

## Techno-economic Analysis of Ranque-Hilsch Vortex Tube Aided Hybrid Cooling and Drying System

Merve SENTURK ACAR\*, Halit ARAT\*\*and Oguz ARSLAN\*\*

\*Electricity and Energy Department, Tavşanlı Vocational School, Dumlupınar University, Kutahya, Turkey

\*\*Mechanical Engineering Department, Engineering Faculty, Dumlupınar University, Kutahya, Turkey

[merve.senturkacar@dpu.edu.tr](mailto:merve.senturkacar@dpu.edu.tr), [halit.arat@dpu.edu.tr](mailto:halit.arat@dpu.edu.tr), [oguz.arslan@dpu.edu.tr](mailto:oguz.arslan@dpu.edu.tr)

**ABSTRACT:** In this study, the Ranque-Hilsch vortex tube (RHVT) aided hybrid cooling - drying system was investigated by the energy analysis. In this aim, many systems were designed and evaluated with net present value (*NPV*) in the viewpoint of Life Cycle Cost (*LCC*). The RHVT experiments were performed for 9 different helical vortex generators, 3 different control valve angle ( $\alpha=30^{\circ}, 45^{\circ}, 60^{\circ}$ ), 3 different vortex tube body, 5 different opening position of the control valve and 5 different inlet flow pressure (201.325-601.325 kPa). The maximum energy efficiency of RHVT aided hybrid cooling - drying system was determined as 0.160 and 0.146 for the summer mode ( ambient temperature of 306.15 K) and winter mode ( ambient temperature of 289.15 K), respectively. *NPV* of this hybrid system was calculated as 816.08 €. The highest value of *NPV* was calculated as 35,784.39 € for the helical vortex generator of 0, control valve angle of  $30^{\circ}$ , and the 1<sup>st</sup> type of RHVT body.

**Keywords:** Economic analysis, Energy analysis, Hybrid of Cooling and drying, Vortex tube.

### 1. Introduction

As a result of the rapid increase of the population in our country and the world, the sustainability of the food supply and energy sources have gained importance. The drying was a regional method which was used for in long term storage of foods in our country. The cold storage of food was the other method which was used decades. The usage of these methods leads to higher energy consumption. The energy consumed in the food sector is used at the rate of 29 % of for the processes of heating, and at the rate of 16 % for the refrigeration and freezing process [1]. Therefore, the studies, aimed to improve the energy efficiency of the methods used in food preservation, have been important.

Ranque-Hilsch vortex tube (RHVT) consists of a principal tube, which a high-pressure gas stream enters tangentially, and splits into hot and cold temperature streams [2-3]. In literature, the hot and cold streams of RHVT were analyzed by energy and exergy analysis [4-5]. Sommers and Jacobi studied to improve the heat transfer on the air side of the cooling

evaporator used with vortex generator [6]. Aydın and Baki investigated the design parameters and performances of counterflow vortex tubes experimentally [7].

The studies in the literature are focused on the cold outlet flow of RHVT. In this literature, this cold stream was focused on the usage of refrigeration systems. There is no investigation about the hot stream of RHVT. Also, the studies in the literature about the cooling performance of the studies just argue whether the cold stream is available or not.

At this point of view, the design of refrigeration system is not available. In this study, the usage of both cold and hot streams was investigated in a hybrid cooling - drying system from the energy and economic point of view. R-143a was used as the refrigerant fluid in the vapor compression cooling system.

### 2. Design of Ranque-Hilsch Vortex Tube

The features of the helical generators and RHVT bodies were given in Table 1 and 2 which were used

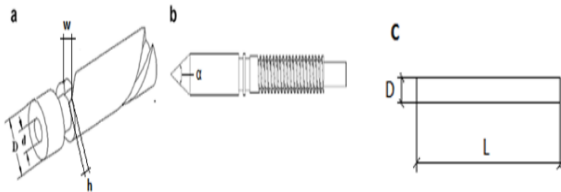
in the experiments. The helical vortex generator was shown in Fig 1. Three different control valve angle were used ( $\alpha=30^{\circ}, 45^{\circ}, 60^{\circ}$ ). The pressure of the inlet flow of RHVT was changed between 201.325 kPa and 601.325 kPa. The control valve opening position was changed (5 positions) which was adjusted the outlet flow rate of RHVT.

**Table 1.** Features of helical vortex generators.

Generator	h (mm)	w (mm)	d (m)	D (mm)
0	2.0	4.5	6.15	12
A	2.0	4.5	3.3	12
B	2.0	4.5	5.1	12
C	2.0	4.5	6.0	12
D	2.0	4.5	7.1	12
M	1.5	6.0	3.3	12
N	1.5	6.0	5.1	12
O	1.5	6.0	5.7	12
J	1.5	6.0	7.1	12

**Table 2.** Features of RHVT body.

RHVT Body No	L (mm)	D (mm)	L/D
1	480	12	40
2	350	12	29.17
3	210	12	17.5

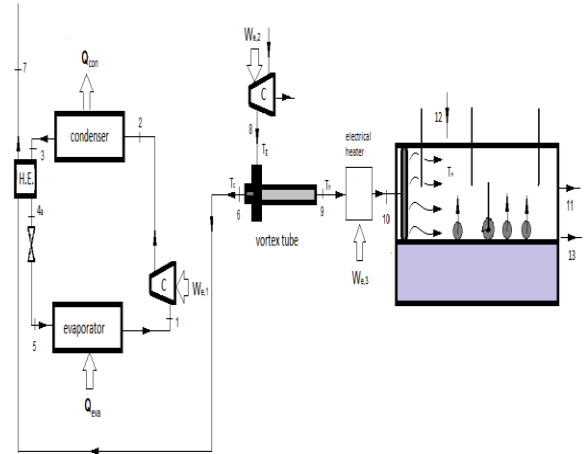


**Figure 1.** The helical generator (a), control valve (b) and RHVT body (c).

### 3. The RHVT Aided Hybrid Cooling and Drying System (RHVTHCD)

RHVTHCD flow diagram was shown in Fig 2. The cold stream of RHVT was integrated into the refrigeration cycle with the heat exchanger after the evaporator. The cold stream works as a cooling fluid. The compressed refrigerant enters to the condenser at point 2. The refrigerant enters to the heat exchanger at point 3 and the cold stream of RHVT takes the heat of refrigerant and leaves the heat exchanger at point 7. The refrigerant pressure is decreased to the inlet pressure of the compressor at the exit of the throttling valve (5). After the ambient air is compressed then inlets to RHVT (8). The hot exit of RHVT (9) was integrated into the drying system before the electrical

heater. The heated drying air (10) inlets to the dryer and takes the moisture of the fresh tomatoes and then leaves the dryer (11). The fresh tomatoes enter to the dryer point 12 and the dried tomatoes leave the dryer point 13.



**Figure 2.** The flow diagram of the RHVTHCD.

The cooling stage of the hybrid system was designed for different compressor discharge pressure and suitable evaporator temperatures of cooling tomatoes ( $T_1= 275.15\text{K}, 276.15\text{K}, 277.5 \text{K}$ ).

**Table 3.** The inlet and outlet air temperatures of the dryer and dried product temperature.

Design No	$T_{10}$ (K)	$T_{11}$ (K)	$T_{13}$ (K)
1	328.15	326.15	326.15
2	328.15	323.15	323.15
3	328.15	318.15	318.15
4	328.15	313.15	313.15
5	328.15	308.15	308.15
6	333.15	331.15	331.15
7	333.15	328.15	328.15
8	333.15	323.15	323.15
9	333.15	318.15	318.15
10	333.15	313.15	313.15
11	338.15	336.15	336.15
12	338.15	333.15	333.15
13	338.15	328.15	328.15
14	338.15	323.15	323.15
15	338.15	318.15	318.15
16	343.15	341.15	341.15
17	343.15	338.15	338.15
18	343.15	333.15	333.15
19	343.15	328.15	328.15
20	343.15	323.15	323.15

The thermodynamic properties of refrigerant were determined by REFPROP. The efficiency of the electrical heater was determined by the experiments which were made with resistance. The dryer inlet and outlet air temperatures and drying periods were identified with Page drying model for tomatoes. The inlet and outlet air temperatures of the dryer and dried product temperature ( $T_{13}$ ) were given Table 3. The fresh product inlet temperature ( $T_{12}$ ) was taken equal to the ambient temperature of 306.15 K for summer period (5 months) and 289.15 K for winter period (7 months).

#### 4. Energy Analysis

The following assumptions were handled in the energy analysis;

- 1) Potential and kinetic energy effects are negligible,
- 2) The compressor has an electrical ( $\eta_{el}$ ) and mechanical efficiencies ( $\eta_{mec}$ ) of 90%,
- 3) The compressor has an isentropic efficiency 70%,
- 4) The heat exchanger has an efficiency 70%,
- 5) The electrical heater has an electrical efficiency ( $\eta_{e,h}$ ) 27%,
- 6) The RHTV outlet stream flows pressure equals to environment pressure (101.325kPa),
- 7) The reference state is 101.325kPa and 293.15K.

Under these assumptions, the governing energy equations of the RHVTCT were obtained as following.

Evaporator;

$$\dot{Q}_{con} = \dot{m}_{sa} \cdot (h_1 - h_{4a}) \quad (1)$$

Condenser;

$$\dot{Q}_{eva} = \dot{m}_{R-143a} \cdot (h_3 - h_2) \quad (2)$$

Heat exchanger;

$$\dot{m}_6 \cdot (h_7 - h_6) = \dot{m}_{sa} \cdot (h_3 - h_{4a}) + \dot{Q}_{he} \quad (3)$$

Compressor 1 (vapor compression cooling system);

$$\dot{W}_1 = \dot{m}_{r-143a} \cdot (h_2 - h_1) \quad (4)$$

And electrical power of compressor 1;

$$\dot{W}_{e,1} = \frac{\dot{W}_1}{\eta_{mec} \cdot \eta_{el}} \quad (5)$$

Compressor 2 (RHVT);

$$\dot{W}_2 = R \cdot T \cdot \ln \frac{P_{1,a}}{P_8} \cdot \dot{m}_8 \quad (6)$$

And electrical power of compressor 2;

$$\dot{W}_{e,2} = \frac{\dot{W}_2}{\eta_{mec} \cdot \eta_{el}} \quad (7)$$

RHVT;

$$\dot{Q}_{RHVT} = \dot{m}_6 \cdot h_6 + \dot{m}_9 \cdot h_9 - \dot{m}_8 \cdot h_8 \quad (8)$$

Throttling valve;

$$\dot{m}_{R-143a} \cdot h_3 = \dot{m}_{R-143a} \cdot h_4 \quad (9)$$

Electrical heater;

$$\dot{W}_3 = \dot{m}_9 \cdot (h_{10} - h_9) \quad (10)$$

And electrical power of electrical heater

$$\dot{W}_{e,3} = \frac{\dot{W}_3}{\eta_{e,h}} \quad (11)$$

Dryer;

$$\dot{m}_{10} \cdot (h_{10} - h_{11}) = \dot{m}_{12} \cdot c_{p,13-12} \cdot (T_{13} - T_{12}) + (\dot{m}_{12} - \dot{m}_{13}) \cdot h_{ab,13} \quad (12)$$

Energy efficiency of RHVT aided hybrid cooling and drying system;

$$\eta = \frac{\dot{Q}_{eva} + \dot{m}_{10} \cdot (h_{10} - h_{11})}{\dot{W}_{e,1} + \dot{W}_{e,2} + \dot{W}_{e,3} + \dot{m}_8 \cdot h_{1,a} + \dot{Q}_{RHVT} + \dot{Q}_{he} + \dot{m}_{12} \cdot c_{p,12} \cdot T_{12}} \quad (13)$$

#### 5. Economic Analysis

The life cycle cost (LCC) of RHVTHCD occurs by the investment costs ( $C_{ic}$ ), salvage cost ( $C_{sc}$ ), operating costs ( $C_{oc}$ ), maintenance costs ( $C_{mc}$ ) and operating income ( $C_{oi}$ ).

$$LCC_{RHVTHCD} = C_{ic} + C_{sc} + C_{mc} + C_{oi} + C_{oc} \quad (14)$$

The salvage cost of the hybrid system was taken as 10% of the investment cost [8-9].

$$C_{sc} = C_{ic} \cdot 0.10 \quad (15)$$

The maintenance cost of the hybrid system was taken as 2% of the investment cost of the hybrid system [8-9].

$$C_{mc} = C_{ic} \cdot 0.02 \quad (16)$$

The operating income of the hybrid system includes cooling and drying earnings.

$$C_{oi} = (\dot{m}_{13,33} \cdot t_{33} + \dot{m}_{13,16} \cdot t_{16}) \cdot 15.66 + \left( \frac{((\dot{m}_{cp,33} \cdot t_{33}) + (\dot{m}_{cp,16} \cdot t_{16}))}{3600 \cdot 24} \right) \cdot 0.0047 \quad (17)$$

where  $\dot{m}_{13,33}$ ; mass of dried product summer period (kg/s),  $t_{33}$ ; summer period (s),  $\dot{m}_{13,16}$ ; mass of dried product winter period (kg/s),  $t_{16}$ ; winter period (s),  $\dot{m}_{cp,33}$ ; cooled product summer period (kg/s) and  $\dot{m}_{cp,16}$ ; cooled product winter period (kg/s). The unit price of dry product and cooled product is 15.66 € and 0.0047 €, respectively [10-13].

Operating costs of the system;

$$C_{OC} = C_e + C_{fp} \quad (18)$$

where  $C_e$ ; electrical costs of the hybrid system and  $C_{fp}$ ; fresh product costs which is dried in the drying system. Electrical costs;

$$C_e = ((\dot{W}_{e,1(33)} + \dot{W}_{e,2(33)} + \dot{W}_{e,3(33)}) \cdot t_{33}) + ((\dot{W}_{e,1(16)} + \dot{W}_{e,2(16)} + \dot{W}_{e,3(16)}) \cdot t_{16}) \cdot 0.107 \quad (19)$$

where 0.107 €/kW is the electrical energy[14].

The fresh product costs;

$$C_{fp} = \frac{((\dot{m}_{12,33} \cdot t_{33}) + (\dot{m}_{12,16} \cdot t_{16}))}{t_k} \cdot 0.6 \quad (20)$$

where  $\dot{m}_{12,33}$ ; mass of the fresh product summer period,  $\dot{m}_{12,16}$ ; the mass rate of the fresh product winter period and 0.6 €/kg is the cost of 1kg fresh product (tomatoes) [15].

The net cash flow;

$$C_T = (C_{fp} + C_{oi} + C_e + C_{mc}) \cdot (1 + i)^{t-1} \quad (21)$$

in this equation,  $i$ ; the interest rate and  $t$ ; the related year time of cash flow.

The Net Present Value (NPV) of hybrid system;

$$NPV = (C_{sc} + C_{ic}) + \sum_{t=0}^{ol} \frac{C_T}{(1+j)^t} \quad (22)$$

where  $ol$ ; operating life of the hybrid system,  $j$ ; the discount rate.

In this study, the operating life of the hybrid system has been added to calculations as 20 years. The discount and interest rates were taken as 9% and 7.25%, respectively [16-17].

## 6. Results and Discussion

Handling the operating parameters as  $T_{1,a}=306.15$  K,  $T_{10}=328.15$  K,  $P_2=1700$  kPa, RHVT generator type of 0,  $\alpha=30^\circ$ ,  $P_8=601.325$  kPa, 1<sup>st</sup> RHVT body, 3<sup>rd</sup> opening position, the change of energy efficiency ( $\eta$ ) with different dryer air inlet and outlet temperature differences ( $\Delta T_{10-11}$ ) and evaporator temperature ( $T_1$ ) was obtained as seen in Fig 3.

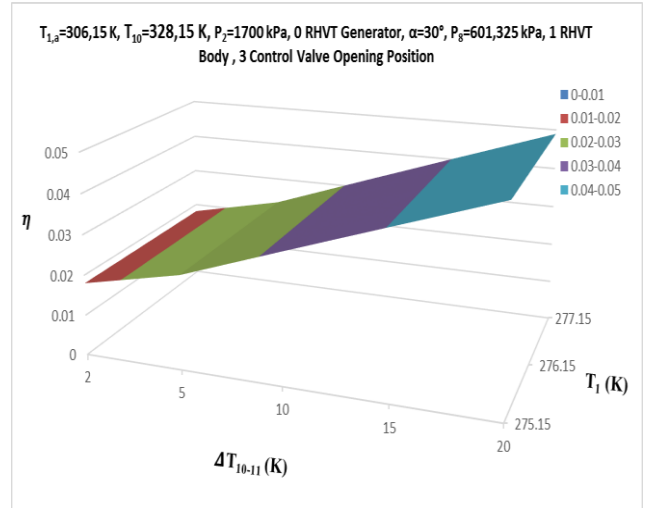


Figure 3. The variation of  $\eta$  versus  $\Delta T_{10-11}$  and  $T_1$ .

Fig 3 shows that the energy efficiency values of the hybrid system increase by the increase of the evaporator temperature and  $\Delta T_{10-11}$ . The energy efficiency of proposed hybrid system ranges between 0.0179 and 0.04897. Taking the evaporator temperature as  $T_1=277.15$  K, the change of energy efficiency ( $\eta$ ) with different dryer air inlet and outlet temperature differences ( $\Delta T_{10-11}$ ) and dryer air inlet temperature ( $T_{10}$ ) was obtained as seen in Fig 4.

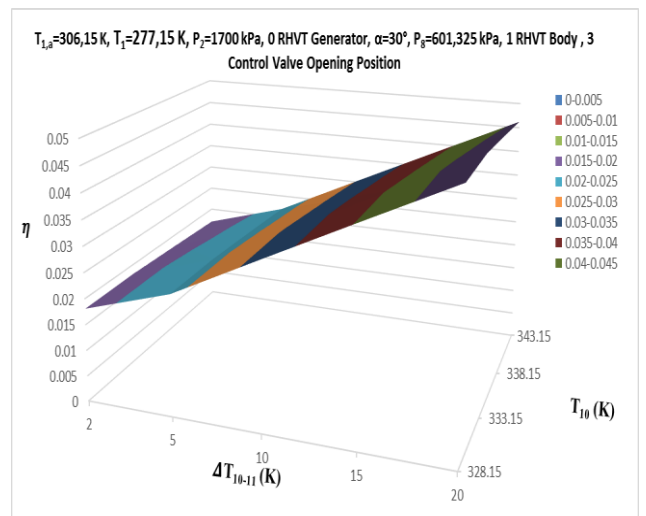
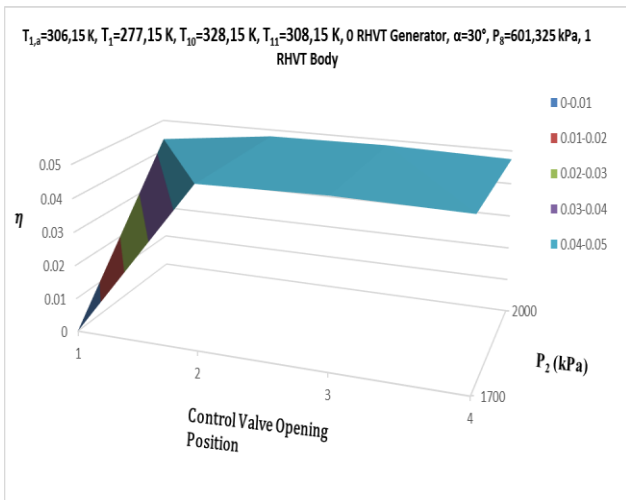


Figure 4. The variation of  $\eta$  versus  $\Delta T_{10-11}$  and  $T_{10}$ .

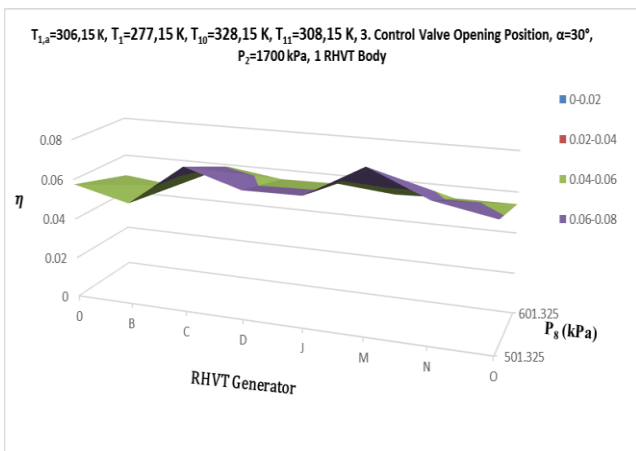
According to Fig 4, the energy efficiency of hybrid system increase with the increase of inlet air temperature of the dryer (between 328.15 and 333.15 K) and with the decrease of inlet air temperature of the dryer (between 333.15 and 343.15 K). At this stage, the energy efficiency of proposed hybrid system ranges between 0.01807 and 0.04897.

For  $T_{1,a}=306.15$  K,  $T_1=277.15$  K,  $T_{10}=328.15$  K,  $T_{11}=308,15$ K, RHVT generator type of 0,  $\alpha=30^\circ$ , 1<sup>st</sup> RHVT body, the change of energy efficiency according to the different positions of opening control valve and the different outlet pressures of compressor ( $P_2$ ) are given in Fig 5.



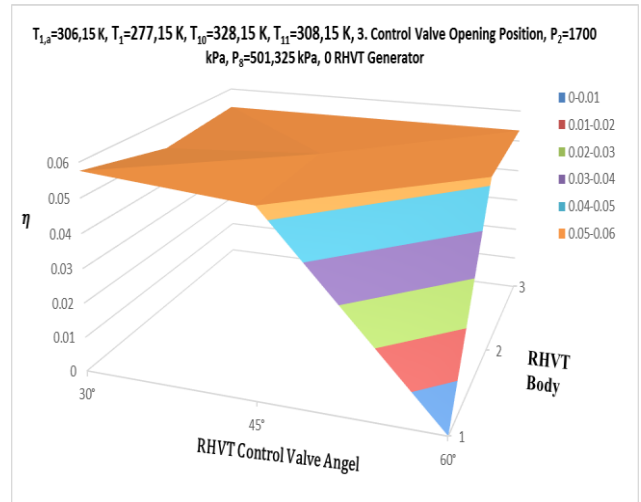
**Figure 5.** The variation of  $\eta$  versus  $P_2$  and control valve opening position.

According to Fig 5, the highest energy efficiency is recorded as 0.04897 for the hybrid system in the case of 1700 kPa outlet pressure of compressor and 3<sup>rd</sup> control valve opening position. The variation of  $\eta$  versus  $P_8$  and RHVT generators are given in Fig 6.



**Figure 6.** The variation of  $\eta$  versus  $P_8$  and RHVT generator.

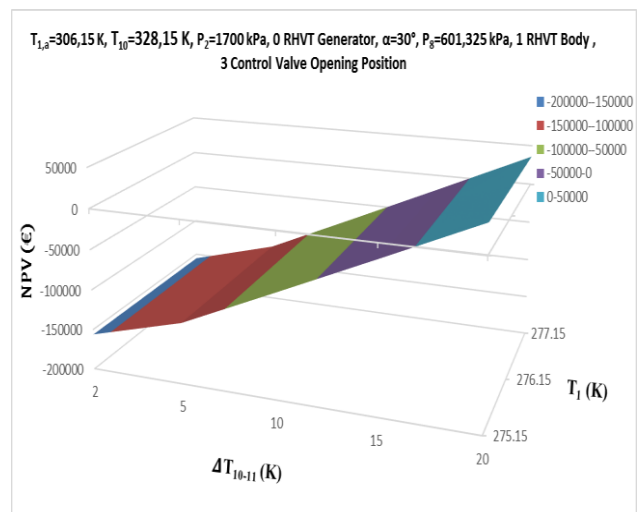
According to Fig 6, it is seen that the highest energy efficiency value is obtained at RHVT inlet stream flow pressure of 501.325 kPa and M type of RHVT generator. The energy efficiency of proposed hybrid system ranges between 0.0460 and 0.07973 for this case. The variation of  $\eta$  according to the control valve angel and body type of RHVT is given Fig 7.



**Figure 7.** The variation of  $\eta$  versus RHVT control valve angel and RHVT body.

As seen from Fig 7, the highest energy efficiency of the hybrid system was obtained as 0.0575 for control valve angle of  $30^\circ$  and 1<sup>st</sup> RHVT body.

Handling the operating parameters as  $T_{1,a}=306.15$  K,  $T_{10}=328.15$  K,  $P_2=1700$  kPa, RHVT generator type of 0,  $\alpha=30^\circ$ ,  $P_8=601.325$  kPa, 1<sup>st</sup> RHVT body, 3<sup>rd</sup> opening position, the variation of NPV versus  $\Delta T_{10-11}$  and  $T_1$  are given in Fig 8.



**Figure 8.** The variation of NPV versus  $\Delta T_{10-11}$  and  $T_1$ .

According to Fig 8, NPV increases with the increase of  $\Delta T_{10-11}$  and  $T_1$ . NPV of proposed hybrid system ranges between 156.250 € and 35,784.39 €. According to energy and economic analysis, the values belonging to the best design of the proposed system of temperatures, pressures, flow rate, energy value were demonstrated in Table 4.

**Table 4.** The characteristics values of the optimum system.

	$T$ (°C)	$P$ (kPa)	$\dot{m}$ (kg/s)	$E$ (kW)
1,a	306.15	101.325	0.056153	24.2857
1	277.15	700.51	0.00277	1.08275
2	321.43	1700	0.00277	1.16211
3	310.04	1700	0.00277	0.72080
4a	294.23	1700	0.00277	0.64605
5	277.15	700.51	0.00277	0.64605
6	282.85	101.325	0.00416	1.70242
7	301.883	101.325	0.00416	1.78212
8	306.15	601.325	0.05615	24.2250
9	333.85	101.325	0.05199	23.9362
10	328.15	101.325	0.05199	17.4347
11	308.15	101.325	0.05242	18.0292
12	306.15	101.325	0.00047	0.63968
13	308.15	101.325	3.3E-05	0.04514

## 7. Conclusion

In this study, the RHVT aided hybrid cooling - drying system was investigated from the viewpoint of energy and economic analysis. The optimum operating parameters of the RHVT aided hybrid cooling and drying system was defined as  $T_{1,a}$  306.15K, 0 RHVT body, 3<sup>rd</sup> control valve opening position, 30° control valve angel,  $P_8$  601.325 kPa,  $P_2$  1700kPa,  $T_{10}$  328.15 K and  $T_{11}$  308.15 K. The energy efficiency of the hybrid system was determined as 0.04897. For this case, NPV was calculated as 35,784.39 € which means this system is profitable for the investment taking the lifetime of the system into account.

## Acknowledgement

This study was supported by Scientific Research Projects Unit of Dumlupinar University (DPUBAP) with the project no of 2013/5.

## References

[1] M. Okos, N. Rao, S. Drecher, M. Rode, J. Kozak, "Energy usage in the food industry", American Council for an Energy-Efficient Economy, 1998.  
[2] G.J. Ranque, "Experiences sur la detente giratoire avec production simultanees 'un echappement d'air

chaud et d'air froid", Journal de Physique et Le Radium (in French) 7 (4), 1933, 112–114.

[3] M.H. Saidi, M.S. Valipour, "Experimental modeling of vortex tube refrigerator", Applied Thermal Engineering 23, 2003, p.1971–1980.

[4] Y. Xue, M. Arjomandi, R. Kelso, "Energy analysis within a vortex tube", Experimental Thermal and Fluid Science 52, 2014, p.139-145.

[5] V. Kırmacı, "Exergy analysis and performance of a counter flow Ranque–Hilsch vortex tube having various nozzle numbers at different inlet pressures of oxygen and air", International Journal of Refrigeration 32, 2009, p. 1626-1633.

[6] A.D. Sommers, A.M. Jacobi, "Air-side heat transfer enhancement of a refrigerator evaporator using vortex generation", International Journal of Refrigeration 28, 2005, p.1006–1017.

[7] O. Aydın, M. Baki, "An experimental study on the design parameters of a counterflow vortex tube", Energy 31, 2006, p.2763-2772.

[8] T. Güler, M. Yücedağ, "Feasibility report of atmosphere controlled cold storage facility", Direct Business Support Program, Fırat Development Agency (in Turkish), 2011.

[9] A. Yoğunlu, M. Yüzeroğlu, S. Hopoğlu, S. G. Gökçe, "The analysis and pre-feasibility of frozen fruit and vegetable sector", Fırat Development Agency (in Turkish), 2013.

[10] [http://isotsatise.com/kategori-6-KURU\\_KISLIKLAR.html](http://isotsatise.com/kategori-6-KURU_KISLIKLAR.html), The market price of dried tomatoes, [Last access: September 2015], (in Turkish).

[11] <http://www.organikim.com.tr/organik-domates-kurusu-200-gr>, The market price of dried tomatoes, [Last access: September 2015], (in Turkish).

[12] <http://tazekuru.com/urunler/64/kurutulmus-domates-100g-?gclid=CLiLs--chMgCFSLnwgod-rgHiA>, The market price of dried tomatoes [Last access: September 2015], (in Turkish).

[13] Anonym, (2015), Unit costs for cooling product, oral interviews with cold storage worker, Yalova, Turkey.

[14] EMRA, (2015), Energy Market Regulatory Authority board decision, Decision No: 5666, Official Gazette of Turkish Republic No. 29401, June 29, 2015 (in Turkish).

[15] <http://www.adana-bld.gov.tr/halfiyatlari.html>, The price of fresh tomatoes, [Last access: September 2015] (in Turkish).

[16] <http://www.tcmb.gov.tr/wps/wcm/connect/tcmb+tr/tcmb+tr/main+menu/duyurular/basin/2015/duy2015-13>, CBRT (Central Bank of Republic of Turkish), Interest rate, [Last access: September 2015].

[17] <http://www.tcmb.gov.tr/wps/wcm/connect/tcmb+tr/tcmb+tr/main+menu/para+politikasi/reeskont+ve+av>

[ans+faiz+oranlari](#), CBRT (Central Bank of Republic of Turkish), Discount rate, [Last access: September 2015]