Numerical Modeling of Non-stationary Processes in Cryogenic Mechanical Thermocompressor

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***Abstract*—Mechanical thermocompressor is suited for gas compression with constant volume by means of heat addition. Its classic modelling theories usually do not take into account non-isothermal conditions of its chambers. This article deals with thermocompressor modelling with non-isothermal hot and cold chambers. The results of the modelling show significant loss (16.45%) of compressor delivery due to non-isothermal conditions of its chambers. Maximal mass mean temperature deviations for cold and hot chambers are equal to 14.98% and -8.5% correspondingly.**

***Index Terms*—Thermocompressor, isochoric process, regenerator**

I. Introduction

Mechanical thermocompressor is a device suited for converting of heat energy directly into potential energy of pressure. Application of such device requires presence of the heat source which will create a oscillations of pressure. Efficiency of thermocompressor depends on relation of temperatures of higher and lower heat sources. Almost all previous studies describe thermocompressors which operate with heat source with temperature higher than temperature of environment.

First concept of thermal compression was suggested by V. Bush [1]. Previous researches study kinematics of thermocompressor or analyse its efficiency [2]-[4]. Detailed analysis of stationary processes in thermocompressor with non-ideal component characteristics was made in [5].

Recent researches were focused on investigation of thermocompressors with angular gap regenerators [6] and operation of thermocompressor together with pulse tube [7].

Developed thermocompressor is planned to be used to fasten a process of re-gasification in cryogenic fueling tank. Cryogenic fueling tank is a device which can both store and gasify cryogenic fluids and is developed by Samara National Research University [8].

Thermocompressor can be used in any case where there is a free source of heat and/or cold (combined case is most beneficial because it has a high relation of temperatures. As it was shown in [9] it is possible to achieve a pressure ratio equal to a 75% from relation of temperatures.

II. Description of Thermocompressor and Mathematical Model

This paper presents mathematical model and results of the calculation of the working processes in thermocompressor considering both non-ideal characteristics of the thermocompressor components and working fluid and non-stationarity of the processes itself.

Physically mechanical thermocompressor is a cylindrical tube with displacer and regenerator which could be combined in one unit (Fig. 1).

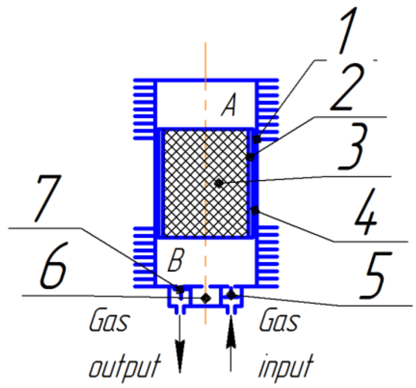

Figure 1. Principal scheme of the mechanical thermocompressor: 1 – thermocompressor shell; 2 – displacer-regenerator shell; 3 – regenerator; 4 – gap; 5 – input valve; 6 – rod; 7 – output valve; A – hot chamber; B – cold chamber.

Technically thermocompressor consists of such elements as: piston with regenerator inside; piston rod; heat exchangers to compensate non-perfection of regenerator; input and output valves, cold and hot chambers.

Working cycle of thermocompressor starts from high dead centre when both valves are closed and volume of the cold chamber is maximal (point 1). Piston starts to move downside pushing working fluid through regenerator to a hot chamber. Due to Amontons's Law of Pressure-Temperature, pressure is increasing because mean temperature of the working fluid inside the thermocompressor is increasing. When the pressure is high enough, output valve opens and working fluid with given pressure starts to be charged to a customer. When piston reaches low dead centre, output valve closes (point 3) and working fluid stops to be charged to a customer.

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Piston starts to move upside pushing working fluid back to a cold chamber. Pressure is decreasing due to decrease of the mean temperature inside the thermocompressor. When pressure drops low enough, input valve opens, charging new portion of working fluid inside the thermocompressor. Working cycle of thermocompressor is presented on Fig. 2.

Thus, for one working cycle thermocompressor rise the pressure of certain amount of gas Δm, which enters the TC through the input valve and leaves the TC from the output valve during one cycle.

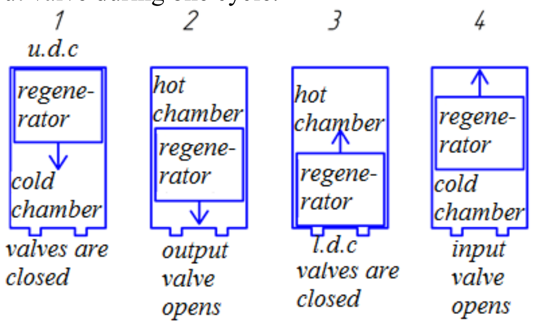

Figure 2. Working cycle of the thermocompressor.

There are three areas in thermocompressor which can be formally separated from each other: cold chamber. regenerator and hot chamber. They are connected by enthalpy flow. Input and output valves in given case are located in cold chamber so mass exchange with environment is executed through cold chamber.

Before non-stationary processes of the thermocompressor can be studied it need to be designed. Input data for thermocompressor designing is:

- Temperature of the high and low heat sources,
- Initial pressure,
- Pressure which opens output valve,
- Desired mass output,
- Drive frequency.

Algorithm for thermocompressor designing is presented, for example, in [10] and allow to calculate thermocompressor geometric characteristics, characteristics of the regenerator including its efficiency.

Mathematical model of the thermocompressor working processes consists of solution of three energy equation for every thermocompressor element together with consolidated condition equation for all elements.

Assumptions for calculation:

- Lack of pressure drops in valves,
- Displacer rod has a zero volume,
- Temperatures of regenerator walls in both chambers are constant,
- Working fluid is charged in cold chamber with temperature equal to a temperature of cold chamber wall,
- Pressure in both chambers as well as in regenerator is the same,
- Working fluid is ideal gas.

Chambers volumes changes were set as harmonics:

$$V_h(\varphi)=V_{hd}+0{,}5\cdot V_d\cdot\left(1-\cos(\varphi)\right);\quad V_c(\varphi)=V_{cd}+0{,}5\cdot V_d\cdot\left(1+\cos(\varphi)\right). \tag{1}$$

where V_h, V_c are volumes of the hot and cold chambers correspondingly; V_{hd}, V_{cd} are dead volumes of corresponding chamber; V_d is displaceable volume and φ – crank angle.

There are always three mutually depending variables. In case when valves are closed, they are T_h – temperature of the hot chamber; T_c – temperature of cold chamber and p – pressure in thermocompressor. In case when valves are open instead of pressure p such variable is mass inside thermocompressor m_{tc}.

Energy equation for a gas chambers:

$$dI+dQ=dU+dL \tag{2}$$

Enthalpy change is caused by mass exchange with regenerator and environment through valves.

$$dI_h=c_{ph}\cdot T_{rh}\cdot dm_{rh} \tag{3}$$

for a hot chamber and

$$dI_c=c_{pc}\cdot T_{rc}\cdot dm_{rc}+c_{pc}\cdot T_{wc}\cdot dm_{tc} \tag{4}$$

for cold chamber.

Here C_{ph} and C_{pc} – are isobaric heat capacities of corresponding chambers, T_{rh} and T_{rc} – temperature of the gas which ejected in chamber from regenerator (these values are substituted by chamber temperatures in cases when gas is ejected into regenerator); m_{rh} and m_{rc} – masses of gas which are ejected from/to regenerator.

Temperatures T_{rh} and T_{rc} depend on regenerator efficiency η_{reg}:

$$T_{rh}=T_h-\eta_{reg}\cdot\left(T_h-T_c\right);\quad T_{rc}=T_c+\eta_{reg}\cdot\left(T_h-T_c\right). \tag{5}$$

Heat is added or rejected from the chamber by means of heat exchange with walls of thermocompressor and heat conductivity losses through regenerator:

$$dQ_h=\alpha_h F_h\left(T_{wh}-T_h\right)d\tau-\lambda_{reg}F_{reg}\frac{(T_h-T_c)}{b_{reg}};\quad dQ_c=\alpha_c F_c\left(T_{wc}-T_c\right)d\tau+\lambda_{reg}F_{reg}\frac{(T_h-T_c)}{b_{reg}}, \tag{6}$$

where α_h, α_c – heat transfer coefficients from the walls of corresponding chamber to a working fluid in this chamber; F_h, F_c – contact surface area between corresponding chamber and working fluid; λ_{reg} – heat conductivity of regenerator material; F_{reg} – cross-section area of regenerator; b_{reg} – regenerator thickness.

The change of internal energy in hot chamber is equal to

$$dU_h=c_{vh}\cdot T_{rh}\cdot dm_{rh}+c_{vh}\cdot m_h\cdot dT_h \tag{7}$$

for hot chamber and

$$dU_c=c_{vc}\cdot T_{rc}\cdot dm_{rc}+c_{vc}\cdot T_{wc}\cdot dm_{tc}+c_{vc}\cdot m_c\cdot dT_c \tag{8}$$

for cold chamber.

Here C_{vh}, C_{vc} are isochoric heat capacities of corresponding chambers, m_h and m_c are working fluid masses in corresponding chambers.

Work against external forces were calculated by change of both pressure and volumes:

$$dL_h = pdV_h; \quad dL_c = pdV_c, \tag{9}$$

System is closed by condition equation for all thermocompressor elements:

$$p = \frac{m_{tc}R}{\frac{V_h}{T_h} + \frac{V_{reg}}{T_{reg}} + \frac{V_c}{T_c}} \tag{10}$$

III. Modelling Results

The results of the calculations are the dependencies of pressure and temperatures on chambers volume or crank angle.

Initial data for thermocompressor designing and further modelling is presented in Table I.

TABLE I. Initial Data for Calculation

Parameter	Value
Input pressure, P_{in}	101325
Pressure ratio, π_{tc}	1.5
Cold chamber wall temperature T_{wc}, K	100
Hot chamber wall temperature T_{wh}, K	300
Dead volume of chambers, m^3	$2{,}5\cdot10^{-4}$
Displaceable volume, m^3	$5\cdot10^{-3}$
Thermocompressor cylinder diameter, m	0.15
Regenerator volume, m^3	$3{,}55\cdot10^{-3}$
Frequency, Hz	1
Working fluid	Nitrogen

Distribution of the gas temperature in both chambers for isothermal and non-isothermal calculations is presented on Fig. 3 (all plots are presented for 2 working cycles).

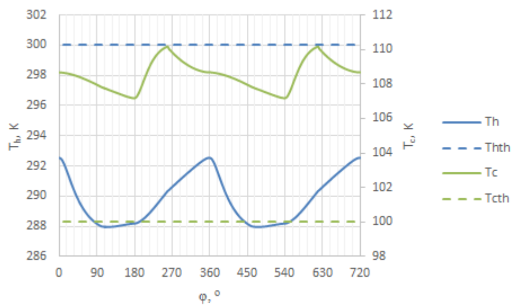

Figure 3. Distribution of the gas temperature in hot and cold chambers.

Maximal temperature deviation for hot chamber is -12.04 K and for cold chamber is 10.18 K or -4.01% and 10.18% correspondingly. Similar arithmetic mean values are -10.24 K and 8.52 K or -3.41% and 8.52% correspondingly. However, arithmetic mean values do not show appropriate picture because the mass of the gas in chambers is changed during the working cycle. So mass mean value of temperature is more appropriate for estimation of temperature deviation:

$$\delta T_{mhi} = \frac{T_{hi} - T_{hth}}{T_{hth}} \cdot \frac{m_{hi}}{m_{hmean}}; \quad \delta T_{mci} = \frac{T_{ci} - T_{cth}}{T_{cth}} \cdot \frac{m_{ci}}{m_{cmean}}. \tag{11}$$

Calculated maximal mass mean values increased and are equal to 14.98% and -8.5% for cold and hot chambers correspondingly.

p-V diagrams for hot and cold chamber are presented on Fig. 4 and Fig. 5.

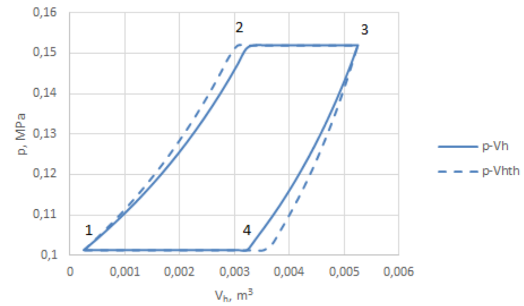

Figure 4. Axial section of thermocompressor with transverse fins

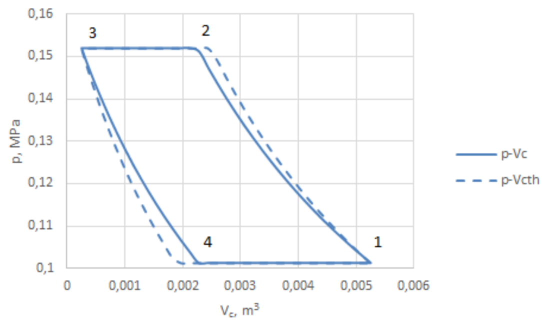

Figure 5. p-V diagram for cold chamber.

As it can be seen from the diagrams, working cycles of chambers are inversely directed to each other and the area of the working cycle in case of isothermal thermocompressor is larger than in case of non-isothermal one.

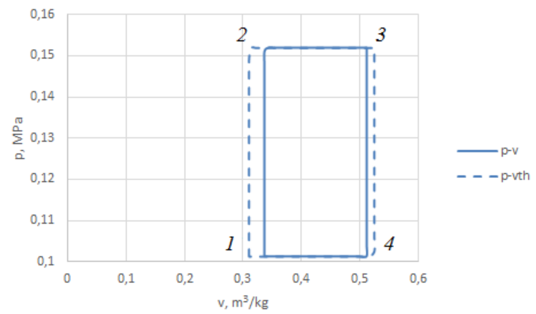

Figure 6. p-v diagram for thermocompressor.

For entire thermocompressor p-V diagram can be plotted for specific volume of charged mass only (total volume of thermocompressor does not change so its classic p-V diagram will be represented as single line). Such diagram is presented on Fig. 6. The larger area inside the cycle, the more compressor delivery will thermocompressor produce.

Pressure distribution for thermocompressor for both isothermal and non-isothermal calculations is presented on Fig. 7. It can be seen that isothermal plot is slightly sharper than non-isothermal one which results in higher width of horizontal areas of the plots. These areas correspond to a discharge and charge processes and the higher this width the more thermocompressor delivery.

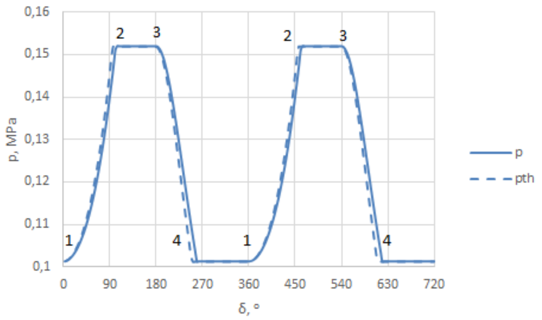

Figure 7. Pressure change in thermocompressor.

Thermocompressor delivery was calculated by expression:

$$Q_{tc} = \frac{\Delta m_{tci}}{\rho_{ci}} \cdot p_{in}\left(\pi_{tc} - 1\right) \tag{12}$$

Calculated values of thermocompressor delivery are equal to 125.09 J in case of isothermal chambers and 104.5 J in case of non-isothermal chambers. So, non-isothermal thermocompressor has a 16.45% less delivery and non-isothermal conditions of chambers must be taken into account during thermocompressor designing. This deviation will increase with increase of the thermocompressor frequency because time condition does not affect isothermal compressor but will affect non-isothermal thermocompressor by means of decrease of heat exchange time, for example.

Efficiency of thermocompressor can be estimated as compressor delivery relation to a energy spent on thermocompressor drive. This energy is spent on overcoming of friction losses in regenerator and thermocompressor rod:

$$\eta = \frac{Q_{tc} \cdot f}{N} \tag{13}$$

The value in contrast with thermodynamic COP can exceed 1 because in ideal thermocompressor there is no friction and required drive power is equal to 0. This coefficient can only be used for comparison of thermocompressors and is not appropriate for comparison with other types of compressors. For given cases the values of these coefficient are equal to 10.45 and 12.5 for non-isothermal and isothermal cases correspondingly.

From the point of view of cryogenic working fluid cold energy utilization it is possible to estimate an efficiency of its utilization. The algorithm of cold energy values calculation is presented in [11]. Compressor delivery is related to a cold energy value which is calculated by expression:

$$Q_{tc} = \int_{T_c}^{T_h} m_{tc} \cdot c_p \cdot dT \tag{14}$$

So final expression for cold energy utilization efficiency will be calculated as:

$$\eta_{ce} = \frac{Q_{tc}}{Q_{ce}} \tag{15}$$

This efficiencies values for isothermal and non-isothermal compressors are equal to 3.17% and 2.86% correspondingly. This means that thermocompressor is not the best way to utilize a working fluid cold energy and must be applied in special case where additional pressure is needed and for cold energy utilization it is better to use a organic Rankine cycle or other energy generation cycle.

IV. Conclusion

This article deals with working process modelling in mechanical thermocompressor with considering non-isothermal condition. Algorithm for non-stationary thermocompressor calculation is developed and allow to estimate the influence of non-isothermal condition on thermocompressor performance.

Calculation results show significant influence of non-isothermal condition which results in maximal mass mean temperature deviation equal to 14.98% and -8.5% for cold and hot chambers correspondingly. This deviation causes loss of compressor delivery equal to 16.45%. Thus, we can make a conclusion that non-isothermal conditions must be taken into account during thermocompressor modelling and designing.

Several factors affect the efficiency of the thermocompressor. These factors include:

- Regeneration efficiency;
- Geometric characteristics (heat exchange area, length of the cylinder which will affect the piston velocity and consequently gas flow velocity);
- Time factor which will influence by means of working cycle frequency;
- Dead volume;
- Application of heat transfer intensifications such as cylinder finning;
- Non-isothermal conditions of chambers walls.

Influence of these factors will be investigated in further researches.

Thermocompressor efficiency was estimated from the point of view of energy conversation and cryogenic fluid cold energy utilization. First method is appropriate for comparison of thermocompressors only. Efficiencies values are 12.5 and 10.45 for isothermal and non-isothermal cases. The second method allow to compare the efficiency of thermocompressor cold energy utilization with other plants which utilize a cold energy of cryogenic working fluid. The values of such efficiencies for thermocompressor are quite low are equal to 3.17%

and 2.86% for isothermal and non-isothermal compressors correspondingly. Such low values do not allow to use a thermocompressor as an effective mean of cold energy utilization.

The results of this work will be considered during designing of the thermocompressor which will operate together with cryogenic fueling tank [8] to fasten gasification of liquid in this tank.

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REFERENCES

[1] V. Bush, "Apparatus for compressing gases," US Patent 2,157,229, 1939.

[2] P. Arques, "Piston movement in thermocompressor," in *Proc. 1997 32nd Intersociety Energy Conversion Engineering Conference*, vol. 2, 1997, pp. 1003-1015.

[3] H. Karabulut, "Thermodynamic analysis of bush engine," *GU Journal of Science*, vol. 16, pp. 135-144, 2003.

[4] A. Kornhauser, "Analysis of an idealized Stirling thermocompressor," in *Proc. 31st Intersociety Energy Conversion Engineering Conference*, Part 1 (of 4); Washington, DC, USA, vol. 2, 1996, pp. 1331-1336.

[5] M. J. Edwards and R. B. Peterson, "Reciprocating thermocompressor performance with non-ideal component characteristics," *Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, vol. 221, no. 4, pp. 459-472, June 2007.

[6] W. Y. Lin, X. H. Wu, J. L. Yang, and L. W. Yang, "Experimental study and numerical analysis of thermocompressors with annular regenerators," *International Journal of Refrigeration*, vol. 36, no. 4, pp. 1376-1387, June 2013.

[7] W. Ji, X. Xue, J. Wang, Y. Zhou, L. Chen, and W. Zhu, "Coupling study of a novel thermocompressor driven pulse tube refrigerator," *App. Therm. Eng*., vol. 51, no. 1-2, pp. 630-634, 2013.

[8] E. V. Blagin, A. I. Dovgyallo, S. O. Nekrasova, D. V. Sarmin, and D. A. Uglanov, "Estimation of the energy efficiency of cryogenic filled tank use in different systems and devices," *App. Therm. Eng.*, vol.101, pp. 537-544, 2016.

[9] M. Edwards, "Design, modelling, and performance of a miniature reciprocating thermocompressor," Master's thesis, Mechanical Engineering, Oregon State University, 2005.

[10] E. V. Blagin, A. I. Dovgyallo, D. A. Uglanov, and S. S. Dostovalova, "Thermodynamic calculation of thermomechanical compressor," SSAU Reporter (in Russian)., vol. 41, 2013, pp. 25-30.

[11] E. V. Blagin, A. I. Dovgyallo, and D. A. Uglanov, "About LNG energy utilization efficiency estimation," *Proc. Eng.*, vol. 152, pp. 209-218, 2016.

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