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Heat transfer in concentric annuli with moving cores—fully developed turbulent flow with arbitrarily prescribed heat flux

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1. INTRODUCTION

PROBLEMS involving fluid flow and heat transfer with the moving core of a solid body or liquid in an annular geometry can be found in many engineering practices, such as in manufacturing (e.g. extrusion, drawing and hot rolling), in transportation (e.g. trains travelling at high speed in a long tunnel or underground railways), in nuclear reactor operation (e.g. inverted annular film boiling) etc. In such processes, the fluid flow involved can be either laminar or turbulent and the moving body continuously exchanges heat with the surrounding environment.

In our previous studies, [1, 2], we presented the solutions on the problems of fully developed laminar and turbulent fluid flows and heat transfer in a concentric annulus with a moving core. For the laminar fluid flow [1], the solutions were obtained for the cases of one wall only heated with the other insulated and of the inner and outer tubes for any heat flux ratio. The solutions for the latter were obtained through the influence coefficients [3], which are evaluated from the fundamental solution from the definition.

For the case of turbulent fluid flow, the solution was presented for the condition of a constant heat flux at the inner core only with the outer wall insulated [2]. To complement this, the solutions are presented here, for the turbulent heat transfer in an annulus for the cases of the outer wall only heated with the inner wall insulated, and of the inner and outer walls for any heat flux ratio.

2. ANALYSIS

2.1. *Case for the outer wall only heated and the inner insulated*

For the prediction of temperature distribution and heat transfer rates, a modified mixing length model for flow turbulence is used for the analysis [2]. The intermediate details of the mathematical development of the analysis can be easily deduced from ref. [2] and therefore are not presented here. Appropriate adjustments were made in the present analysis for the matching conditions to accommodate the different boundary condition from that of the case for the inner wall only heated and the outer insulated.

2.2. *Cases for both walls heated independently*

For the cases where both wall surfaces are heated independently, the Nusselt numbers on the two surfaces for any heat flux ratio may be calculated, utilizing the superposition method [1, 3]. This is because the governing energy equation for the present study is linear and homogeneous. The Nusselt numbers for asymmetric heating are then obtained through influence coefficients [4] given as:

$$Nu_j = \frac{Nu_{jj}}{1-\theta_j^*(q_{Rk}/q_{Rj})} \tag{1}$$

where $j = i$ then $k = o$ and $j = o$ then $k = i$. Here, q_{Ri} and q_{Ro} are defined as positive into the fluid.

The Nusselt numbers, Nu_i and Nu_o, are defined as:

$$Nu_j \equiv \frac{h_j 2(R_o - R_i)}{k}. \tag{2}$$

The heat transfer coefficient is defined as:

$$h_j \equiv q_{Rj}/(T_{Rj} - T_b) \tag{3}$$

where $j = i$ or o.

The influence coefficients, θ_i^* and θ_o^*, are defined as [3]:

$$\theta_j^* \equiv \frac{(T_b - T_{Rj})_{kk} \Big/ \left[\dfrac{q_{Rk}\cdot 2(R_o - R_i)}{k}\right]}{(T_{Rj} - T_b)_{jj} \Big/ \left[\dfrac{q_{Rj}\cdot 2(R_o - R_i)}{k}\right]}, \tag{4}$$

where for Case A : $j = i$ and $k = o$ and for Case B : $j = o$ and $k = i$.

3. RESULTS AND DISCUSSION

The range of parameters considered are:

The radius ratio (α) : 0.2, 0.5, 0.8 and 0.99

The relative velocity (U^*) : 0.0, 0.2, 0.4, 0.6, 0.8 and 1.0

The Reynolds numbers (Re) : 10^4, 3×10^4, 10^5 and 10^6

Prandtl numbers (Pr) : 0.72, 10 and 50.

The results are presented in Table 1 and in Figs. 1 and 2. The numerical values for the parameters in the table seem not to change monotonously with increasing values of U^* and this could be due to the combined effects of U^*, α, Pr and Re.

The results for $U^* = 0.0$ are almost identical to those of Kays and Crawford [4]. The insignificant difference is due to the different turbulence models used in the analyses.

The predicted Nusselt numbers for the cases of one wall only heated with the other insulated, Nu_{jj}, for the range of the relative velocity, U^*, between 0.0 and 1.0 are plotted against the radius ratio, α, in Fig. 1 for Reynolds and Prandtl numbers of 10^5 and 0.72, respectively. The effect of the relative velocity is seen to decrease with a decreasing value of α as was the case of the effect on the friction factor as seen previously in ref. [2]. It was also observed that the effect of the relative velocity on heat transfer is the opposite of that of the friction factor; i.e. the heat transfer increases with an increasing value of the relative velocity.

NOMENCLATURE

h	heat transfer coefficient
k	thermal conductivity
Nu	Nusselt number
Pr	Prandtl number
R	radius
Re	Reynolds number
T	temperature
u	fluid velocity in x-direction
U	core velocity
U^*	dimensionless relative velocity, U/u_b.

Greek symbols

α	radius ratio, R_i/R_o
θ^*	influence coefficients.

Subscripts

b	bulk
i	inner
ii	constant heat rate at the inner wall with the outer insulated
j,k	i or o
o	outer
oo	constant heat rate at the outer wall with the inner insulated
R	radius.

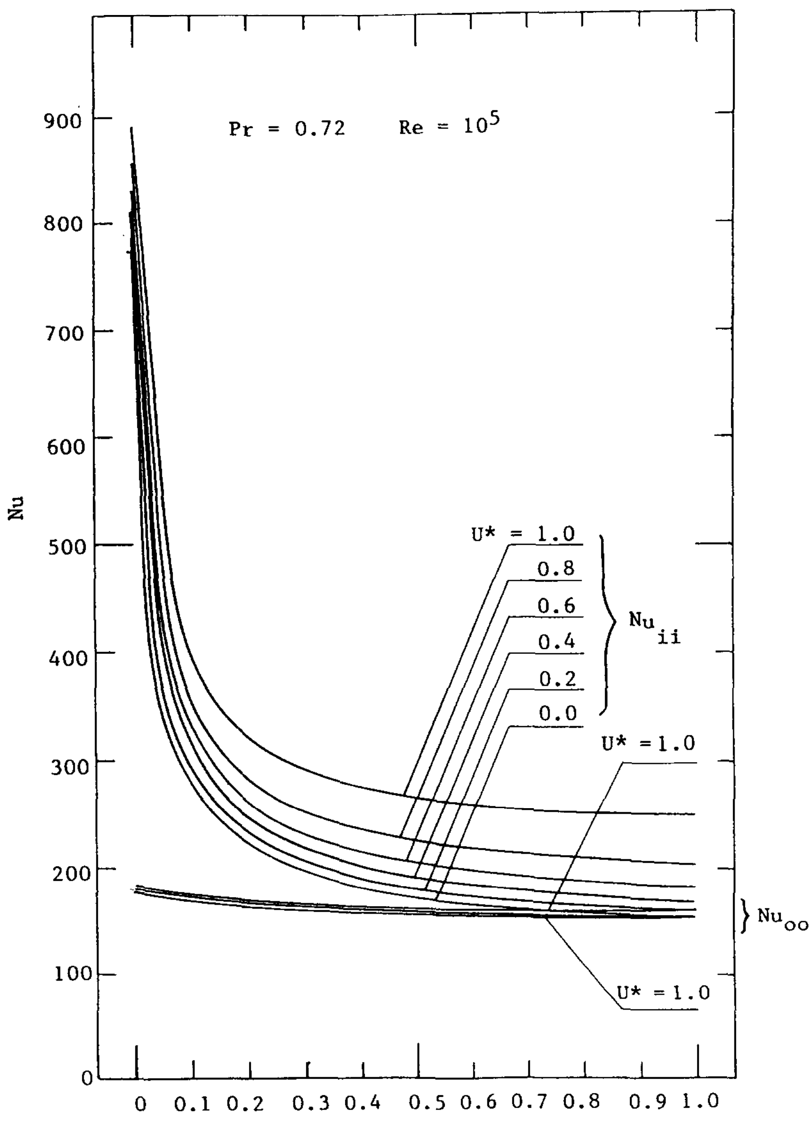

FIG. 1. Nusselt numbers.

Table 1. Nusselt numbers and influence coefficients, fully developed turbulent flow in concentric annuli with moving cores; $0 \leqslant U^* \leqslant 1$

U^*	Re:	10^4				3×10^4				10^5				10^6			
	Pr	Nu_{ii}	Nu_{oo}	θ_i^*	θ_o^*	Nu_{ii}	Nu_{oo}	θ_i^*	θ_o^*	Nu_{ii}	Nu_{oo}	θ_i^*	θ_o^*	Nu_{ii}	Nu_{oo}	θ_i^*	θ_o^*
$\alpha = 2$																	
0.0		41.41	30.40	0.4650	0.0818	89.37	65.47	0.4215	0.0719	224.1	163.2	0.3806	0.0632	1442	1037	0.3101	0.0470
0.2		42.46	30.47	0.4589	0.0817	92.44	65.73	0.4177	0.0712	233.7	163.9	0.3797	0.0623	1523	1041	0.3137	0.0467
0.4	0.72	43.76	30.59	0.4527	0.0812	96.24	66.10	0.4143	0.0702	245.7	164.9	0.3796	0.0610	1626	1047	0.3193	0.0463
0.6		45.43	30.78	0.4469	0.0802	101.2	66.63	0.4115	0.0686	261.6	166.2	0.3808	0.0592	1768	1056	0.3288	0.0455
0.8		47.74	31.05	0.4420	0.0785	108.1	67.35	0.4104	0.0664	284.3	168.1	0.3848	0.0569	1981	1070	0.3464	0.0442
1.0		51.45	31.44	0.4404	0.0756	119.4	68.41	0.4148	0.0631	323.3	170.8	0.3974	0.0535	2383	1095	0.3874	0.0416
0.0		137.2	99.60	0.1347	0.0241	332.6	241.7	0.1224	0.0210	920.3	667.9	0.1158	0.0193	7295	5414	0.1101	0.0155
0.2		140.3	99.14	0.1323	0.0241	341.6	240.9	0.1203	0.0208	948.2	666.4	0.1141	0.0189	7459	5478	0.1080	0.0160
0.4	10	144.3	98.74	0.1300	0.0240	353.1	240.4	0.1183	0.0204	984.2	665.5	0.1125	0.0184	7662	5558	0.1063	0.0164
0.6		149.8	98.46	0.1279	0.0237	368.9	240.1	0.1165	0.0198	1034	665.4	0.1113	0.0178	7943	5659	0.1054	0.0168
0.8		157.9	98.31	0.1262	0.0231	392.5	240.2	0.1155	0.0190	1108	666.3	0.1108	0.0169	8428	5786	0.1075	0.0170
1.0		171.8	98.35	0.1259	0.0221	435.2	240.9	0.1167	0.0179	1250	668.9	0.1133	0.0157	9767	5943	0.1214	0.0164
0.0		237.5	172.5	0.0523	0.0097	590.9	429.5	0.0456	0.0080	1675	1217	0.0429	0.0072	16478	11976	0.0468	0.0050
0.2		242.4	171.4	0.0513	0.0098	604.7	427.3	0.0447	0.0079	1717	1212	0.0420	0.0071	16863	11918	0.0461	0.0055
0.4	50	248.8	170.3	0.0503	0.0099	622.9	425.5	0.0438	0.0078	1773	1208	0.0412	0.0069	17253	11823	0.0455	0.0059
0.6		257.8	169.4	0.0493	0.0098	648.5	424.0	0.0430	0.0076	1852	1205	0.0405	0.0067	17578	11672	0.0454	0.0063
0.8		271.3	168.7	0.0485	0.0097	687.6	423.0	0.0424	0.0074	1973	1204	0.0401	0.0063	17658	11422	0.0462	0.0065
1.0		295.3	168.3	0.0482	0.0093	761.2	422.9	0.0426	0.0070	2214	1205	0.0407	0.0059	17756	10960	0.0511	0.0061
$\alpha = 0.5$																	
0.0		32.00	28.97	0.3431	0.1684	69.05	62.33	0.3123	0.1510	173.3	155.8	0.2827	0.1348	1118	996.5	0.2305	0.1061
0.2		32.91	28.91	0.3371	0.1691	71.63	62.37	0.3079	0.1501	181.2	156.0	0.2804	0.1331	1184	998.1	0.2313	0.1047
0.4	0.72	34.08	28.94	0.3309	0.1687	74.97	62.59	0.3036	0.1481	191.6	156.7	0.2786	0.1303	1272	1002	0.2334	0.1026
0.6		35.66	29.06	0.3250	0.1668	79.50	63.04	0.2998	0.1446	205.9	157.9	0.2778	0.1261	1396	1010	0.2381	0.0993
0.8		37.97	29.31	0.3197	0.1627	86.21	63.81	0.2974	0.1390	227.5	159.9	0.2793	0.1200	1592	1024	0.2492	0.0944
1.0		42.03	29.76	0.3176	0.1548	98.13	65.05	0.3004	0.1301	267.9	163.2	0.2890	0.1108	1997	1052	0.2831	0.0866
0.0		112.2	99.30	0.1067	0.0518	271.7	240.4	0.0965	0.0460	750.5	663.2	0.0909	0.0427	5840	5205	0.0842	0.0378
0.2		114.6	98.17	0.1040	0.0521	278.8	238.2	0.0941	0.0455	773.2	658.4	0.0889	0.0419	5989	5248	0.0817	0.0380
0.4	10	118.0	97.17	0.1014	0.0520	288.8	236.4	0.0917	0.0447	804.5	654.5	0.0868	0.0407	6190	5318	0.0794	0.0381
0.6		123.0	96.33	0.0988	0.0512	303.3	235.0	0.0895	0.0432	850.0	651.7	0.0851	0.0390	6486	5422	0.0777	0.0378
0.8		131.0	95.70	0.0966	0.0496	326.7	234.2	0.0879	0.0411	923.9	650.5	0.0840	0.0366	7017	5573	0.0786	0.0370
1.0		146.4	95.37	0.0958	0.0465	373.9	234.2	0.0887	0.0378	1080	651.9	0.0861	0.0332	8525	5779	0.0919	0.0345
0.0		197.6	174.0	0.0426	0.0209	490.8	432.2	0.0368	0.0176	1389	1222	0.0343	0.0161	13487	11924	0.0365	0.0154
0.2		201.1	171.6	0.0415	0.0213	501.3	427.1	0.0357	0.0175	1422	1210	0.0333	0.0158	13771	11885	0.0351	0.0157
0.4	50	206.4	169.3	0.0403	0.0215	516.6	422.5	0.0347	0.0173	1470	1199	0.0324	0.0154	14091	11819	0.0336	0.0159
0.6		214.4	167.3	0.0392	0.0214	540.0	418.5	0.0336	0.0168	1542	1190	0.0315	0.0147	14415	11690	0.0324	0.0157
0.8		227.8	165.5	0.0381	0.0209	579.0	415.4	0.0328	0.0160	1664	1183	0.0308	0.0138	14661	11437	0.0322	0.0150
1.0		254.8	164.1	0.0375	0.0198	661.6	413.4	0.0329	0.0148	1935	1180	0.0313	0.0125	15412	10883	0.0378	0.0130

$\alpha = 0.8$																	
0.0		29.50	28.65	0.2835	0.2261	63.44	61.55	0.2575	0.2044	158.7	153.7	0.2326	0.1837	1020	985.0	0.1886	0.1474
0.2		30.35	28.52	0.2776	0.2276	65.79	61.43	0.2530	0.2035	165.9	153.7	0.2298	0.1815	1080	984.9	0.1884	0.1452
0.4	0.72	31.46	28.47	0.2715	0.2272	68.92	61.53	0.2848	0.2007	175.6	154.1	0.2273	0.1776	1161	987.4	0.1890	0.1416
0.6		33.01	28.55	0.2655	0.2244	73.31	61.92	0.2440	0.1955	189.3	155.2	0.2255	0.1714	1279	994.2	0.1916	0.1362
0.8		35.36	28.80	0.2597	0.2182	80.06	62.70	0.2408	0.1870	210.9	157.3	0.2257	0.1620	1472	1008	0.1996	0.1281
1.0		39.78	29.30	0.2570	0.2054	92.95	64.12	0.2429	0.1730	254.2	161.0	0.2337	0.1474	1902	1040	0.2304	0.1156
0.0		104.0	100.1	0.0892	0.0708	251.2	241.8	0.0804	0.0634	692.1	665.9	0.0755	0.0593	5336	5147	0.0689	0.0534
0.2		105.9	98.51	0.0866	0.0714	257.1	238.6	0.0779	0.0628	711.7	658.8	0.0733	0.0581	5467	5173	0.0663	0.0533
0.4	10	108.9	97.09	0.0839	0.0713	265.9	235.9	0.0754	0.0615	739.6	652.7	0.0712	0.0563	5653	5229	0.0638	0.0530
0.6		113.5	95.87	0.0813	0.0701	279.4	233.8	0.0731	0.0594	782.2	648.1	0.0692	0.0537	5940	5328	0.0619	0.0521
0.8		121.4	94.93	0.0789	0.0675	302.4	232.3	0.0712	0.0560	854.7	645.6	0.0678	0.0499	6477	5488	0.0622	0.0503
1.0		138.0	94.37	0.0777	0.0625	353.2	232.0	0.0717	0.0507	1022	646.3	0.0694	0.0445	8104	5728	0.0750	0.0461
0.0		183.5	176.3	0.0359	0.0286	454.8	436.9	0.0308	0.0243	1284	1233	0.0286	0.0244	12 378	11 913	0.0298	0.0228
0.2		186.2	173.0	0.0348	0.0293	463.0	429.8	0.0297	0.0242	1312	1216	0.0276	0.0220	12 599	11 873	0.0283	0.0230
0.4	50	190.5	169.9	0.0337	0.0296	476.2	423.4	0.0287	0.0239	1353	1201	0.0266	0.0213	12 862	11 816	0.0267	0.0229
0.6		197.9	167.0	0.0325	0.0295	497.6	417.7	0.0276	0.0232	1420	1187	0.0257	0.0203	13 152	11 705	0.0252	0.0223
0.8		210.9	164.6	0.0314	0.0287	535.7	413.1	0.0267	0.0219	1539	1177	0.0249	0.0189	13 415	11 455	0.0246	0.0209
1.0		239.8	162.6	0.0306	0.0268	624.3	410.1	0.0267	0.0199	1828	1172	0.0253	0.0168	14 499	10 836	0.0309	0.0175
$\alpha = 0.99$																	
0.0		28.75	28.72	0.2566	0.2540	61.67	61.59	0.2327	0.2303	153.9	153.7	0.2097	0.2075	986.2	984.7	0.1695	0.1676
0.2		29.57	28.54	0.2509	0.2557	63.93	61.39	0.2281	0.2293	160.9	153.5	0.2068	0.2050	1044	983.6	0.1688	0.1649
0.4	0.72	30.65	28.46	0.2449	0.2554	66.97	61.44	0.2235	0.2261	170.2	153.8	0.2041	0.2004	1122	985.4	0.1688	0.1605
0.6		32.18	28.52	0.2388	0.2522	71.30	61.79	0.2190	0.2200	183.8	154.8	0.2020	0.1931	1238	991.7	0.1706	0.1539
0.8		34.57	28.77	0.2332	0.2446	78.09	62.60	0.2155	0.2098	205.4	157.0	0.2016	0.1819	1431	1006	0.1773	0.1441
1.0		39.19	29.31	0.2299	0.2289	91.66	64.12	0.2171	0.1926	250.7	161.0	0.2087	0.1641	1878	1039	0.2070	0.1287
0.0		101.0	100.8	0.0808	0.0800	243.7	243.3	0.0726	0.0718	670.5	669.4	0.0680	0.0673	5148	5139	0.0617	0.0610
0.2		102.8	99.05	0.0782	0.0808	249.1	239.7	0.0702	0.0712	688.5	661.1	0.0659	0.0659	5270	5156	0.0591	0.0607
0.4	10	105.5	97.43	0.0756	0.0806	257.3	236.5	0.0677	0.0697	714.8	653.9	0.0638	0.0638	5446	5205	0.0567	0.0601
0.6		109.9	96.02	0.0730	0.0792	270.2	234.0	0.0654	0.0671	755.8	648.4	0.0618	0.0607	5726	5301	0.0546	0.0588
0.8		117.7	94.92	0.0706	0.0761	293.0	232.3	0.0634	0.0630	827.4	645.2	0.0603	0.0562	6260	5466	0.0546	0.0565
1.0		135.0	94.27	0.0693	0.0698	345.9	231.8	0.0637	0.0565	1001	645.8	0.0616	0.0495	7956	5725	0.0674	0.0513
0.0		178.3	177.9	0.0326	0.0322	441.1	440.3	0.0278	0.0276	1244	1242	0.0258	0.0255	11 948	11 927	0.0267	0.0264
0.2		180.5	174.2	0.0315	0.0331	448.3	432.3	0.0268	0.0275	1268	1222	0.0248	0.0250	12 140	11 885	0.0251	0.0265
0.4	50	184.4	170.7	0.0304	0.0335	460.3	424.9	0.0258	0.0271	1307	1204	0.0238	0.0242	12 373	11 830	0.0235	0.0263
0.6		191.3	167.4	0.0293	0.0334	480.6	418.4	0.0247	0.0262	1370	1189	0.0229	0.0230	12 641	11 727	0.0219	0.0255
0.8		204.1	164.6	0.0282	0.0325	518.0	413.1	0.0238	0.0247	1487	1177	0.0221	0.0213	12 901	11 480	0.0212	0.0236
1.0		234.1	162.4	0.0274	0.0300	610.3	409.6	0.0237	0.0222	1787	1171	0.0224	0.0187	14 153	10 824	0.0280	0.0195

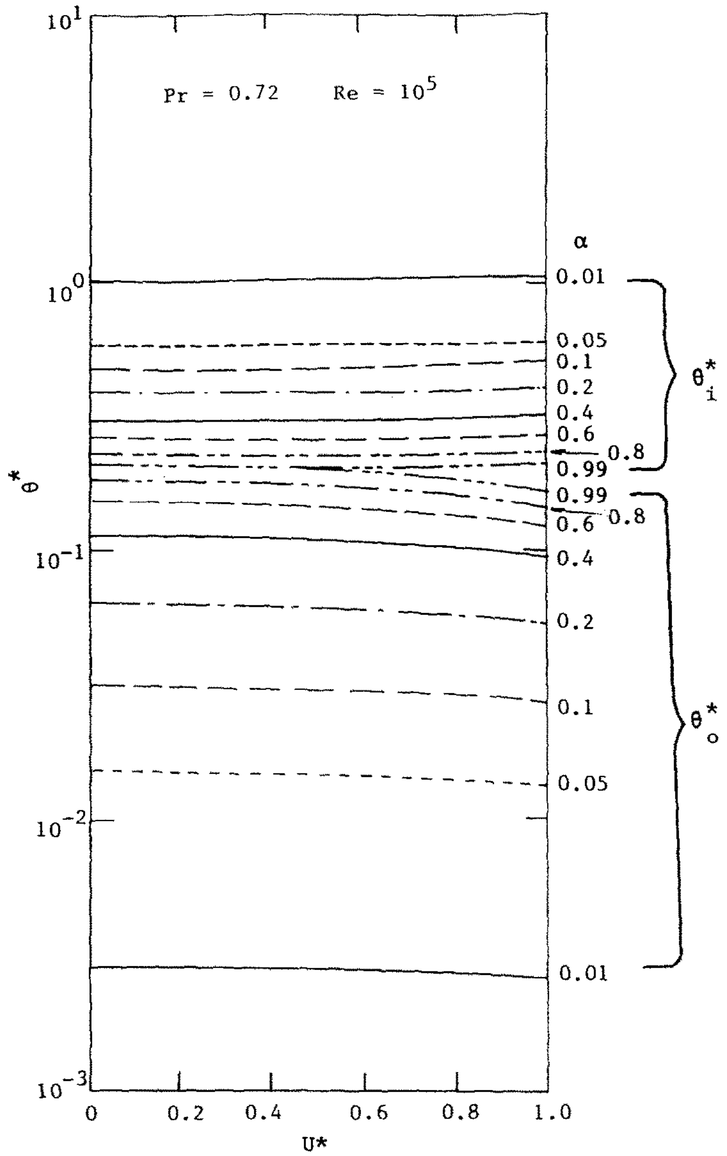

FIG. 2. Influence coefficients.

Figure 1 also illustrates that while the effect of the relative velocity on Nu_{ii} (the Nusselt number for the case of the inner wall only heated and the outer insulated) is significant, the effect on Nu_{oo} (the case of the outer wall heated and the inner insulated) is negligibly small. The same trend can be observed for the effect of the radius ratio on heat transfer. The analytical study of Kays and Crawford [4] of concentric annuli with stationary cores (i.e. $U^* = 0$) is compared with that of the present analysis. Despite the different method of analysis employed, it was seen that the agreement is very good for the range of the parameters studied. No comparison was made for the case of concentric annuli with moving cores (i.e. $U^* > 0$) with other works as there is none available in the open literature.

From Table 1, it can be seen that the Nusselt numbers increase with increasing values of Prandtl number but its combined effect with that of the relative velocity on the Nusselt number are similar to those observed in Fig. 1.

The effects of α and U^* on the influence coefficients, θ_i^* and θ_o^*, are shown in Fig. 2 for Reynolds and Prandtl numbers of 10^5 and 0.72, respectively.

4. CONCLUDING REMARKS

A complete solution for the fully developed turbulent flow and heat transfer in concentric annuli with moving cores has been made.

The study showed that for equal conditions, increasing relative velocity were observed for the following changes:

(i) a decrease in friction factor; and
(ii) an increase in Nusselt number.

However, the effect of the relative velocity seems to diminish with decreasing value of the radius ratio and especially for the case of the outer wall heated and the inner wall insulated.

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Study of heat transfer from buried nuclear waste canisters

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INTRODUCTION

IT HAS been proposed that radioactive waste from nuclear power plants be disposed of in cylindrical containers by burying them under the surface of the earth. To carry out safety analysis and to gauge the impact of this proposal on the environment it is necessary to determine flow patterns and heat transfer rates in the vicinity of these containers. The present work gives a summary of analytical and numerical results for temperature distribution in and around a canister buried in a saturated porous medium. Heating of the canister surface takes place because of the decay of radioactive waste contained within it. It is important to know the maximum and minimum temperatures on the cylinder surface since they decide the magnitude of the transport coefficients and the extent of the thermal stresses. Heat transfer from the cylinder to its surroundings will occur due to one of the following mechanisms: conduction, buoyancy-driven convection of the pore fluid and forced convection due to natural ground water movement. Solutions for these problems are available when the surface of the cylinder has a prescribed temperature. Results have been presented here for a single and an array of cylinders with specified heat flux on their surface.

FORMULATION

Fluid flow in a saturated homogeneous isotropic porous medium is taken to be governed by Darcy's law,

$$\mathbf{u} = -K(\nabla p + \rho g \mathbf{k})/\mu \tag{1}$$

the incompressibility constraint $\nabla \cdot \mathbf{u} = 0$ and the energy transport equation,

$$T_t + \mathbf{u} \cdot \nabla T = \alpha \nabla^2 T. \tag{2}$$

In the absence of buoyancy effects ρ is a constant and $\nabla \cdot \mathbf{u} = \nabla \cdot K \nabla p = 0$. For flow past a single cylinder buried in a uniform medium, K is a constant and $\nabla^2 p = 0$. Using potential theory the velocity components can be determined as $u - iv = U.\,(1 - R^2/z^2)$ where $z = x + iy$ and $i = \sqrt{-1}$. For an array of canisters we solve the equation $\nabla \cdot K \nabla p = 0$ numerically by assigning a small value for K over the cylinders and unit value in the flow region. Equation (2) has been solved here subject to the constant heat flux condition, $-T_r\,(r = R) = q$. The results are presented in dimensionless form using R as the length scale, R^2/α as the time scale, the approach velocity U as the velocity scale and qR as the temperature scale. In free convection problems the velocity scale is α/R. Convection problems are assumed to have reached steady state since they occur in boundary layers.

RESULTS

Conduction limit

In the absence of a super-imposed flow the conduction problem follows the dimensionless equation,

$$T_t = \nabla^2 T$$

subject to $T(t = 0) = 0$. This equation can be solved by Fourier transforms. For an isothermal boundary condition $T(r = 1) = 1$ the solution for the wall heat flux is

$$-T_r|_{r=1} = \left(\frac{2}{\pi}\right)^2 \int_0^\infty \frac{1}{\beta N(\beta)} \exp(-\beta^2 t)\, d\beta$$

where $N(\beta) = J_0^2(\beta) + Y_0^2(\beta)$. This integral is evaluated numerically by Simpson's rule. The conduction solution also describes the local heat flux for steady forced flow parallel to the axis of a cylinder with t replaced by z/Pe. The latter problem has been solved in [1] using boundary-layer analysis. The two solutions are compared in Table 1.

For a heat flux boundary condition $(-T_r(r = 1) = 1)$ we solve for the wall temperature as,

$$T(1, t) = \int_0^\infty \frac{R^2(\beta)}{\beta N(\beta)} [1 - \exp(-\beta^2 t)]\, d\beta$$

where $N(\beta) = J_1^2(\beta) + Y_1^2(\beta)$ and $R(\beta) = J_0(\beta) Y_1(\beta) - J_1(\beta) Y_0(\beta)$. The value of T attained by an isolated canister can increase further if more canisters are present in its neighbourhood. Consider a symmetric array of five canisters, four of which are placed on a square edge d and the fifth is placed at the centre. The temperature of the central canister is obtained by the principle of linear superposition. Calculations show that the minimum temperature is within 98% of the maximum temperature in Table 2.

Free convection [2]

The boundary-layer form of equations governing buoyancy-driven flow and heat transfer are given below.

Table 1. Comparison of heat flux values on an isothermal cylinder

t, z/Pe	Present	[1]
0.5	2.081	2.06
1	1.649	1.597
5	1.071	1.107