HEAT LOSSES IN LONG HOT-WATER PIPELINES DURING TEMPERATURE VARIATION CYCLES

By

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The recent technical development and especially changes in the structure of energy carriers raise quite new aspects in the establishment and operation of heating power plants and extensive district heating systems.

Also calculation methods have to be accommodated to the novel intermittent operating conditions of long transmission pipelines, to investigate cooling and heat-loss processes therein.

1. Problems arising, justification of the periodical origin of temperature cycles

For the investigation of the establishment and running conditions of heating power stations and district heating equipment, new, up-to-date methods are badly needed.

For either back-pressure or condensing type plants, energy output always depends on the heat supply. By operating heat supply plants for peak power, both their competitiveness can be increased and their electric power capacity can help meet peak demands.

By now, electric peak power operation of heating power plants along these lines has become usual; in time of peak load the electric plant supplies the maximum of electric power possible, because heating relies on the heat storage capacity of both the extended network and the heated establishments.

During the peak hours the hourly heat output of the station may be either more or less than actually needed. It matters of course, whether the power plant is of the backpressure or the condensation type. The daily total heat output corresponds to the heat demand depending on the daily average temperature, even in the period of peak consumption. In this way, during the period of peak load, the heat supply station can make full use of its established maximum power capacity and at the same time meet the heat consumers' needs [1, 2, 3].

Many are the parameters that may influence the method of peak power operation. For the sake of illustration two typical methods will be described.

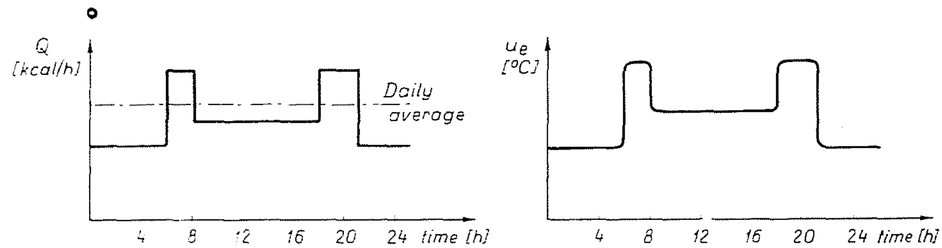

Fig. 1

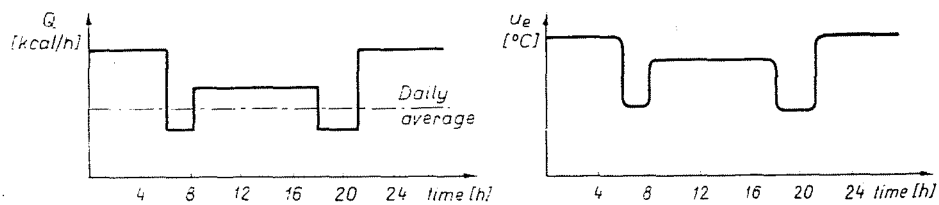

Fig. 2

The so-called backpressure (Fig. 1) and condensing-type (Fig. 2) power plants are characterized by "positive" and "negative" peak operation, respectively. In Figs. 1 and 2, the corresponding supplied heat and on flow cycles water temperature are illustrated for a period of 24 hours.

Both methods of operation are linked to peculiar problems in connection with the maximum exploitation.

2. Problems

Problems may be related with

— the establishments to be supplied with heat energy for heating or other purposes; and

— the running conditions of the transmission pipelines.

For problems relating to district-heated construction see [4] and [5] as well as a paper of ours now under preparation. Here the behaviour of a pipeline in cyclic operation will be investigated.

It has to be known first:

a) How much time needs the hot water to arrive at a given consumer, and at what temperature?

b) What should be the starting water temperature to deliver water at the temperature wanted by that customer?

c) What is the 24 hours' heat loss throughout a pipeline in cyclic operation as compared with that in continuous service conditions?

3. Composition of the investigation

a) A general solution has been found, permitting to derive a corresponding relationship.

b) In a series of tests, the fall in temperature has been measured along the long hot-water pipeline to find the periodical variations.

c) By substituting the obtained characteristic data for a certain long pipeline into the derived formulae, the fall in temperature at predetermined points during a cycle could be assessed.

d) Comparison between measured and calculated values supported the suitability of our mathematical model and pointed out some new ways of computation. In addition, some interesting features of this method have been recognized: these will be discussed later.

e) To help design work a system of nomographs has been constructed and recommendations are given for their correct use.

4. The general solution

The general condition is described by a differential equation, written with the intermediary of some simplifying assumptions that seemed in our experiments to be justified because

a) in any cross section, the temperature distribution is but a function of time reducing the problem to an unidimensional one;

b) the internal heat-transfer by convection is negligible;

c) the temperature of the consumer station (heated object) at the end of the pipeline has no back action on the temperature distribution in the pipeline water; hence the distribution corresponds to a pipeline infinite in one direction;

d) the "heat source" for the backward line is the heat consumer, namely, from what was told under c), it is obvious that the forward water conduit and the backward water conduit should be considered separately;

e) the initial distribution of temperature in the water conduit is negligible: an assumption inherent with the nature of the process;

f) the temperature around the pipeline is constant.

The temperature distribution in the flowing fluid is described by the partial differential equation [6]

$$c\varrho \frac{\partial u}{\partial t} = \lambda \frac{\partial^2 u}{\partial x^2} - c\,\varrho\, w \frac{\partial u}{\partial x} - k \frac{p}{q} (u - u_k) \tag{1}$$

where

u [°C]	— water temperature;
u_k [°C]	— ambient temperature;
t [h]	— time;
x [m]	— longitudinal coordinate of the pipeline;
c [kcal/kg °C]	— specific heat;
ϱ [kg/m³]	— density of water;
λ [kcal/m h °C]	— heat conduction coefficient of water;
w [m/h]	— velocity of water;
k [kcal/m² h °C]	— heat transfer factor referred to the pipe outer surface;
p [m]	— outer circumference of the pipe;
q [m²]	— area of the inner cross section of the pipe.

On the right side of the partial differential equation (1), the first member is the heat conduction in the fluid, the second member the temperature variation upon flow and the third one the temperature drop due to heat transfer to the surroundings.

The predetermined temperature of the water fed into the pipeline at the beginning (Fig. 1) varies with time. This is expressed by the boundary condition:

$$u(0, t) = g(t) \tag{2}$$

the function $g(t)$ being known.

The orders of magnitude of the coefficients in Eq. (1) are evident from the following list:

$$c\varrho = 10^3 \text{ kcal/kg m}^3$$
$$\lambda = 5 \cdot 10^{-1} \text{ kcal/m h °C}$$
$$c\varrho w = 3 \cdot 10^6 \text{ kcal/m}^2 \text{ h °C}$$
$$k\frac{p}{q} = 2 \cdot 10^2 \text{ kcal/m}^2 \text{ h °C}$$

The coefficient of the member for heat conduction is seen to be negligible, at least in first approximation because its order of magnitude is less by at least 10^3 than those of the others. Consequently, Eq. (1) simplifies into:

$$\frac{\partial u}{\partial t} = -w\frac{\partial u}{\partial x} - \frac{kp}{c\varrho q}(u - u_k) \tag{3}$$

Let us introduce the notations:
overtemperature: $u - u_k = \vartheta$

and

$$\frac{k \cdot p}{c \varrho w} = b$$

thus

$$\frac{\partial \vartheta}{\partial t} + w \frac{\partial \vartheta}{\partial x} + b \vartheta = 0 \tag{4}$$

where

$$w > 0 \quad \text{and} \quad b > 0$$

Let us introduce, by transformation, the following independent variables:

$$\xi = \frac{x}{w} - t \qquad \text{and} \qquad \eta = \frac{x}{w} + t$$

Thus overtemperature is expressed by the ordinary differential equation:

$$\frac{\partial \vartheta}{\partial \eta} - \frac{b}{2} \vartheta = 0 \tag{5}$$

The general solution of Eq. (5):

$$\vartheta = C(\xi)\, e^{-\frac{b}{2}\eta}$$

where C arbitrary function of the variable has to comply with actual conditions.

With the original variables x and t, the above solution will take the form:

$$\vartheta = C\left(\frac{x}{w} - t\right) e^{-\frac{b}{2}\left(\frac{x}{w} + t\right)} \tag{6}$$

Boundary condition (2) can be rewritten as:

$$\vartheta(0, t) = f(t) \tag{7}$$

where

$$f(t) = g(t) - u_k$$

Therewith Eq. (6) takes the form:

$$C\left(\frac{x}{w} - t\right) = f\left(t - \frac{x}{w}\right) e^{\frac{b}{2}\left(t - \frac{x}{w}\right)}$$

Thus, the overtemperature

$$\vartheta = f\left(t - \frac{x}{w}\right) e^{-\frac{b}{w}x} \tag{8}$$

The boundary condition at the end point of the pipeline, where $x = l$, is

$$\vartheta(l, t) = h(t) \tag{9}$$

In view of (6) we obtain:

$$C\left(\frac{x}{w} - t\right) = h\left(t + \frac{l - x}{w}\right) e^{\frac{b}{2}\left(\frac{2l-x}{w} + t\right)}$$

hence

$$\vartheta = h\left(t + \frac{l - x}{w}\right) e^{\frac{b}{w}(l-x)} \tag{10}$$

This equation determines the temperature of the water to be fed in, necessary to obtain hot water of prescribed temperature at the end of the pipeline; naturally, both temperatures are expressed as a function of time:

$$\vartheta(0, t) = h\left(t + \frac{l}{w}\right) e^{\frac{b}{w}l} \tag{11}$$

Prior to explaining the application of the general solution, the validity of the above relationship will be supported by its physical interpretation.

Let us analyze Eq. (8) referring to the temperature drop:

$$\frac{\vartheta}{f\left(t - \frac{x}{w}\right)} = e^{-\frac{b}{w}x}$$

a) For an infinite velocity of waterflow $w = \infty$, $e^{-\frac{b}{w}x} = 1$, there is no temperature drop;

b) for an absolutely perfect pipe insulation, with a zero heat transfer factor $b = \frac{kp}{c\varrho q} \to 0$, or $e^{-\frac{b}{w}x} = 1$, there is no temperature drop;

c) for $x = 0$, of course, there can be no cooling at this point either.

5. Observation and measurement data of the cooling process along a given long pipeline for cyclic temperature variations

In Table 1, the dimensional and lay-out data of a given long pipeline are compiled. The main line serves for supplying a heating system of $u_e = 150\ °C$ inlet temperature and $u_r = 80\ °C$ return temperature; column 4 contains design heat quantities based on the calculated dimensions for the given temperature conditions 150/80 °C.

(As already mentioned, the tested main line is a part of an entire network, since there are two other main lines for other directions.)

The process of measurements and the results are shown in Fig. 3. (Notice that these measurements had other objects too and referred to various aspects; in this paper, all these are passed by for the sake of uniform treatment.)

Fig. 3 truly reflects the retarding movement of the water and the temperature drop.

The following extreme average values of temperature were found:

$\Delta\vartheta$ — measured average of the maxima	17.1 °C
$\Delta\vartheta$ — measured average of the minima	8.5 °C

For the sake of better understanding, curves in Fig. 4 show results modified in Figs. 1 and 2.

6. Measurement of the temperature drop. Comparison between measured and calculated values

For cooling-down calculations, first the rate of water flow in the line sections has to be known since all the other quantities in Eq. (8) are either known, or calculable from values in Table 1.

Based on the relation in Fig. 3, the time needed by the wave of temperature variation to reach the end of the pipeline could be stated. According to our measurements:

$$t_{\text{meas.}} = t_{\text{eff.}} = 3.9 \text{ h} \cdot 3600 = 14\,040 \text{ sec}$$

Relying on data in Table 1, the quantity of water to pass the single sections of the tabulated dimensions has been calculated (column 2, Table 2) together with the design flow rate (column 3, Table 2); from these rates and the lengths of the single sections, the time has been calculated where the water would reach the end point of the section (column 4, Table 2). Addition of the sectional times delivers the total time, where the water would reach the end point of the pipeline:

$$t_{\text{calc.}} = 8013 \text{ sec}$$

Consequently, the ratio of design to effective quantities of water passing the pipeline is:

$$\frac{t_{\text{calc.}}}{t_{\text{meas.}}} = \frac{t_{\text{design}}}{t_{\text{eff.}}} = \frac{8013}{14\,040} = 0.566$$

Let us stop here for a moment, to see the special advantages offered by operating a power plant for electric peak load. In general, our long pipelines


u_e [°C]
140
130
120
110
100
90
80
70
6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 1 2 3 4 5 6
Time [h]
Initial temperature u_e at starting point
Temperature arriving u_e at the endpoint

Fig. 3

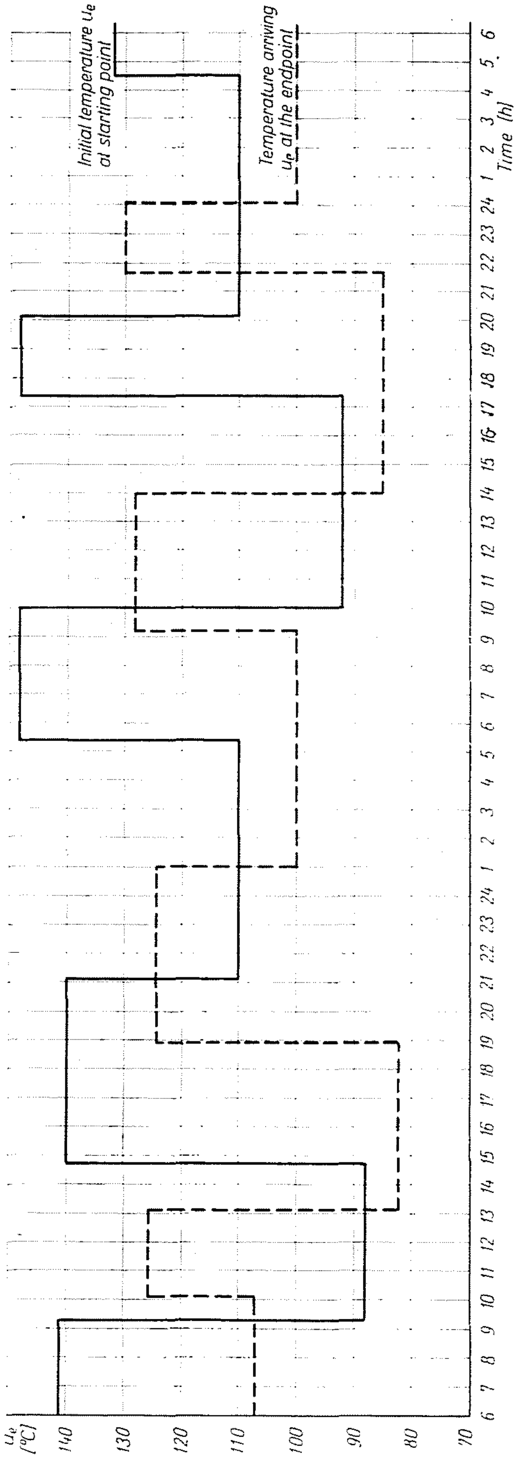
Initial temperature u_e at starting point
Temperature arriving u_e at the endpoint
Time [h]
u_e [°C]
140
130
120
110
100
90
80
70
6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 1 2 3 4 5 6

Fig. 4

Table 1

Line section No.	Diameter of section d [mm]	Length of section [m]	Heat quantity supplied $\dot{Q}$ [kcal/h]	Insulation kind and thickness	Layout
1	300	8	45 986 000	Glass wool. 10 cm thick	Over the grade; overhead line
2		80	35 986 000		
3		120	35 861 000		
4		160	35 441 000		
5		16	35 174 000		
6		84	33 632 000		
7		84	32 532 000		
8		40	31 917 000		
9		4	30 062 000		
10		90	28 828 000		
11		70	22 778 000		
12		100	19 347 000		
13		108	17 017 000		
14		200	16 127 000		
15		280	14 483 000		
16		66	13 103 000		
17		72	13 044 000		
18		74	11 854 000		
19		30	9 856 000		
20		142	9 816 000		
21		176	9 736 000		
22	250	270	9 673 000		
23		64	9 486 000		
24		22	8 736 000		
25		176	7 536 000		
26	200	574	4 436 000	Glass wool 7 cm thick	Underground in concrete channel
27	150	333	1 433 000		
28	100	220	300 000	Glass wool 4 cm thick	
29	65	114	125 000		

miss the necessary measuring instruments, so they do not lend themselves for measurements. In fact, as was shown by the presented example, a single test series on a peak operating plant demonstrated the inherent unsteadiness of the water supply, giving thereby hints on how to regulate the system. As a matter of fact, no further financial means are needed, neither to disassemble the pipeline anywhere.

In this relation, remind that the tested network comprises two other main lines. Without further precisions the measurements delivered the necessary information on the unsteadiness and insufficiencies of the heating implements connected to the main lines. Based on the ratio

$$\frac{t_{\text{design}}}{t_{\text{eff.}}}$$

a formal simplification has been introduced: some sections that seemed to be too short have been united: this contraction is realized in Table 3, showing the new decisions of the main line.

In Table 3, the effective cooling process data are tabulated. Based on the ratio t_{design} to t_{eff}, the effective time (column 2) where the water was at the end point of each section, and the rate (columns 3 and 4) could be calculated (in m/s and m/h); in column 5, the values of the exponent

$$\frac{b}{w}x = \frac{k \cdot p}{c \varrho q} \cdot \frac{l}{w}$$

Table 2

Line section No.	Theoretical heat quantity supplied: $\dot{V} = \frac{\dot{Q}}{c \cdot \varrho \cdot \Delta u}$ [m³/h]	Theoretical flow speed: w [m/s]	Theoretical flow time: [sec]
1	695	2.66	3.0
2	545	2.1	38.0
3	540	2.1	57.0
4	533	2.1	76.0
5	532	2.1	7.6
6	510	2.0	42.0
7	485	1.86	45.1
8	480	1.85	21.6
9	454	1.74	2.3
10	440	1.69	53.2
11	335	1.28	54.7
12	291	1.12	90.0
13	257	0.99	110.0
14	244	0.94	213.0
15	220	0.84	333.0
16	198	0.76	87.0
17	197	0.75	95.0
18	182	0.68	115.0
19	149	0.57	52.6
20	148	0.57	249.1
21	146	0.56	314.3
22	145	0.8	337.5
23	144	0.8	80.0
24	132	0.74	16.5
25	114	0.63	220.0
26	67	0.59	973.0
27	21	0.33	1010.0
28	4.5	0.16	1375.0
29	0.9	0.064	1781.2

Table 3

Line section No.	Effective time [sec]	Effective speed w [m/s]	w [m/h]	$\frac{b}{w} x$	Maxima ϑ_e [°C]	ϑ_v [°C]	Minima ϑ_e [°C]	ϑ_v [°C]
1	5.2	1.52	5480	0.00184	144.0	143.7	100	99.8
2	378.0	1.19	4290					
3	118.0	1.05	3780	0.00058	143.7	143.7	99.8	99.7
4	4.1	0.98	3530	0.00049	143.5	143.3	99.7	99.6
5	95.0	0.95	3420					
6	98.0	0.715	2575	0.00049	143.3	143.1	99.6	99.5
7	161.0	0.62	2240	0.0008	143.1	142.9	99.5	99.4
8	197.0	0.55	1980	0.0009	142.9	142.7	99.4	99.3
9	385.0	0.52	1870	0.00019	142.7	142.3	99.3	99.1
10	590.0	0.475	1710	0.0030	142.3	142.0	99.1	98.8
11	328.0	0.42	1510	0.0016	142.0	141.7	98.8	98.5
12	191.0	0.386	1390	0.0009	141.7	141.5	98.5	98.4
13	1080.0	0.322	1160	0.00054	141.5	141.3	98.4	98.3
14	734.0	0.455	1640	0.0049	141.3	140.8	98.3	97.8
15	52.2	0.42	1510					
16	496.0	0.355	1280	0.0031	140.8	140.3	97.8	97.5
17	1740.0	0.33	1190	0.0094	94.3	93.4	51.5	51.05
18	1790.0	0.186	670	0.0144	93.4	92.0	51.05	50.3
19	2470.0	0.089	320	0.0465	92.0	87.9	50.3	48.0
20	3080.0	0.037	133	0.0778	87.9	81.3	48.0	44.4

are tabulated. Based on these values, the maxima and minima of the temperature drop at the respective section points could be calculated.

Calculated and measured data were in a fair accordance, the calculated values being:

$\Delta\vartheta$ average of the maxima: 16.7 °C
$\Delta\vartheta$ average of the minima: 9.6 °C

Some further comments seem to be useful:

a) The heat transmission factor of 1 linear meter of pipeline is expressed by the formula:

$$k = \frac{\pi}{\frac{1}{\alpha_b d_b} + \frac{1}{2\lambda_c} \ln \frac{d_k}{d_b} + \frac{1}{2\lambda_s} \ln \frac{d_s}{d_k} + \frac{1}{\alpha_k d_s}} \tag{12}$$

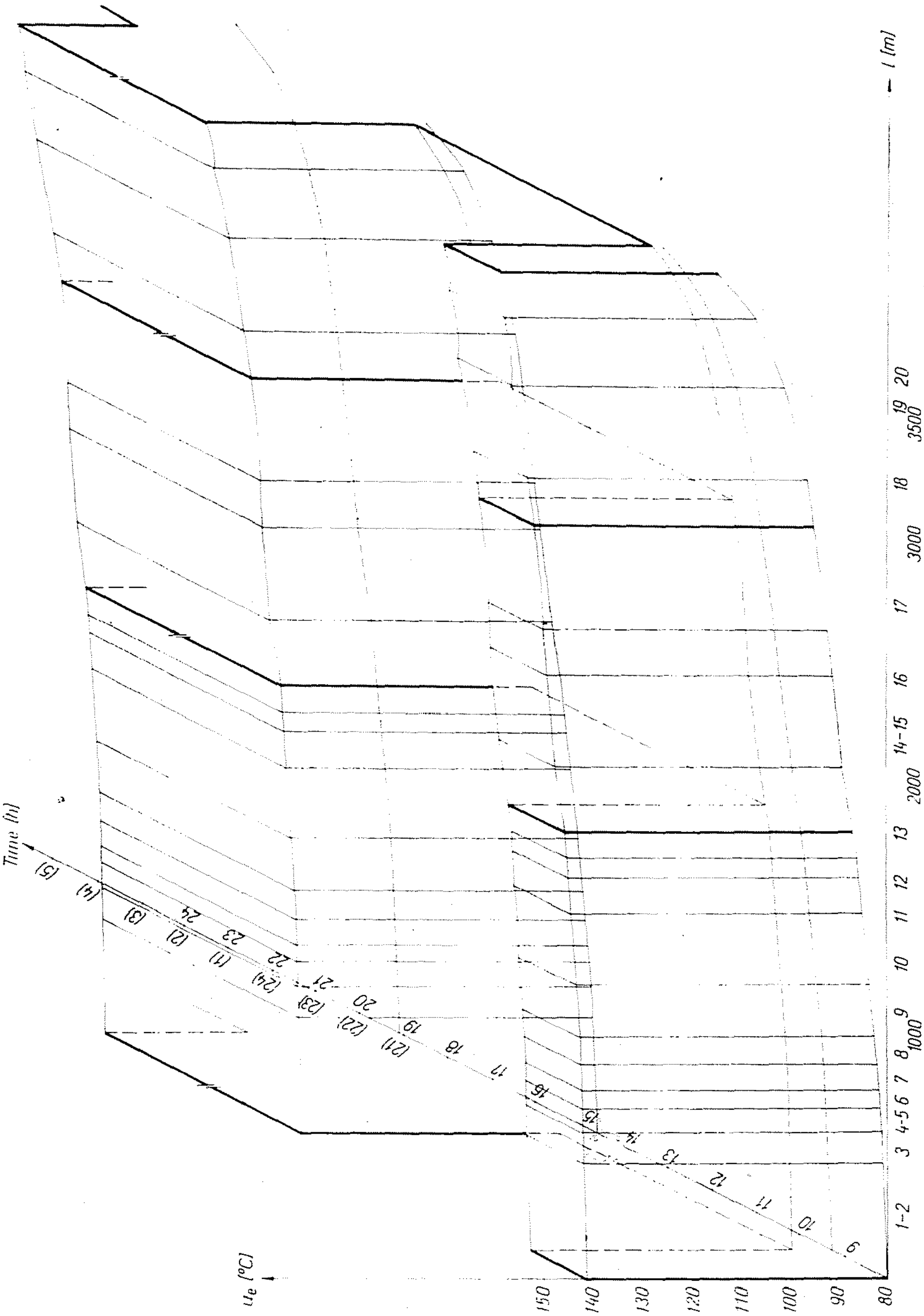

Fig. 5

where

d_k	[m]	— pipe outer diameter;
d_s	[m]	— insulation outer diameter;
α_b	[kcal/m² h °C]	— internal heat transfer coefficient;
d_b	[m]	— pipe inner diameter;
λ_c	[kcal/m h °C]	— heat conduction coefficient of the pipe material;
λ_s	[kcal/m h °C]	— heat conduction coefficient of the insulating material;
α_k	[kcal/m² h °C]	— external heat transfer coefficient.

The above factor k' referring to 1 linear meter, has to be transformed into the factor k, referring to unit area of the outer surface; thus, for each section, the multiplier m is found from the proportion:

$$m = \frac{\text{length (linear meter)}}{\text{outer surface (square meter)}}$$

referring, of course, to the insulation's outer surface. Thus: $k = k' \cdot$ m kcal/m³ pipe outer surface, h, °C.

b) Our calculations always refer to the value of the overtemperature. For pipelines in the open, the relevant value equals the absolute one, since, in our experimental work, the measured value of the ambient temperature was accidentally 0 °C ($u_k = 0$).

On the other hand, the ambient temperature in the protecting concrete channel had to be estimated. To this end, an approximative method has been applied [8], giving in this particular case $u_k = 46$ °C. Using a system of spatial coordinates, we obtained the diagram in Fig. 5, plotting the time of retardation of the water movement vs. temperature drop values.

A further relationship has emerged. Namely, designers were seen to ignore the temperature drop along the pipeline; the design of conventional heating and heat supply equipment is based only on the temperatures of water supplied and returning water. As a matter of fact most of our heating equipment are likely to be over-dimensioned, from these or other causes. In our opinion, however, a temperature drop of 15 to 20 °C should be reckoned with: it would be even more adequate for the designers to reckon with all the relevant physical and technical phenomena throughout the design work. Such an exact and meticulous calculation would yield better results than the actual practice by assuming that neglections in either direction offset each other. In this way a good operation of the system might be taken for granted.

No correct planning and design work can be done without taking decisive economic aspects into consideration. All the influencing factors should be cleared in a thorough investigation, such as this of ours.

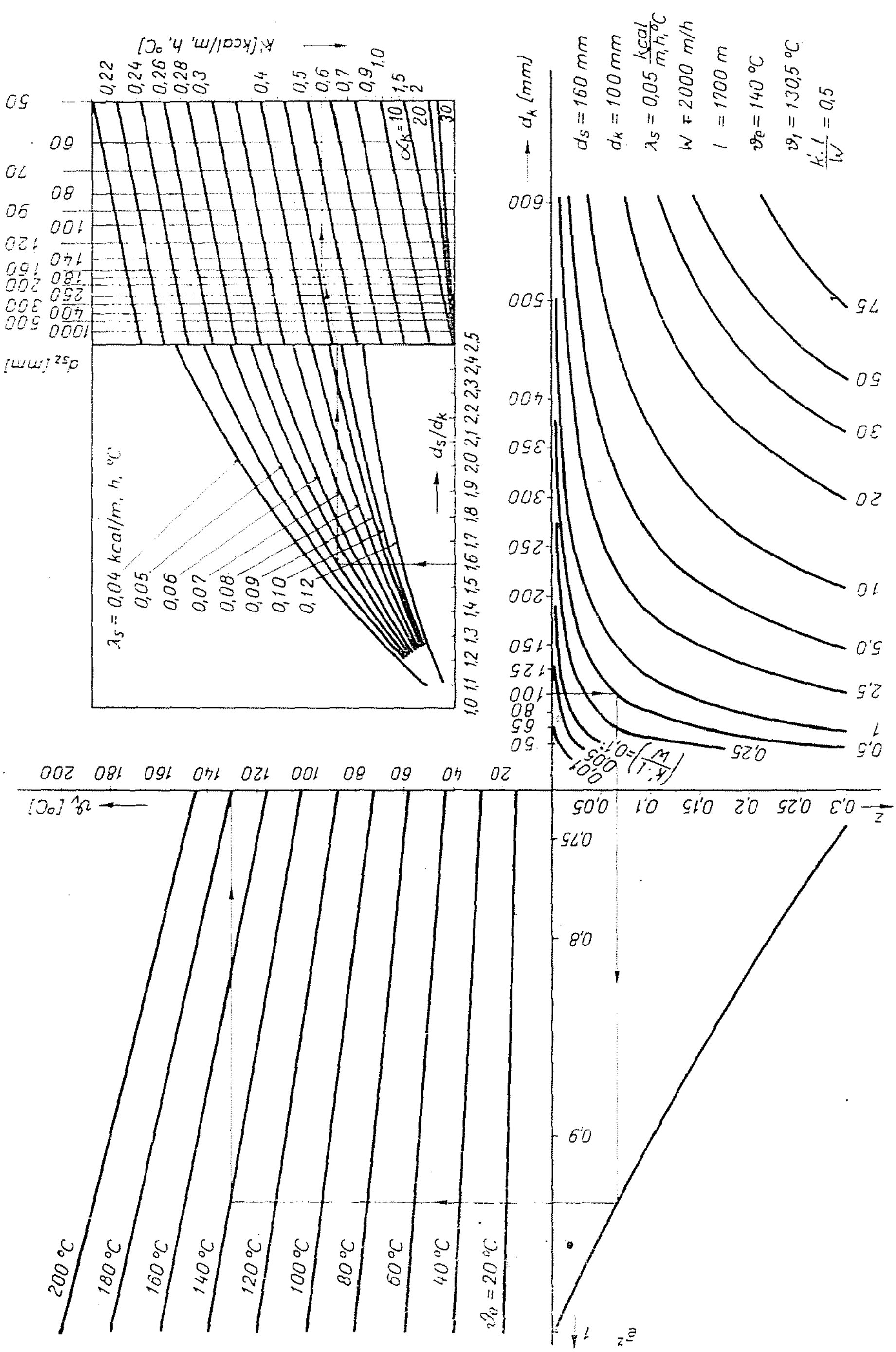
$d_s = 160$ mm
$d_k = 100$ mm
$\lambda_s = 0{,}05 \frac{kcal}{m, h, °C}$
$W = 2000$ m/h
$l = 1700$ m
$\vartheta_e = 140$ °C
$\vartheta_1 = 130{,}5$ °C
$\frac{k \cdot l}{W} = 0{,}5$
d_k [mm]
d_{sz} [mm]
d_s/d_k
k [kcal/m, h, °C]
$\lambda_s = 0{,}04$ kcal/m, h, °C
ϑ_e [°C]
$\vartheta_a = 20$ °C

Fig. 6

7. Nomograph

Practical designers can find help for their calculating work from a nomograph covering the influence of several factors (Fig. 6).

The heat transmission coefficient k' referring to 1 linear meter of the pipeline can be estimated by using the top-right field of the graph. The symbols used in the graph are the same as for Eq. (12). For the bottom right field the value

$$k' \cdot \frac{l}{w}$$

characteristic for the given section is calculated first from this value and the overtemperature at the beginning of the respective section; the overtemperature at the end point of the same section can be found in the top-left field. An example is traced in the nomograph. This graph may be applied to compare the heat losses of long lines in peak operation and in normal service conditions.

For peak power operation, the requirements of the electric energy consumption are mainly decisive. Combining it with the heat balance level for 24 hours, the daily heat supply curve can be plotted. (Example see in Figs 1 and 2.) Since in general Hungarian systems operate by supplying a constant quantity of water, the formula of heat output is:

$$\dot{Q} = \dot{m} \cdot c\,(u_e - u_v) \tag{13}$$

where

$\dot{Q}$ [kcal/h] — heat quantity supplied;
$\dot{m}$ [kg/h] — flowing water quantity;
u_e, u_v [°C] — temperatures of the heating water and of the return water, respectively.

In other words, heat supply is proportional to the temperature drop. From Figs 1 and 2, the diagram in Fig. 7 can be plotted. By means of the curve of regulation (Fig. 8), the required temperatures of the heating water (u_e), and of the return water (u_v) can be determined for estimated outside temperatures to deliver the desired ambient temperature and the respective curves for the peak operation and the normal service will be drawn therefrom.

In normal operating conditions, the heat loss for 24 hours equals:

$$Q_{\text{iner, norm}} = (\dot{m} \cdot \dot{c} \cdot \Delta\vartheta_e + \dot{m} \cdot \dot{c} \cdot \Delta\vartheta_v)\, 24 \ \text{[kcal]} \tag{14}$$

where

$\Delta\vartheta_e$ [°C] — temperature drop in the forward pipeline;
$\Delta\vartheta_v$ [°C] — temperature drop in the return pipeline.

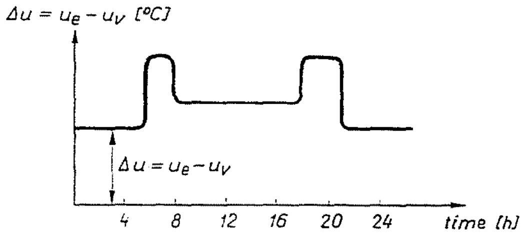

Fig. 7

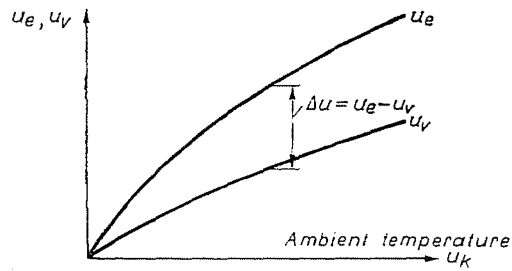

Fig. 8

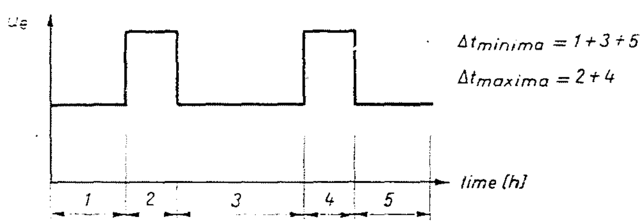

Fig. 9

In conditions of peak operation, the heat loss is:

$$Q_{\text{iner, peak}} = \dot{m} \cdot \dot{c} \cdot \Delta\vartheta_{e,\,\max} \cdot \Delta t_{\max} + \dot{m} \cdot \dot{c} \cdot \Delta\vartheta_{v,\,\max} \cdot \Delta t_{\max} + \\ + \dot{m} \cdot \dot{c} \cdot \Delta\vartheta_{e,\,\min} \cdot \Delta t_{\min} + \dot{m} \cdot \dot{c} \cdot \Delta\vartheta_{v,\,\min} \cdot \Delta t_{\min} \text{ [kcal]} \quad (15)$$

where

$\Delta\vartheta_{e,\,\max}$ [°C] — temperature drop in the forward line during peak periods;

$\Delta\vartheta_{v,\,\max}$ [°C] — temperature drop in the backward line during peak periods;

$\Delta\vartheta_{e,\,\min}$ [°C] — temperature drop in the forward line during minimum periods;

$\Delta\vartheta_{v,\,\min}$ [°C] — temperature drop in the backward line during minimum periods;

$\Delta t_{\max}$, $\Delta t_{\min}$ — periods (in h) of peaks and minima, respectively (Fig. 9).

The varying values of the temperature drops ϑ_e and ϑ_v can easily be found in the nomograph; thus, the values resulting from Eqs (14) and (15) can be compared; in other words, one can state, which of the three variants

$$\dot{Q}_{iner,norm} \gtreqless \dot{Q}_{iner,peak}$$

actually prevails.

Last but not least, the economic effect of a peak operation, namely, whether it is a factor decreasing or increasing losses will be obvious already when starting.

Acknowledgement

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Summary

A general method of computation to determine the temperature drop and the heat losses in hot water supply networks during cyclic temperature variation has been derived and justified by measurements, leading to the following advantages:

a) The time of arrival and the temperature of water arriving at a certain predetermined consumer can be estimated.

b) Once time and temperature are known, the right time of starting and the initial water temperature can be determined.

c) The heat loss can be calculated, and its respective values at normal service and in peak operation conditions can be compared.

d) Calculation and dimensioning is simplified by the use of a special nomograph.

e) In addition, a method lends itself to regulate an existing extended heat supply system erected without previous measurements.

f) Recommendations are given to increase design accuracy.

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