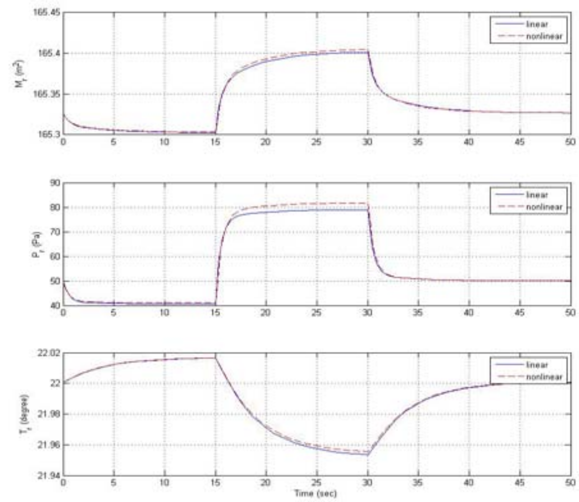

Fig. 6. Performance Comparison of Nonlinear vs. Linearized Systems

The performance comparison between the linearized system and the original system is shown in 6. It is clear that the linearized system exhibits almost same behavior as the nonlinear system, except that the linear model leads to a bit more overpressure when the inlet valve has a larger open area than the steady-state position as shown during the $15-30sec$ period.

B. System Constraints

The control signal A_{in} is bounded by its minimal and maximal values, i.e.,

$$0 \leq A_{in}(k) \leq 0.014 \ (m^2). \tag{12}$$

The inlet shutter is supposed to be driven by a DC-motor, thereby the changing rate, denoted as ΔA_{in}, of the inlet valve's open area is limited by the dynamics of the DC-motor and the corresponding mechanical system. For our case, there is

$$-0.003 \leq \Delta A_{in}(k) \leq 0.003 \ (m^2/sec). \tag{13}$$

There is no specific constraint regarding to system output P_r, which is used for feedback purpose. The indoor temperature T_r is not used by the control at this stage.

C. Constraint MPC

The developed controller consists of a KF for state estimation and a constraint MPC solution based on estimated/measured states. The sampling frequency is selected as 100 Hz (w.r.t. the room pressure dynamic is much fast than its thermal dynamic). The prediction horizon and control horizon are determined through several try-and-error tunings. In order to evaluate the developed advanced control strategy, a PI controller is also developed based on the existing control

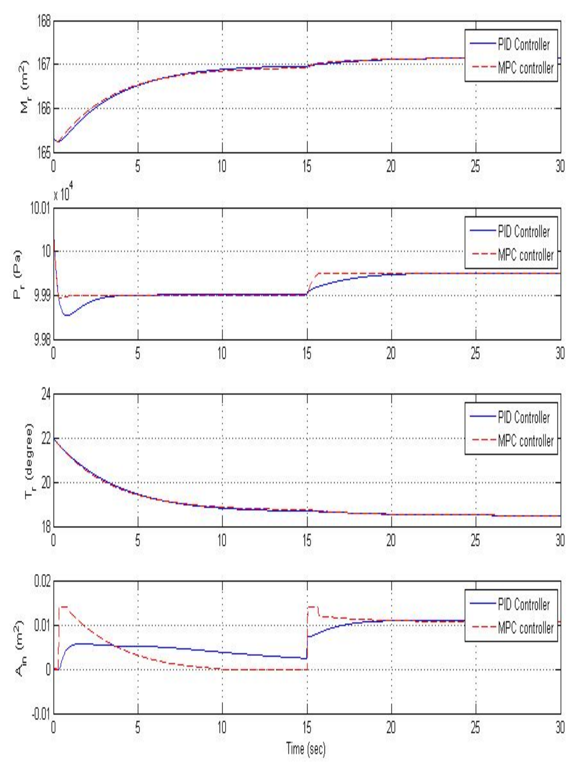

Fig. 7. Performance Comparison of MPC vs. PID Controllers)

system. The system performances controlled by MPC and PI-control are compared. As shown in Fig.7, it is obvious that the MPC controlled system has much faster response and less overshoot w.r.t. the change of set-points of the indoor pressure P_r.

The robustness of the controlled systems is also investigated. As shown in Fig.8, the operating condition is slightly different from the steady-state situation. Furthermore, some random feature is added into the ambient temperature. The PI controller let the indoor pressure drift away from the expected level - more than 50 Pa over the ambient pressure during 12-15 sec. In contrast, the MPC controlled system succeeded in keeping the required pressure during this period. Finally the MPC leads the controlled system with fast response and without aggressive behaviors.

V. DISCUSSION AND CONCLUSIONS

The offshore indoor climate control is investigated by using advanced optimal control techniques. After a simple mathematical model is derived based on the single zone concept, a constraint MPC controller is developed to manipulate the inlet opening area so as to control the indoor pressure. Compared with the system controlled by a PI controller which is developed based on the existing control system, the system controlled by MPC shows clearly fast and moderate responses, as well as better robustness to disturbances. The proposed methods can be easily extended to handle multiple zones frame as well.

The current investigation only considers the pressure control by manipulating the inlet valve (corresponding to the shutter opening area in the real system). The control of the indoor air quality is not explicitly considered yet, it could be considered by introducing the control mechanism of the outlet valve, then the considered system becomes a MIMO case. Furthermore, temperature control is not explicitly considered either; it could be considered by coordinating the heating (e.g. radiators) control system as well. If we also consider the pressurization control, the control of offshore indoor climate leads to a very sophisticated and challenging optimization problem. How to formulate and solve this generalized problem is part of our future work. Due to safety issues and the fact that current pressurization unit is going to be updated, the developed control system has not been implemented and tested. However, this conceptual development and the simulation investigations show clear potential to improve the offshore indoor climate control by using some advanced control techniques.

VI. ACKNOWLEDGMENTS

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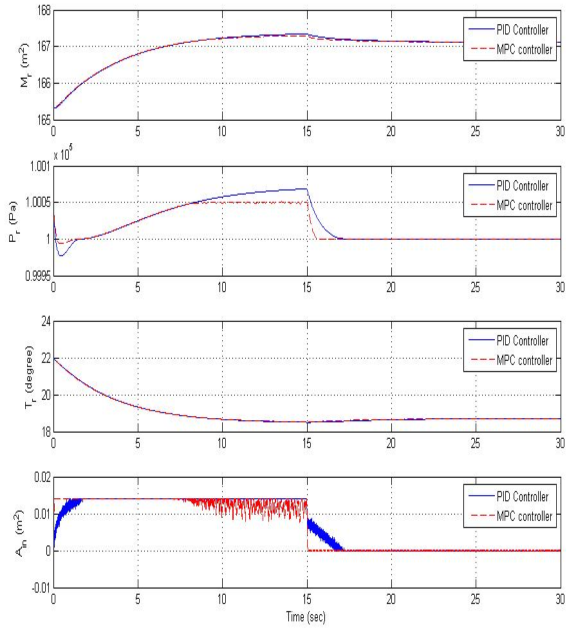

Fig. 8. Performance Comparison of MPC vs. PID under Disturbance

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