Arab Gulf J. scient. Res. **2** (2), pp. 731-742 (1984)

Forced Convective Heat Transfer in the Condensation of Pure Vapour Jets on a Vertical Tube

O.A. Hamed and G.A. Sorial

Chemical Engineering Dept., Faculty of Engineering, University of Jordan, Amman, Jordan; and Faculty of Technology, University of Gezira, Wadi Medani, Sudan

ABSTRACT. Experimental investigations for the effect of jetwise flow of condensing steam on a single vertical tube have been carried out. The variables covered included orifice diameter, heat load and steam velocity. Particular emphasis has been laid on making direct comparison between normal flow of vapour and high jet vapour velocities.

It has been found that the steamside coefficient of heat transfer was increased by the jetwise flow to values of 2.0 to 9.55 times the value for stationary vapour. The corresponding vapour pressure loss to induce the jetwise flow was 4.7 and 50.0 cm Hg, respectively.

The high dependence of the heat transfer enhancement on the jet driving force is presented. The jetwise flow approach may find direct applications in industries where high pressure steam is available.

The mechanism of laminar film condensation on a vertical isothermal flat plate maintained at constant temperature below saturation temperature of the surrounding quiescent vapour was first studied by Nusselt in 1916 (Kern 1965). Since then, there has been further substantial analytical work directed towards evaluating the validity of Nusselt's results and increasing the extent of the problem domain.

In recent years, researchers have been developing conditions which will give a higher heat flux than that obtained by normal filmwise condensation. These techniques are:

1. Developing dropwise condensation by the addition of promotors to the vapour stream or coating them on the cooling surface. Dropwise condensation has not been utilized industrially because this form of condensation is unstable (Le Fevre and Rose 1969).

2. Rotational, vibrational and electrostatic forces were used to decrease the condensate film thickness but they have the disadvantage of requiring complicated equipment (Williams *et al.* 1968).

3. Modification of shape of cooling surface by use of roughened surfaces and finned tubes (Thomas and Hayes 1970).

4. Changes in process variables have also been shown to influence the heat transfer coefficient and the total flux, *e.g.* the degree of superheat, pressure of operation and vapour velocity (Silver and Simpson 1961, O'Bara *et al.* 1967, Furman and Hampson 1959, Rohsenow and Chio 1961, Leonard and Joseph 1967, Hamed 1972, Simpson 1969, Brochman 1965, Osment *et al.* 1965).

It has been demonstrated that the mass velocity of the condensing vapours improved the intensity of heat transfer. However, most investigators were faced by non-uniform vapour velocity distribution throughout the condensing surface. High vapour velocities were at the inlets and decelerating on flowing along the cooling surface due to the phase change. Some attempts (Osment *et al.* 1965, Simpson 1969, Hamed 1972) have been carried out to offset this problem.

Osment *et al.* (1965) investigated the effect of installing baffles and honeycomb grids in the steam inlet branch of the condenser. However, they found that such an arrangement gave improved steam distribution over the tube bundles but caused greater differences between the overall heat transfer coefficient at the top and bottom row tubes, and the heat transfer coefficients for the whole bundle were practically unaffected.

Simpson (1969) reported a new type of condenser which was divided into four parts with a central major access lane for the steam; part of the vapour branches to the upper two quadrants and part to the lower two quadrants. This type of condenser would enable, to some extent, a better distribution of the steam flow across the bundles of cooling tubes, but the arrangement of these access lanes is still based on the experience of the condenser designer and there is no accurate analysis of this arrangement.

In recent years, the use of the gas and vapour jet impingement as a high performance technique for heating or cooling a surface has been attempted. Several investigators have reported very high convective heat transfer rates for air cooling multi-jet systems; these include Garden and Cobonpue (1962); Huang (1963); Chance (1974); and Hollworth and Berry (1978).

Hamed (1972) investigated the effect of steam impingement on a single horizontal tube. The condensing vapour was admitted through a perforated tube. The vapour disintegrates and flows in a jet-wise mode, impinging on the condensing surface. The high velocity vapour jets sweep the condensate layer as well as any non-condensing gas film off the condensing surface. This resulted in an improvement in the heat transfer coefficients.

Hamed's investigation initiated the present work. The objective of this paper is to present the results of an investigation into the effect of the impinging jets on vertically falling films and to examine the parameters which control the magnitude of the jet velocities.

Experimental

A flow diagram of the experimental set-up is shown in Fig. 1. Its main elements are: a mechanical separator for removal of entrained moisture, a super-heater to provide a slight degree of super-heat, a measuring apparatus of the type developed by Furman and Hampson (1959) for determining the noncondensables in steam, the test condenser, a secondary condenser for condensing the uncondensed steam from the test section, and measuring vessels for collecting the condensate from test and secondary condensers.

A schematic diagram of the experimental condenser is shown in Fig. 2. For jet-wise mode, the steam was introduced into the condenser shell by means of a perforated tube of 1.27 cm inside diameter and extending the full length of the condenser. 170 orifices of the same diameter were drilled along one side of the tube facing the condensing tube, equidistant and spaced from each other 6.85 mm. Five different perforated tubes were used having orifices with diameter 0.7, 0.8, 0.9, 1.04 and 1.55 mm. The vapour feed pipe was spaced 7.0 mm from the condensing surface. It was blocked at one end and silver soldered to the top and bottom

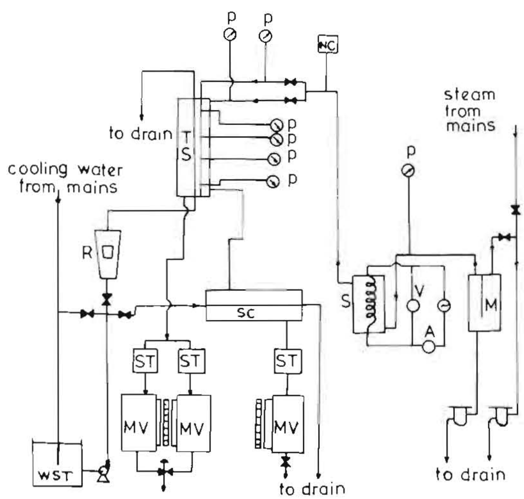

Fig. 1. Experimental setup. TS: testsection; SC: secondary condenser; WST: cooling water storage tank; R: rotameter; ST: steam trap; MV: measuring vessel; M: mechanical Seperater; V: voltmeter; A: ammeter; S: superheater; NC: non-condensable measuring apparatus.

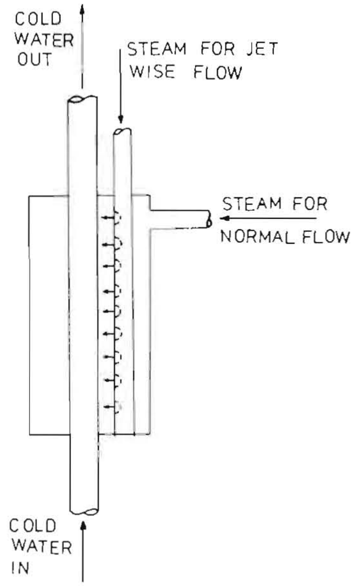

Fig. 2. A schematic diagram of the experimental condenser.

chambers of the test section. For normal flow, the steam was admitted to the condenser shell through a 2.54 cm tube.

The copper cooling tube was having an inside diameter of 1.27 cm and thickness of 1.59 mm. Three thermocouples were placed along different locations on the outside surface of the cooling tube and were brazed around the circumference of the cooling tube. The average heat transfer coefficient was thus determined from the three circumferential thermocouples. This was also checked by using the following indirect measurements and calculations. The water side coefficient h_w was estimated from Sieder and Tate equation (Kern 1965):

$$Nu = 0.027\,(Re)^{0.8}(Pr)^{0.4}\left[\frac{\mu}{\mu_w}\right]^{0.14} \tag{1}$$

The steam-film heat transfer coefficient, h_f, was then estimated from

$$\frac{1}{h_f} = \frac{1}{U_o} - \frac{d_o}{h_w d_i} - \frac{x_w d_o}{k_w d_m} \tag{2}$$

Overall heat transfer coefficients were obtainable directly from the experimental observations. The heat transfer, as obtained from the water side measurements, was checked by a balance with the steam side from all tests. It was found that agreements within 1-2 percent was usual, the difference never exceeded 7 percent. All the calculations were based on the heat transferred to the cooling water since this was simple to determine.

Steam jet velocities, $\bar{V}_{sj}$, through the manifold tube were estimated from Bernoulli's equation for a compressible fluid (Coulson and Richardson 1977):

$$\bar{V}_{sj} = C_D \sqrt{2 \left[\int_{P2}^{P1} \frac{dP}{\varrho} \right]} \qquad (3)$$

The coefficient of discharge, C_D, was taken to be equal to 0.62 (Zijnen 1951).

Results and Discussion

The experimental work carried out comprised of three seperate sets of heat transfer measurements.

Set 1

In this set, direct comparison were made between the effect of normal and jet-wise flow of steam on the intensity of heat transfer. For both modes, the cooling conditions were kept constnat, *i.e.* an inlet cooling water temperature of 295 ± 2 K and water velocities ranged from 3.3 to 1.1 m sec^{-1}. The vapour pressure

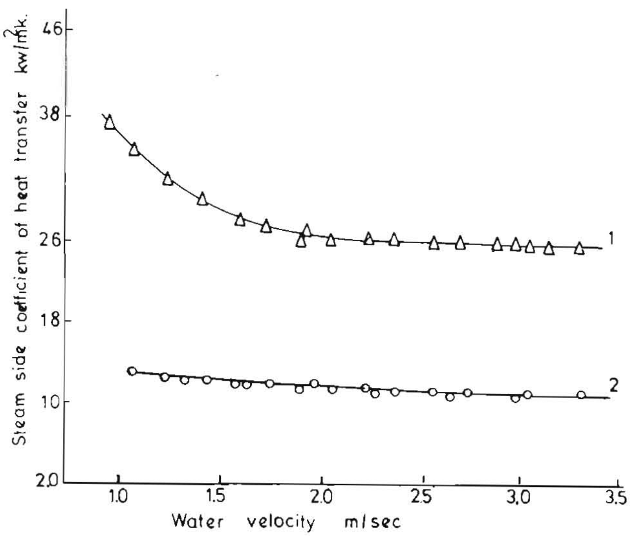

Fig. 3. Comparison between jetwise and non-jetwise steam flow
(1) Jetwise flow, steam manifold pressure = 110 kN/m^2
(2) Stationary steam.

in the condenser shell was also kept constnat to maintain a vapour saturation temperature of 374 K. The manifold steam perssure was 110 kN/m^2 with 2-3 deg.K of superheat. The experimental results are plotted on Fig. 3.

For the same cooling water velocity, the condensate film heat transfer coefficients produced by the jetwise flow were always higher than those for normal flow. The improvement in the film heat transfer coefficient ranged between 140 and 180 percent for cooling water velocities of 3.3 to 1.1 m sec^{-1}, respectively.

The visual observations revealed that for jetwise flow the hydrodynamics of the condensating film were completely disturbed. Instead of having a continuous increasing condensate film as for the normal flow, the film was broken into a series of ripples producing a very thin condensate film facing the manifold tube. The decrease in the thickness of the condensate film was due to the combined effect of the momentum and frictional drags imposed by the high velocity jets which were impinging on the slow moving condensate film.

It is evident from Fig. 3 that for the jetwise flow and a water velocity below 1.6 m sec^{-1}, the steamside coefficient of heat transfer was strongly increasing with decreasing water velocity. For small water velocities, the heat flux was low and the condensate film was much thinner. Thus, the direct impingement of the high velocity vapour jets will increase the outside tube wall temperature significantly, resulting in an increase in the film heat transfer coefficient.

Set 2

The effect of the orifice size on the intensity of heat transfer was investigated. Five groups of experiments with orifices of diameter 0.7, 0.8, 0.9, 1.04 and 1.55 mm, respectively, were examined. In each group, the steam manifold tube consisted of 170 identical orifices. For the whole set, the inlet cooling water temperature was 294 $\pm$ 2 K and the water velocities ranged from 3.3 to 1.1 m sec^{-1}. For each orifice size, the steam pressure in the manifold tube was adjusted to maintain a constant vapour saturation temperature of 375 K in the condenser shell. Steam in the manifold tube was always superheated by 2-3 deg. K.

It is evident from Fig. 4 that orifices of small diameter, which required a high pressure drop to maintain a constant vapour pressure of about 106 kN/m^2 in the condenser shell, were producing high film heat transfer coefficients. The high pressure drop will genenerate jets of high vapour velocities (Bernoulli's effect) which would sweep the condensate film off the cooling surface and lower its thermal resistance.

Figure 4 also shows that large orifices of diameter 1.55 mm almost gave the same heat transfer results as those obtained by normal flow under the same cooling conditions. This indicates that in this case the low velocity vapour jets striking the cold tube surface were unable to influence the condensate film thickness, resulting in a low film heat transfer coefficient.

Set 3

In this set, the orifice sizes which gave the highest improvement in the intensity of heat transfer, which had been determined to be 0.07 cm in set 2, was used to investigate the effect of the variation of the manifold steam pressure on the intensity of heat transfer. Five groups of experiments were conducted using pressure values of 108.22, 121.64, 144.30, 156.48 and 169.79 kN/m^2, respectively. For each group, the manifold steam pressure was superheated by 2-3 deg. K and the inlet cooling water temperature was about 295 K.

The experimental results are plotted on Fig. 5. It was observed that for a constant cooling water velocity, high manifold steam pressure produced high steam film heat transfer coefficients. Increasing the steam pressure would increase the vapour jet velocities which in turn would decrease the steamside thermal resistance as shown on Fig. 6. The dependence of the heat transfer enhancement on the vapour pressure loss is shown on Fig. 7.

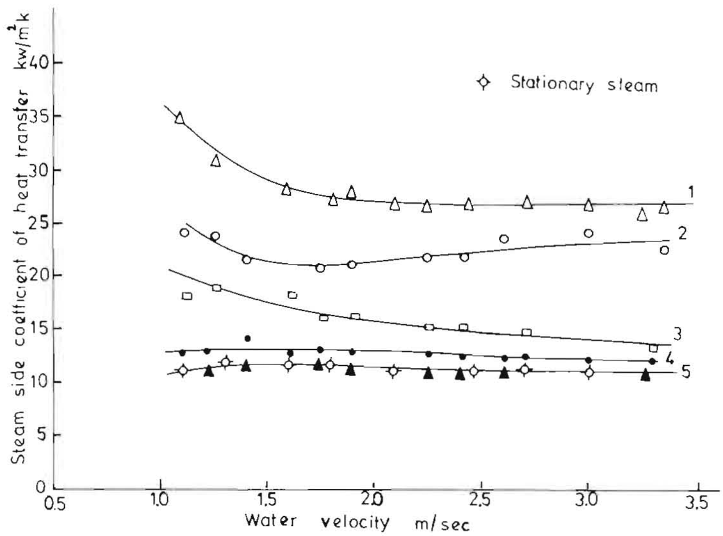

Fig. 4. The effect of orifice size on the steamside coefficient of heat transfer
(1) Orifice diameter D = 0.7 mm, manifold steam pressure P = 110.0 kN/m^2;
(2) D = 0.8 mm; P = 107.6 kN/m^2;
(3) D = 0.9 mm, P = 106.6 kN/m^2;
(4) D = 1.04 mm, P = 106.3 kN/m^2;
(5) D = 1.55 mm, P = 106,1 kN/m^2.

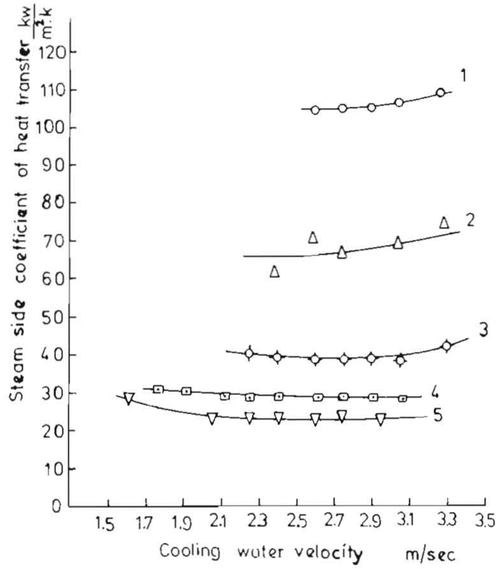

Fig. 5. Effect of steam manifold pressure P on the steamside coefficient of heat transfer
(1) P = 169.74 kN/m^2;
(2) P = 156.46;
(3) P = 144.3;
(4) P = 121.64;
(5) P = 108.22.

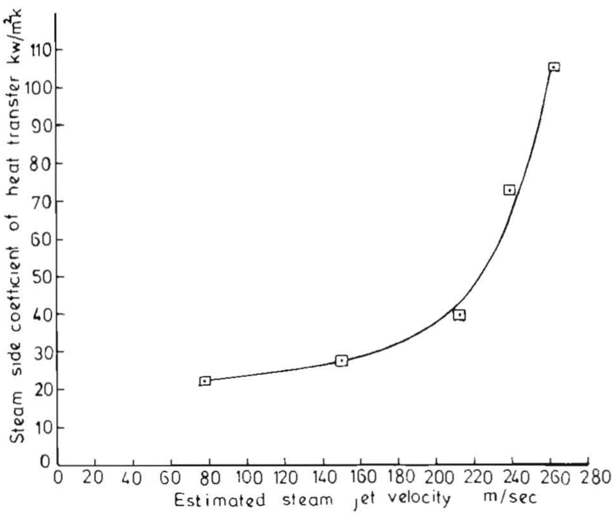

Fig. 6. The effect of the steam jet velocity on the steamside coefficient of heat transfer (Cooling water velocity = 3.0 m/sec).

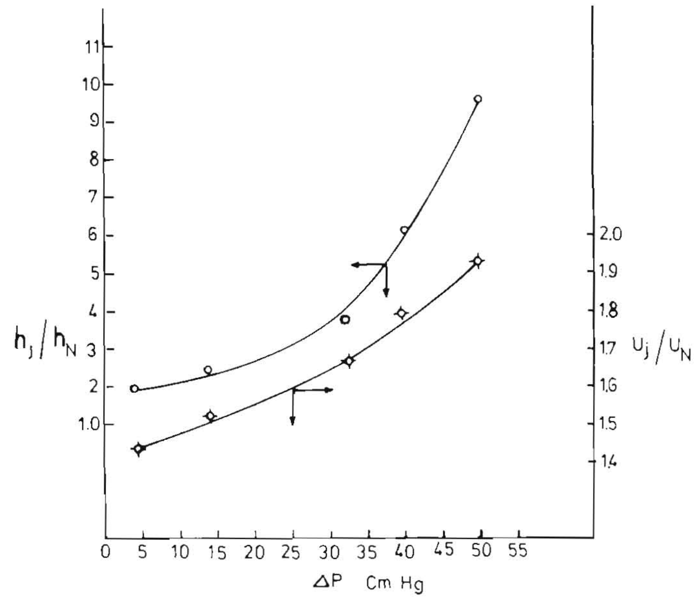

Fig. 7. The dependence of the heat transfer enhancement on the jet driving force (ΔP) (Cooling water velocity = 3.0 m/sec).

When high pressure steam is available, *e.g.* utilization of high pressure exhaust steam in a waste heat recovery system, the concept of heat transfer enhancement by jetwise flow may be feasible. Current investigations are extending the single tube condenser system to a multitube system. Economic factors are also being studied in detail.

Conclusions

1. The experimental findings indicate that the jetwise flow approach has substantially enhanced the steamside coefficient of heat transfer. The enhancement was believed to be due to the combined effect of the momentum and frictional drags exerted by the high velocity steam jets on the slowly moving condensate film.

2. The results indicate the high dependence of the heat transfer improvement by the jetwise flow on the vapour pressure loss.

3. The concept of heat transfer enhancement utilizing jetwise flow may find applications in the design of heat transfer systems where high steam pressure is readily available.

Nomenclature

d Condensing surface diameter
h Film heat transfer coefficient
K degree kelvin
k thermal conductivity
kW kilowatt
Nu Nusselt number
Re Reynold number
Pr Prandtl number
U Overall heat transfer coefficient
$\bar{V}$ average velocity
x tube thickness
ϱ density
μ absolute viscosity at the bulk conditions
P pressure

Subscripts

j Jetwise flow
w tube wall
N normal flow
O outside
m mean
i inside
S_j steamjet
2 condenser shell
1 manifold tube.

References

Brochman, J. (1965) *Condensation of High Velocity Vapours of Steam and Freon-11*, Diss. Techn., University of Karlsruhe, Germany, 315 p.

Chance, J.L. (1974) Experimental investigation of air impingement heat transfer under an array of round jets, *Tappi*, **57** (6) : 108-112.

Coulson, J.M. and **Richardson, J.F.** (1977) *Chemical Engineering*, Vol. 1, Pergamon Press, p. 26.

Furman, T. and **Hampson, H.** (1959) Experimental investigation into the effect of cross flow with condensation of steam and steam-gas mixtures on a vertical tube, *Proc. Inst. mech. Engrs* **173** : 149-169.

Garden, R. and **Cobonpue, J.** (1962) Heat Transfer between a flat plate and jets of air impinging on it, *Proc. of 2nd International Heat Transfer Conference*, ASME, New York, N.Y. : 454-460.

Hamed, O.A. (1972) *Forced Convective Heat Transfer in the Condensation of Steam and Other Vapours*, Ph.D. Thesis University of Newcastle, England, 176 p.

Hollworth, B.R. and **Berry, R.D.** (1978) Heat transfer from arrays of impinging jets with large jet-to-jet spacing, *Trans. Am. Soc. mech. Engrs* **100** : 352-357.

Huang, G.C. (1963) Investigation of heat transfer coefficients for air flow through round jets impinging normal to a heat transfer surface, *Am. Soc. mech. Engrs J. Heat Transf.* **85** : 237-243.

kern, D.Q. (1965) *Process Heat Transfer*, McGraw-Hill Kogakusha, Tokyo, 871 p.

Le Fevre, E.J. and **Rose, J.W.** (1969) Dropwise condensation, *in*: **Donaldson, R. (ed.)**. *Bicentary of the James Watt Patent for a Separate Condenser for the Steam Engine*, University of Glasgow, Glasgow, 224 p.

Leonard, W.K. and **Joseph, E.** (1967) The effect of vapour velocity on condensation on a vertical surface, *A.I.Ch.E.J.* **13** : 401-402.

O'Bara, J.T., Killian, E.S. and **Roblee, L.H.S., Jr.** (1967) Dropwise condensation of steam at atmospheric and above atmospheric pressures, *Chem. Eng. Sci.* **22** : 1305-1314.

Osment, B.D.J., Speirs, R.R.M. and **Ragman, W.** (1965) *The Effect of Baffles and Honeycomb Grids in the Steam Inlet Branch of the Condenser*, National Engineering Laboratory, Glasgow, Report No. 193.

Rohsenow, W.M. and **Chio, H.Y.** (1961) *Heat, Mass and Momentum Transfer*, Prentice Hall, 537 p.

Silver, R.S. and **Simpson, H.C.** (1961) The condensation of superheated steam, *Proc. of a Conference Held at the National Engineering Laboratory, Glasgow* : 39-69.

Simpson, H.C. (1969) *in*: **Donaldson, R. (ed.)** *Bicentary of the James Watt Patent for a Seperate Condenser for the Steam Engine*, University of Glasgow, Glasgow, 224 p.

Thomas, D.G. and **Hayes, P.H.** (1970) High performance heat transfer surfaces, *Ind. eng. Chem.* **62** (2) : 4-9.

Williams, A.G., Nandapurkar, S.S. and **Holland, F.A.** (1968) A review of methods for enhancing heat transfer rates in surface condensers, *Chem. Engr* **233** : 367-373.

Zijnen, B.G.V. (1951) Flow through uniformly tapped pipes, *Appl. Sci. Res.* **A3** : 144-162.

(Received o4/04/1983;
in revised form 06/03/1984)

الحِمْلُ الحرارى القسرى أثناء تكثيف البخار النقى المنفوث على أُنبوب رأسى

عثمان أحمد حمد وجورج عياد سوريال

قسم الهندسة الكيميائية - كلية الهندسة - الجامعة الأردنية - عمان - الأردن ؛ وكلية التكنولوجيا - جامعة الجزيرة - وادمدنى - السودان .

أُجريت الفحوصات التجريبية لدراسة تأثير نفث بخار الماء على سطح أنبوبي مفرد رأسي . شملت المتغيرات قطر الثقب ، الكمية الحرارية وسرعة البخار. ولقد تم التركيز على عمل مقارنة بين البخار الثابت والبخار المنفوث بسرعات عالية .

أظهـرت النتائج أن معامل الانتقال الحرارى الفردى للبخار المنفوث كان يتر واح ما بين **٢,٠** إلى **٩,٥٥** أضعاف نظيره للبخار الثابت. وكان الفاقد في ضغط البخار **٤,٧** و **٥٠** سم زئبق على التوالي .

بيَّن البحث الصلة الوثيقة بين تحسين الانتقال الحرارى وقوة دفع البخار المنفوث، يمكن تطبيق نظام البخار المنفوث في الصناعات التي بها بخار ماء ذو ضغط عال .